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(54) REFRIGERATION CYCLE DEVICE, AIR CONDITIONER, AND HEAT EXCHANGER

(57) An air conditioner (1) is a refrigeration cycle apparatus in which incompatible oil is employed as refrigerating machine oil, and includes a compressor (10) to compress refrigerant, a first heat exchanger (30) to condense the refrigerant output from the compressor, a pressure reducing device (40) to reduce a pressure of the refrigerant output from the first heat exchanger, and a second heat exchanger (50) to evaporate the refrigerant output from the pressure reducing device and output the resultant refrigerant to the compressor. The second heat exchanger includes a heat transfer tube having a groove formed on an inner surface of the heat transfer tube. The groove of the heat transfer tube is formed such that an inner surface area of the heat transfer tube on a downstream side of the heat transfer tube is smaller than an inner surface area on an upstream side of the heat transfer tube.

FIG.7

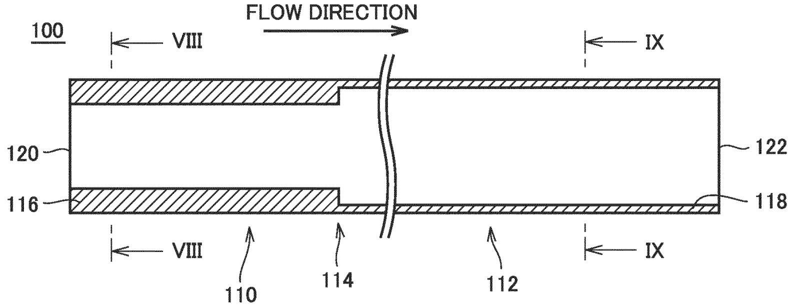

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FIG.8

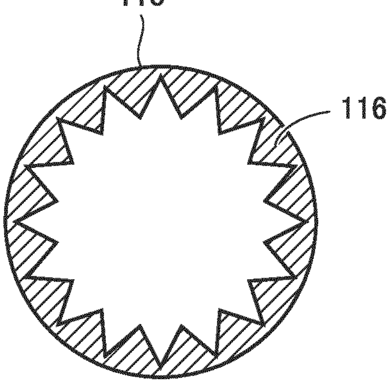
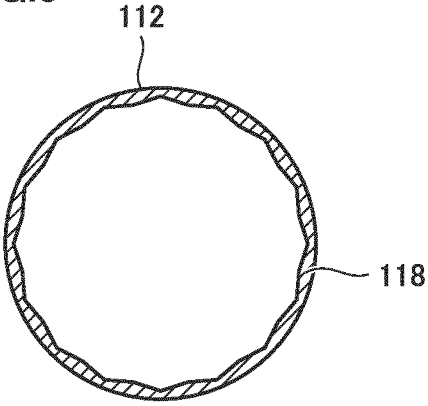


FIG.9



Description

TECHNICAL FIELD

[0001] The present disclosure relates to a refrigeration cycle apparatus, an air conditioner, and a heat exchanger.

BACKGROUND AR

[0002] WO2019/180817 (PTL 1) discloses a heat exchanger used for a refrigeration cycle apparatus. In this heat exchanger, grooves are provided on an inner surface of a heat transfer tube. Accordingly, increase of the surface area of the tube inner surface and agitation of fluid, for example, are achieved to thereby enhance the heat transfer capability of the heat exchanger (see PTL 1).

[0003] Japanese Patent Laying-Open No. H4-45753 (PTL 2) also discloses such a heat exchanger. In this heat exchanger, at least two types of helical grooves that are different in groove depth are provided on an inner surface of a heat transfer tube, and the groove depth is made smaller on the fluid inlet side of the heat transfer tube and larger on the fluid outlet side thereof, in consideration of the heat transfer capability and the pressure loss (see PTL 2).

CITATION LIST

PATENT LITERATURE

[0004]

PTL 1: WO2019/180817
PTL 2: Japanese Patent Laying-Open No. H4-45753

SUMMARY OF INVENTION

TECHNICAL PROBLEM

[0005] Refrigerating machine oil having weak compatibility with liquid refrigerant (the oil is hereinafter referred to as "incompatible oil") can be used to reduce the amount of refrigerant dissolved in the refrigerating machine oil, and thereby reduce the amount of refrigerant enclosed in a refrigeration cycle. "Incompatible oil" refers to a refrigerating machine oil in which the amount of the oil and refrigerant that dissolve in each other is smaller so that the oil and the refrigerant are likely to be separated into two layers, relative to "compatible oil" in which a larger amount of the oil and refrigerant dissolve in each other. While it is difficult to clearly distinguish the compatible oil and the incompatible oil from each other, those skilled in the art would be able to identify the incompatible oil as being an oil in which the amount of the oil and refrigerant that dissolve in each other is apparently smaller relative to the compatible oil.

[0006] In a low-pressure side heat exchanger (evaporator), the flow regime of refrigerant on the downstream side where the dryness fraction of the refrigerant is higher is annular flow or annular mist flow, and thus the liquid phase is pushed toward the wall surface to flow along the tube wall, while the gas phase flows through the tube center. Therefore, if incompatible oil is used for a refrigeration cycle apparatus, the oil that is separated from refrigerant on the downstream side may form an oil film and remain on the tube wall due to its high viscosity. If the oil film is formed on the tube wall, the heat transfer capability of the heat exchanger is deteriorated and the pressure loss is increased.

[0007] The present disclosure is made to solve such a problem, and an object of the present disclosure is to suppress, for a refrigeration cycle apparatus in which incompatible oil is used as refrigerating machine oil, deterioration of the heat transfer capability and increase of the pressure loss, in a low-pressure side heat exchanger.

SOLUTION TO PROBLEM

[0008] A refrigeration cycle apparatus in the present disclosure is a refrigeration cycle apparatus in which incompatible oil is employed as refrigerating machine oil, and includes: a compressor configured to compress refrigerant; a first heat exchanger configured to condense the refrigerant output from the compressor; a pressure reducing device configured to reduce a pressure of the refrigerant output from the first heat exchanger; and a second heat exchanger configured to evaporate the refrigerant output from the pressure reducing device to output the refrigerant to the compressor. The second heat exchanger includes a heat transfer tube having a groove formed on an inner surface of the heat transfer tube. The groove of the heat transfer tube is formed such that an inner surface area per unit length of the heat transfer tube on a downstream side of the heat transfer tube is smaller than an inner surface area per unit length of the heat transfer tube on an upstream side of the heat transfer tube.

ADVANTAGEOUS EFFECTS OF INVENTION

[0009] The refrigeration cycle apparatus enables suppression of deterioration of the heat transfer capability and suppression of increase of the pressure loss, in the second heat exchanger (low-pressure side heat exchanger).

BRIEF DESCRIPTION OF DRAWINGS

[0010]

Fig. 1 is an overall configuration diagram of an air conditioner shown as one example of a refrigeration cycle apparatus according to Embodiment 1.
Fig. 2 shows a flow of refrigerant in the air condition-

er.

Fig. 3 shows a flow of refrigerant during heating operation.

Fig. 4 schematically shows an influence of the oil circulation rate on the capacity ratio of a refrigeration cycle.

Fig. 5 schematically shows a state of refrigerant and refrigerating machine oil that flow through a heat transfer tube of a low-pressure side heat exchanger, when incompatible oil is used as refrigerating machine oil.

Fig. 6 is a Baker map showing flow regimes of gas-liquid two-phase refrigerant flowing through the heat transfer tube.

Fig. 7 conceptually illustrates a configuration of the inside of a heat transfer tube in an indoor heat exchanger shown in Fig. 1.

Fig. 8 shows one example of a cross section of a first part of the heat transfer tube.

Fig. 9 shows one example of a cross section of a second part of the heat transfer tube.

Fig. 10 shows one example of a specific configuration of the indoor heat exchanger shown in Fig. 1.

Fig. 11 is a block diagram showing one example of a hardware configuration of a controller.

Fig. 12 is a flowchart illustrating one example of a process performed by the controller in Embodiment 2.

Fig. 13 is a flowchart illustrating one example of a process performed by the controller in Modification 1 of Embodiment 2.

Fig. 14 is a flowchart illustrating one example of a process performed by the controller in Modification 2 of Embodiment 2.

Fig. 15 shows a placement of a sensor that detects the refrigerant flow regime in the indoor heat exchanger in Embodiment 3 and its modifications.

DESCRIPTION OF EMBODIMENTS

[0011] In the following, embodiments of the present disclosure are described in detail with reference to the drawings. In the drawings, the same or corresponding parts are denoted by the same reference characters, and description thereof is not herein repeated.

Embodiment 1

[0012] Fig. 1 is an overall configuration diagram of an air conditioner shown as one example of a refrigeration cycle apparatus according to Embodiment 1. Referring to Fig. 1, an air conditioner 1 includes an outdoor unit 2 and an indoor unit 3. Indoor unit 3 is disposed in a target space (indoors) to be air-conditioned by air conditioner 1, and outdoor unit 2 is disposed outside the target space (outdoors, for example).

[0013] Outdoor unit 2 includes a compressor 10, a four-way valve 20, an outdoor heat exchanger 30, a fan 32, a pressure reducing device 40, tubes 62 to 66, 72, temperature sensors 81 to 84, and a controller 90. Indoor unit 3 includes an indoor heat exchanger 50, a fan 52, and temperature sensors 85, 86. Outdoor unit 2 and indoor unit 3 are connected to each other through tubes 68, 70.

[0014] Tube 62 connects a discharge port of compressor 10 and a port p1 of four-way valve 20 to each other. Tube 64 connects a port p2 of four-way valve 20 and outdoor heat exchanger 30 to each other. Tube 66 connects outdoor heat exchanger 30 and pressure reducing device 40 to each other. Tube 68 connects pressure reducing device 40 and indoor heat exchanger 50 to each other. Tube 70 connects indoor heat exchanger 50 and a port p3 of four-way valve 20 to each other. Tube 72 connects a port p4 of four-way valve 20 and a suction port of compressor 10 to each other.

[0015] Compressor 10 compresses refrigerant sucked from tube 72 and outputs the resultant refrigerant to tube 62. Compressor 10 is configured to be capable of adjusting the operating frequency in accordance with a control signal from controller 90. The operating frequency of compressor 10 is adjusted to adjust the output of compressor 10. Any of various type of compressors may be employed as compressor 10. For example, a compressor of rotary type, reciprocating type, scroll type, screw type, or the like may be employed.

[0016] Four-way valve 20 allows port p1 and port p2 to communicate with each other and allows port p3 and port p4 to communicate with each other. Accordingly, tube 62 and tube 64 are connected together and tube 70 and tube 72 are connected together. Four-way valve 20 is capable of switching the state of connection between ports p1 to p4, in accordance with a control signal from controller 90. Specifically, during heating operation, four-way valve 20 allows port p1 and port p3 to communicate with each other and allows port p2 and port p4 to communicate with each other. Accordingly, during heating operation, tube 62 and tube 70 are connected together, and tube 64 and tube 72 are connected together.

[0017] Outdoor heat exchanger 30 is configured to cause refrigerant flowing through a heat transfer tube disposed therein to exchange heat with outdoor air. In outdoor heat exchanger 30, high-temperature high-pressure superheated vapor (refrigerant) flowing therein from tube 64 exchanges heat with outdoor air (discharges heat) and is accordingly condensed into liquid phase and the resultant liquid refrigerant is output to tube 66. During heating operation, refrigerant flowing from tube 66 into outdoor heat exchanger 30 exchanges, in outdoor heat exchanger 30, heat with outdoor air (sucks heat) and is accordingly evaporated into superheated vapor that is output to tube 64. Fan 32 is disposed along with outdoor heat exchanger 30 to blow outdoor air toward outdoor heat exchanger 30.

[0018] Pressure reducing device 40 is configured in the form of an electronic expansion valve, for example, and its opening Op is adjusted in accordance with a control signal from controller 90. As opening Op is changed in the closing direction, the refrigerant pressure on the outlet side of pressure reducing device 40 is decreased and the refrigerant dryness fraction is increased. As opening Op is changed in the opening direction, the refrigerant pressure on the outlet side of pressure reducing device 40 is increased and the refrigerant dryness fraction is decreased. Pressure reducing device 40 reduces the pressure of refrigerant that is output from outdoor heat exchanger 30 to tube 66 and outputs the resultant refrigerant to tube 68. During heating operation, pressure reducing device 40 reduces the pressure of refrigerant that is output from indoor heat exchanger 50 to tube 68 and outputs the resultant refrigerant to tube 66.

[0019] Indoor heat exchanger 50 is configured to cause refrigerant flowing through a heat transfer tube disposed therein to exchange heat with air in a target space to be air-conditioned. In indoor heat exchanger 50, refrigerant flowing therein from tube 68 exchanges heat with air in the target space (sucks heat) and is accordingly evaporated into superheated vapor that is output to tube 70. During heating operation, high-temperature high-pressure superheated vapor (refrigerant) flowing from tube 70 into indoor heat exchanger 50 exchanges, in indoor heat exchanger 50, heat with air in the target space (discharges heat) and is accordingly condensed into liquid phase, and the resultant liquid refrigerant is output to tube 68. Fan 52 is disposed along with indoor heat exchanger 50 to blow air toward indoor heat exchanger 50.

[0020] Temperature sensor 81 detects temperature T1 of refrigerant on the inlet side of outdoor heat exchanger 30 (output side thereof during heating operation), and outputs the detected temperature value to controller 90. Temperature sensor 82 detects temperature T2 of refrigerant on the output side of outdoor heat exchanger 30 (inlet side thereof during heating operation), and outputs the detected temperature value to controller 90. Temperature sensor 83 detects temperature T3 of the heat transfer tube of outdoor heat exchanger 30 (the condensation temperature during cooling operation, evaporation temperature during heating operation), and outputs the detected temperature value to controller 90.

[0021] Temperature sensor 84 detects temperature T4 (outdoor air temperature) of the place where outdoor unit 2 (outdoor heat exchanger 30) is disposed, and outputs the detected temperature value to controller 90. Temperature sensor 85 detects temperature T5 of the heat transfer tube of indoor heat exchanger 50 (evaporation temperature during cooling operation, condensation temperature during heating operation), and outputs the detected temperature value to controller 90. Temperature sensor 86 detects temperature T6 (indoor temperature) of the target space in which indoor unit 3 (indoor heat exchanger 50) is disposed, and outputs the detected temperature value to controller 90.

[0022] Controller 90 performs control of each device in air conditioner 1. Principal control to be performed by controller 90 is as follows: controller 90 controls the operating frequency of compressor 10 and opening Op of pressure reducing device 40, based on detected values of temperature sensors 81 to 86, such that air conditioner 1 performs desired air conditioning. Controller 90 also switches the state of four-way valve 20, depending on whether to perform cooling operation or heating operation.

[0023] Fig. 2 shows a flow of refrigerant in air conditioner 1. In Fig. 2, the flow of refrigerant during cooling operation is shown. Referring to Fig. 2, refrigerant changed into high-temperature high-pressure vapor state by compressor 10 is supplied through four-way valve 20 to outdoor heat exchanger 30. In outdoor heat exchanger 30, the refrigerant exchanges heat with outdoor air (discharges heat) and is accordingly condensed (liquefied) into high-pressure liquid refrigerant.

[0024] The refrigerant passing through outdoor heat exchanger 30 is reduced in pressure by pressure reducing device 40 into low-temperature low-pressure refrigerant that is supplied to indoor heat exchanger 50. In indoor heat exchanger 50 (low-pressure side heat exchanger), the refrigerant exchanges heat with air in a target space (sucks heat) and is accordingly evaporated (vaporized) into low-pressure gas refrigerant. After this, the refrigerant is passed through four-way valve 20 and sucked again into compressor 10.

[0025] During heating operation as shown in Fig. 3, four-way valve 20 is switched to reverse the refrigerant flow direction relative to the refrigerant flow direction during cooling operation. In this case, therefore, indoor heat exchanger 50 is the high-pressure side and outdoor heat exchanger 30 is the low-pressure side. In the following, however, cooling operation is described in which outdoor heat exchanger 30 is the high-pressure side heat exchanger (first heat exchanger) and indoor heat exchanger 50 is the low-pressure side heat exchanger (second heat exchanger).

[0026] In the refrigeration cycle apparatus, a groove (depression and protrusion) is provided on the inner surface of the heat transfer tube in the heat exchanger, to thereby enable increase of the surface area per tube unit length of the tube inner surface (the surface area is hereinafter referred to as "inner surface area"), and enable enhancement of the heat transfer capability of the heat exchanger.

[0027] In the refrigeration cycle apparatus, oil (refrigerating machine oil) is present in the compressor in order to ensure the lubricity of the compressor. During operation of the compressor, the refrigerating machine oil is discharged to the refrigerant circuit together with flow of refrigerant output from the compressor into the refrigerant circuit. The oil discharged into the refrigerant circuit circulates together with the refrigerant in the refrigerant circuit and then returns to the compressor.

[0028] As this refrigerating machine oil, incompatible oil having low compatibility with liquid refrigerant can be used to reduce the amount of refrigerant dissolved in the refrigerating machine oil and thereby reduce the amount of refrigerant enclosed in the refrigeration cycle apparatus.

[0029] If incompatible oil is used as refrigerating machine oil, however, the oil that is discharged into the refrigerant circuit may form an oil film and remain on the tube wall due to its high viscosity, on the downstream side of the low-pressure side heat exchanger where the dryness fraction of the refrigerant is higher, and consequently cause deterioration of the heat transfer capability of the heat exchanger and increase of the pressure loss. This is detailed in the following.

[0030] Fig. 4 schematically shows an influence of the oil circulation rate on the capacity ratio of a refrigeration cycle. Oil circulation rate is an indicator of the amount of refrigerating machine oil discharged into a refrigerant circuit, and defined for examples as a weight ratio between refrigerant and the refrigerating machine oil that circulate in the refrigerant circuit (weight ratio (wt%) of the oil to the refrigerant). A higher oil circulation rate means that the amount of the oil discharged from the compressor into the refrigerant circuit is larger. The capacity ratio is an indicator of the degree of decrease of the refrigeration cycle capacity under specific operating conditions. In this example, the refrigeration cycle capacity for an oil circulation rate of 0 is defined as 1, and the refrigeration cycle capacity ratio depending on the oil circulation rate is shown.

[0031] Referring to Fig. 4, as the oil circulation rate increases, the refrigeration cycle capacity ratio decreases. If incompatible oil is used as refrigerating machine oil, the oil circulation rate may be higher to thereby cause decrease of the refrigeration cycle capacity. The reason for the decrease of the refrigeration cycle capacity due to increased oil circulation rate as well as the reason for the increase of the oil circulation rate when the incompatible oil is used, are described later herein.

[0032] Fig. 5 schematically shows a state of refrigerant and refrigerating machine oil that flow through a heat transfer tube of the low-pressure side heat exchanger, when incompatible oil is used as refrigerating machine oil. Referring to Fig. 5, in the heat transfer tube of the low-pressure side heat exchanger, the refrigerant flows in the form of gas-liquid two-phase flow made up of a liquid refrigerant 102 and a gas refrigerant 104. The incompatible refrigerating machine oil is present in the form of oil droplets 106 in liquid refrigerant 102.

[0033] On the upstream side of the heat transfer tube, heat exchange between refrigerant and outdoor air does not proceed considerably, and the refrigerant dryness fraction is lower. As the refrigerant flows through the heat transfer tube, the refrigerant exchanges heat with outdoor air (sucks heat) and is accordingly evaporated (vaporized). Thus, on the downstream side of the heat transfer tube, the refrigerant dryness fraction is higher.

[0034] On the upstream side (lower dryness fraction) of the heat transfer tube, the refrigerant flow regime is often slug flow or stratified flow. Flow regime refers to a specific form of flow identified as belonging to the same category among categories into which gas-liquid two-phase flow through a tube is visually classified. When the flow regime is slug flow or stratified flow, oil droplets 106 in liquid refrigerant 102 are caused to flow toward the downstream side, together with liquid refrigerant 102. On the downstream side of the heat transfer tube, the refrigerant dryness fraction increases, so that the flow regime often changes to annular flow or annular mist flow.

[0035] Fig. 6 is a Baker map showing flow regimes of gas-liquid two-phase refrigerant flowing through a heat transfer tube. In Fig. 6, the vertical axis represents the quantity corresponding to the flow rate of the refrigerant, and the horizontal axis represents the quantity corresponding to the ratio of the liquid phase flow to the gas phase flow.

[0036] Referring to Fig. 6, the flow regimes typically include bubble flow, slug flow, stratified flow, annular flow, and annular mist flow, for example. A group of dots 95 is plotted to represent a state of refrigerant flowing through the heat transfer tube, for each refrigerant dryness fraction x. In this example, it is seen that the refrigerant with a lower dryness fraction x flows in the form of slug flow and the refrigerant with a higher dryness fraction x flows in the form of annular flow. In this example, it is also seen that the flow regime of the refrigerant with a dryness fraction x of about 0.2 changes from slug flow to annular flow.

[0037] The dryness fraction at which the refrigerant flow regime changes can be calculated, for example, from the temperature of the heat exchanger (evaporation temperature), the refrigerant flow rate, and the inner diameter of the heat transfer tube, or the like. The region (position) where the flow regime changes in the heat transfer tube can also be estimated from the enthalpies of saturated liquid and saturated vapor at the evaporation temperature and the calculated dryness fraction (the dryness fraction at which the flow regime changes).

[0038] Referring again to Fig. 5, on the downstream side where the refrigerant dryness fraction is higher, the refrigerant flow regime is annular flow or annular mist flow, and thus liquid refrigerant 102 is pushed toward the wall surface to flow along the tube wall. Therefore, if incompatible oil is used as refrigerating machine oil, the oil that is separated from refrigerant on the downstream side may form an oil film 108 on the tube surface due to its high viscosity. In particular, when the heat transfer tube has a deeper groove on its inner surface for enhancing the heat transfer capability, oil film 108 is more likely to be formed on the tube wall.

[0039] As oil film 108 is formed on the tube wall, the oil stays within the heat transfer tube, and consequently the amount of oil returning to the compressor is decreased and the oil circulation rate is increased. Moreover, oil film 108 thus formed increases the pressure loss when refrigerant flows, and hinders heat transfer between refrigerant and the heat transfer tube, and therefore, the heat transfer efficiency is deteriorated. Further, due to the decreased amount of the oil returning to the compressor, lubrication and reliability of the compressor may also be deteriorated. Thus, if incompatible oil is used as refrigerating machine oil, the oil circulation rate is increased, and consequently the capacity ratio of the refrigeration cycle could be deteriorated significantly.

[0040] In view of the above, for air conditioner 1 according to Embodiment 1, incompatible oil is used as refrigerating machine oil, and a groove (depression and protrusion) is provided on the inner surface of the heat transfer tube of indoor heat exchanger 50 (low-pressure side heat exchanger) such that the inner surface area of the heat transfer tube is smaller on the downstream side (high dryness fraction side) of the heat transfer tube than on the upstream side (low dryness fraction side) of the heat transfer tube.

[0041] More specifically, the heat transfer tube is made up of a first part on the upstream side and a second part on the downstream side, and a groove is formed on the inner surface of the heat transfer tube, such that the inner surface area of the second part is smaller than the inner surface area of the first part. The boundary between the first part and the second part is located in a region where the flow regime of refrigerant flowing through the heat transfer tube is changed to annular flow or annular mist flow. Accordingly, even when incompatible oil is used as refrigerating machine oil, the oil separated from refrigerant can be prevented from forming an oil film on the tube wall in indoor heat exchanger 50. Moreover, on the upstream side of the heat transfer tube, an adequate inner surface area is ensured and thus an adequate heat transfer efficiency can be ensured.

[0042] The boundary between the first part and the second part may be located at a position where the APF (Annual performance Factor) of air conditioner 1 has a maximum value, in consideration of the heat transfer capability and the pressure loss of indoor heat exchanger 50. More specifically, while the above-described boundary is located in a region where the refrigerant flow regime is changed, the position of the boundary may be positioned based on the aforementioned region when air conditioner 1 is operated under conditions that makes APF maximum, rather than defining the boundary position from the aforementioned region when air conditioner 1 is operated under predetermined outdoor air conditions, for example. In this way, energy saving of air conditioner 1 can be achieved.

[0043] In Embodiment 1, alkyl benzene oil, for example, is used as incompatible refrigerating machine oil (incompatible oil). The incompatible oil that can be used, however, is not limited to this, and any of other refrigerating machine oils that can be recognized by those skilled in the art as an oil in which the amount dissolved in refrigerant is apparently smaller relative to compatible oil, may be used.

[0044] Fig. 7 conceptually illustrates a configuration of the inside of the heat transfer tube in indoor heat exchanger 50 shown in Fig. 1. Fig. 7 schematically shows a cross section, in the direction of refrigerant flow, of a heat transfer tube 100 from an inlet 120 to an outlet 122 of heat transfer tube 100.

[0045] Referring to Fig. 7, in air conditioner 1 according to Embodiment 1, heat transfer tube 100 of indoor heat exchanger 50 (low-pressure side heat exchanger) includes a first part 110 located upstream with respect to a boundary 114 and a second part 112 located downstream with respect to boundary 114. In order to enhance heat transfer between outdoor air and refrigerant flowing through heat transfer tube 100, a groove is formed on the inner circumferential surface of heat transfer tube 100.

[0046] Fig. 8 shows one example of a cross section of first part 110 of heat transfer tube 100. Fig. 9 shows one example of a cross section of second part 112 of heat transfer tube 100.

[0047] Referring to Figs. 8 and 9, the depth of a groove 118 formed on the inner circumferential surface of second part 112 is shallower than the depth of a groove 116 formed on the inner circumferential surface of first part 110. As such grooves 116, 118 are formed, the inner surface area of second part 112 smaller than the inner surface area of first part 110. The depth of groove 118 formed on the inner circumferential surface of second part 112 may be substantially zero.

[0048] Referring again to Fig. 7, boundary 114 between first part 110 and second part 112 is located in a region where the refrigerant flow regime changes from slug flow or stratified flow to annular flow or annular mist flow. Accordingly, formation of an oil film on the tube wall in second part 112, by oil separated from refrigerant, due to change of the flow regime to annular flow or annular mist flow, can be suppressed. As a result, increase of the pressure loss and deterioration of the heat transfer efficiency due to the oil film can be suppressed. Moreover, decrease of the amount of oil returning to compressor 10 can also be suppressed, and thus deterioration in lubrication and reliability of compressor 10 can also be suppressed. In contrast, in first part 110 in which the flow regime is slug flow or stratified flow, an adequate surface area of the tube inner surface can be ensured to thereby ensure an adequate heat transfer efficiency.

[0049] The position of boundary 114 of heat transfer tube 100 is defined in a region where the refrigerant flow regime changes from slug flow or stratified flow to annular flow or annular mist flow. The region (position) where the refrigerant flow regime changes in heat transfer tube 100 can be estimated, for example, in the following way. Specifically, the dryness fraction of refrigerant at which the flow regime changes can be calculated from temperature T5 (evaporation temperature) of the heat transfer tube in indoor heat exchanger 50 detected by temperature sensor 85, the flow rate of refrigerant flowing through the refrigerant circuit, and the inner diameter of heat transfer tube 100, for example. Then, the region where the refrigerant flow regime is changed can be estimated from the enthalpies of saturated liquid and saturated vapor at the above temperature T5 (evaporation temperature) and the calculated dryness fraction (the dryness fraction at which the flow regime changes).

[0050] Heat transfer tube 100 is actually composed of a plurality of tubes connected in series to each other, and first part 110 and second part 112 are each composed of a combination of tubes. More specifically, the position of boundary 114 between first part 110 and second part 112 is defined in a region where the refrigerant flow regime changes, and a plurality of tubes are combined such that a connecting part (boundary 114) that connects a group of tubes forming first part 110 and a group of tubes forming second part 112 to each other is included in a region where the refrigerant flow regime changes. In other words, a plurality of tubes are combined such that boundary 114 is located at a joint of tubes, rather than at a location on any single tube. Thus, it is unnecessary to prepare a tube in which the inner surface area of the tube changed within the tube, and accordingly the parts cost can be saved.

[0051] Fig. 10 shows one example of a specific configuration of indoor heat exchanger 50 shown in Fig. 1. Referring to Fig. 10, indoor heat exchanger 50 includes a plurality of tubes 124, 125, a plurality of connecting tubes 126, and a plurality of fins 128.

[0052] A plurality of tubes 124, 125 are arranged in parallel at regular intervals. A plurality of fins 128 are formed to surround each of a plurality of tubes 124, 125. A plurality of connecting tubes 126 each connect adjacent tubes 124 or 125 alternately on the left side and on the right side to thereby connect, in series, the plurality of tubes 124, 125 arranged in parallel.

[0053] Among a plurality of tubes 124, 125, a plurality of upstream tubes 124 correspond to first part 110 shown in Fig. 7, and a plurality of downstream tubes 125 correspond to second part 112 shown in Fig. 7. Specifically, the inner surface area of each tube 125 is smaller than the inner surface area of each tube 124. Connecting tube 126 that connects the most downstream tube 124 among the plurality of tubes 124 and the most upstream tube 125 among the plurality of tubes 125 to each other corresponds to boundary 114 shown in Fig. 7.

[0054] With such a configuration, it is unnecessary to prepare, as heat transfer tube 100 forming indoor heat exchanger 50, a tube in which the inner surface area changes within the tube, and thus first part 110 and second part 112 can be formed easily.

[0055] As seen from the above, Embodiment 1 enables suppression of deterioration in heat transfer capability of indoor heat exchanger 50 (low-pressure side heat exchanger) as well as suppression of increase of the pressure loss.

Embodiment 2

[0056] In indoor heat exchanger 50 (low-pressure side heat exchanger), as the surrounding environment such as outdoor air temperature changes, the region (position) where the refrigerant flow regime is changed to annular flow or annular mist flow varies. As the region where the refrigerant flow regime is changed varies, this region may become non-coincident with the position of boundary 114 between first part 110 and second part 112 of heat transfer tube 100, which may result in a problem that an oil film is formed disadvantageously on the tube wall, for example.

[0057] In Embodiment 2, therefore, controller 90 controls the operating state of air conditioner 1 such that the region (position) where the refrigerant flow regime is changed approaches boundary 114 between first part 110 and second part 112.

[0058] Fig. 11 is a block diagram showing one example of a hardware configuration of controller 90. Referring to Fig. 11, controller 90 includes a CPU (Central Processing Unit) 132, a RAM (Random Access Memory) 134, a ROM (Read Only Memory) 136, an input unit 138, a display unit 140, and an I/F unit 142. RAM 134, ROM 136, input unit 138, display unit 140, and I/F unit 142 are connected to CPU 132 through a bus 144.

[0059] CPU 132 deploys and executes, on RAM 134, programs stored in ROM 136. The programs stored in ROM 136 are each a program in which a process procedure for controller 90 is written. In this air conditioner 1, respective devices in air conditioner 1 are controlled in accordance with these programs. Control of these devices is not limited to processing by software, and may also be processing by dedicated hardware (electronic circuit).

[0060] Fig. 12 is a flowchart illustrating one example of a process performed by controller 90. In this flowchart, one example of a process procedure for control to make the region where the refrigerant flow regime is changed coincident with boundary 114. A series of steps shown in this flow chart are performed repeatedly in predetermined cycles during operation of air conditioner 1 (during operation of compressor 10).

[0061] Referring to Fig. 12, controller 90 detects the refrigerant flow regime at boundary 114, shown in Fig. 7, of heat transfer tube 100 in indoor heat exchanger 50 that is a low-pressure side heat exchanger (step S10).

[0062] The refrigerant flow regime at boundary 114 can be detected, for example, in the following way. From refrigerant temperature T1 on the inlet side of outdoor heat exchanger 30, temperature T2 of refrigerant on the outlet side of outdoor heat exchanger 30, temperature T3 (condensation temperature) of the heat transfer tube in outdoor heat exchanger 30, temperature T4 (outdoor air temperature) of the location where outdoor unit 2 (outdoor heat exchanger 30) is placed, temperature T5 (evaporation temperature) of heat transfer tube 100 in indoor heat exchanger 50, and temperature T6 (indoor temperature) of a target space to be air-conditioned where indoor unit 3 (indoor heat exchanger 50) is placed, which are detected respectively by temperature sensors 81 to 86, the refrigeration cycle of air conditioner 1 can be determined with a p-h diagram (pressure vs. specific enthalpy diagram). From this refrigeration cycle (p-h diagram), the dryness fraction of refrigerant at the position of boundary 114 in indoor heat exchanger 50 can be determined. Then, the determined dryness fraction can be applied to the Baker map shown in Fig. 6 to thereby detect (estimate) the refrigerant flow regime at boundary 114.

[0063] When the refrigerant flow regime at boundary 114 is detected in step S10, controller 90 determines whether the detected flow regime is annular flow or annular mist flow (step S20). When it is determined that the flow regime is annular flow or annular mist flow (YES in step S20), controller 90 increases the valve opening of pressure reducing device 40 (step S30). When the refrigerant flow regime at boundary 114 is annular flow or annular mist flow, the point at which the flow regime is changed in indoor heat exchanger 50 is located upstream of boundary 114. In this case, the valve opening of pressure reducing device 40 is increased to raise the evaporation pressure in indoor heat exchanger 50 and decrease the refrigerant dryness fraction at the inlet of indoor heat exchanger 50. As a result, the point at which the flow regime is changed in indoor heat exchanger 50 is shifted toward the downstream side and thus approaches boundary 114.

[0064] In contrast, when it is determined in step S20 that the flow regime is not annular flow or annular mist flow (NO in step S20), controller 90 decreases the valve opening of pressure reducing device 40 (step S40). When the refrigerant flow regime at boundary 114 is not annular flow or annular mist flow (i.e., the flow regime is slug flow or stratified flow, for example), the point at which the flow regime is changed in indoor heat exchanger 50 is located downstream of boundary 114. In this case, the valve opening of pressure reducing device 40 is decreased to lower the evaporation pressure in indoor heat exchanger 50 and increase the refrigerant dryness fraction at the inlet of indoor heat exchanger 50. As a result, the point at which the flow regime is changed in indoor heat exchanger 50 is shifted toward the upstream side and thus approaches boundary 114.

[0065] In step S20, it may be determined whether the refrigerant flow regime is slug flow or stratified flow. When it is determined that the flow regime is slug flow or stratified flow, the process may proceed to step S40 to decrease the valve opening of pressure reducing device 40 and, when it is determined that the flow regime is not slug flow or stratified flow, the process may proceed to step S30 to increase the valve opening of pressure reducing device 40.

[0066] As seen from the above, Embodiment 2 can suppress separation of the position of boundary 114 between first part 110 and second part 112 of heat transfer tube 100 in indoor heat exchanger 50 (low-pressure side heat exchanger), from the region (position) where the refrigerant flow regime is changed to annular flow or annular mist flow.

Modification 1 of Embodiment 2

[0067] In Embodiment 2 described above, the valve opening of pressure reducing device 40 is adjusted to cause the region (position) where the refrigerant flow regime is changed, to approach boundary 114. Alternatively, adjustment of the operating frequency of compressor 10 may be made, instead of the adjustment of pressure reducing device 40.

[0068] Fig. 13 is a flowchart illustrating one example of a process performed by controller 90 in this Modification 1. This flowchart corresponds to the flowchart shown in Fig. 12.

[0069] Referring to Fig. 13, operations performed in steps S 110, S 120 are identical to those performed in steps S10, S20 of Fig. 12, respectively.

[0070] When it is determined in step S120 that the refrigerant flow regime is annular flow or annular mist flow (YES in step S120), controller 90 decreases the operating frequency of compressor 10 (step S130). Accordingly, the flow rate of refrigerant flowing through the refrigerant circuit is decreased, so that the group of dots 95 in the Baker map shown in Fig. 6 is entirely shifted downward in the map. As a result, the point at which the flow regime is changed in indoor heat exchanger 50 is shifted toward the downstream side and thus approaches boundary 114.

[0071] In contrast, when it is determined in step S120 that the flow regime is not annular flow or annular mist flow (NO in step S120), controller 90 increases the operating frequency of compressor 10 (step S140). Accordingly, the flow rate of refrigerant flowing through the refrigerant circuit is increased and the group of dots 95 is entirely shifted upward in the Baker map shown in Fig. 6. As a result, the point at which the flow regime is changed in indoor heat exchanger 50 is shifted toward the upstream side and thus approaches boundary 114.

[0072] In this Modification 1 as well, it may be determined in step S120 whether the refrigerant flow regime is slug flow or stratified flow. When it is determined that the flow regime is slug flow or stratified flow, the process may proceed to step S 140 to increase the operating frequency of compressor 10 and, when it is determined that the flow regime is not slug flow or stratified flow, the process may proceed to step S130 to decrease the operating frequency of compressor 10.

Modification 2 of Embodiment 2

[0073] In order to cause the region (position) where the refrigerant flow regime is changed to approach boundary 114, Embodiment 2 adjusts the valve opening of pressure reducing device 40 and Modification 1 adjusts the operating frequency of compressor 10. Alternatively, the capability (rotational speed) of fan 52 of indoor heat exchanger 50 may be adjusted.

[0074] Fig. 14 is a flowchart illustrating one example of a process performed by controller 90 in this Modification 2. This flowchart also corresponds to the flowchart shown in Fig. 12.

[0075] Referring to Fig. 14, operations performed in steps S210, S220 are identical to those performed in steps S10, S20 of Fig. 12, respectively.

[0076] When it is determined in step S220 that the refrigerant flow regime is annular flow or annular mist flow (YES in step S220), controller 90 decreases the rotational speed of fan 52 of indoor heat exchanger 50 (step S230). As the rotational speed of fan 52 is decreased, the volume of air blown by fan 52 is decreased. The decrease of the volume of air blown by fan 52 has a similar effect as the decrease of the refrigerant flow rate, i.e., has a similar effect to the decrease of the operating frequency of compressor 10. Therefore, as the rotational speed of fan 52 is decreased, the point at which the flow regime is changed in indoor heat exchanger 50 is shifted toward the downstream side and thus approaches boundary 114.

[0077] In contrast, when it is determined in step S220 that the flow regime is not annular flow or annular mist flow (NO in step S220), controller 90 increases the rotational speed of fan 52 for indoor heat exchanger 50 (step S240). Accordingly, the point at which the flow regime is changed in indoor heat exchanger 50 is shifted toward the upstream side and thus approaches boundary 114.

[0078] In this Modification 2 as well, it may be determined in step S220 whether the refrigerant flow regime is slug flow or stratified flow. When it is determined that the flow regime is slug flow or stratified flow, the process may proceed to step S240 to increase the rotational speed of fan 52 and, when it is determined that the flow regime is not slug flow or stratified flow, the process may proceed to step S230 to decrease the rotational speed of fan 52.

Embodiment 3

[0079] In Embodiment 2 and its modifications, temperatures T1 to T6 detected respectively by temperature sensors 81 to 86 are used to detect the refrigerant flow regime at boundary 114 of heat transfer tube 100 in indoor heat exchanger 50. In this Embodiment 3, a sensor capable of detecting the refrigerant flow regime is placed at boundary 114 and the refrigerant flow regime at boundary 114 is detected directly.

[0080] Fig. 15 shows a placement of a sensor that detects the refrigerant flow regime in indoor heat exchanger 50 in Embodiment 3. Fig. 15 corresponds to Fig. 7 described above in connection with Embodiment 1.

[0081] Referring to Fig. 15, the configuration of heat transfer tube 100 is identical to the heat transfer tube shown in Fig. 7. In Embodiment 3, a luminosity sensor 150 is placed at boundary 114 between first part 110 and second part 112.

[0082] Luminosity sensor 150 is a sensor for detecting the flow regime of refrigerant flowing at boundary 114, based on the luminosity detected when light is applied to refrigerant (gas-liquid two-phase flow) flowing at boundary 114. The luminosity to be detected may be the luminosity of transmitted light or the luminosity of reflected light. Based on the fact that the luminosity to be detected varies depending on the refrigerant flow regime, the flow regime of refrigerant flowing at boundary 114 is detected based on the detected luminosity. The relation between the detected luminosity and the flow regime is evaluated in advance through an experiment or the like conducted in advance, and the relation between the detected value of luminosity sensor 150 and the flow regime of the refrigerant flowing at boundary 114 is stored in the form of a map or the like in ROM 136, to thereby enable the flow regime of the refrigerant flowing at boundary 114 to be determined easily from the detected value of luminosity sensor 150.

[0083] Then, based on the flow regime of the refrigerant flowing at boundary 114, which is detected by means of luminosity sensor 150, the point at which the refrigerant flow regime is changed is caused to approach boundary 114, in accordance with the flowchart shown in Fig. 12, 13, or 14.

[0084] As seen from the above, Embodiment 3 can also suppress separation of the position of boundary 114 from the region (position) where the refrigerant flow regime is changed, by detecting the flow regime of refrigerant flowing at boundary 114, using the detected value of luminosity sensor 150.

Modification 1 of Embodiment 3

[0085] An acoustic wave sensor 160 may be placed at boundary 114, instead of luminosity sensor 150, and the refrigerant flow regime at boundary 114 may be detected by acoustic wave sensor 160.

[0086] Referring again to Fig. 15, acoustic wave sensor 160 is also placed at boundary 114 between first part 110 and second part 112. Acoustic wave sensor 160 is a sensor for detecting the flow regime of refrigerant flowing at boundary 114, based on the wave detected when acoustic wave is applied toward refrigerant (gas-liquid two-phase flow) flowing at boundary 114. Based on the fact that the wave to be detected varies depending on the refrigerant flow regime, the flow regime of refrigerant flowing at boundary 114 is detected based on the acoustic wave. The relation between the detected wave and the flow regime is evaluated in advance through an experiment or the like conducted in advance, and the relation between the detected value of acoustic wave sensor 160 and the flow regime of the refrigerant flowing at boundary 114 is stored in the form of a map or the like in ROM 136, to thereby enable the flow regime of the refrigerant flowing at boundary 114 to be determined easily from the detected value of acoustic wave sensor 160.

[0087] In Embodiment 2 and its modifications, the flow regime of refrigerant at boundary 114 is detected and pressure reducing device 40 for example is controlled such that the region (position) where the flow regime is changed approaches boundary 114. Alternatively, the region (position) where the flow regime is changed in heat transfer tube 100 may be estimated and pressure reducing device 40 for example may be controlled such that the region approaches boundary 114. The region where the refrigerant flow regime is changed can be estimated, as described above, from the enthalpies of saturated liquid and saturated vapor at the temperature (evaporation temperature) of heat transfer tube 100 and the dryness fraction at which the flow regime changes. The refrigerant dryness fraction at which the flow regime changes can be calculated from the temperature (evaporation temperature) of heat transfer tube 100, the flow rate of refrigerant flowing through the refrigerant circuit, and the inner diameter of heat transfer tube 100, for example.

[0088] In connection with the above embodiments and their modifications, the air conditioner is described as an example of a refrigeration cycle apparatus. The refrigeration cycle apparatus according to the present disclosure, however, is not limited to the air conditioner and may also be applicable to a refrigeration cycle apparatus used for warehouse or showcase, for example.

[0089] It should be construed that the embodiments disclosed herein are given by way of illustration in all respects, not by way of limitation. It is intended that the technical scope specified in the present disclosure is defined by claims, not by the above description of the embodiments, and encompasses all modifications and variations equivalent in meaning and scope to the claims.

REFERENCE SIGNS LIST

[0090] 1 air conditioner; 2 outdoor unit; 3 indoor unit; 10 compressor; 20 four-way valve; 30 outdoor heat exchanger; 32, 52 fan; 40 pressure reducing device; 50 indoor heat exchanger; 62-72, 124, 125 tubes; 81-86 temperature sensor; 90 controller; 95 group of dots; 100 heat transfer tube; 102 liquid refrigerant; 104 gas refrigerant; 106 oil droplet; 108 oil film; 110 first part; 112 second part; 114 boundary; 116, 118 groove; 120 inlet; 122 outlet; 126 connecting tube; 128 fin; 132 CPU; 134 RAM; 136 ROM; 138 input unit; 140 display unit; 142 I/F unit; 144 bus; 150 luminosity sensor; 160 acoustic wave sensor

Claims

1. A refrigeration cycle apparatus in which incompatible oil is employed as refrigerating machine oil, the refrigeration cycle apparatus comprising:

a compressor configured to compress refrigerant;
a first heat exchanger configured to condense the refrigerant output from the compressor;
a pressure reducing device configured to reduce a pressure of the refrigerant output from the first heat exchanger; and
a second heat exchanger configured to evaporate the refrigerant output from the pressure reducing device to output the refrigerant to the compressor, wherein
the second heat exchanger includes a heat transfer tube having a groove formed on an inner surface of the heat transfer tube, and
the groove is formed such that an inner surface area per unit length of the heat transfer tube on a downstream side of the heat transfer tube is smaller than an inner surface area per unit length of the heat transfer tube on an upstream side of the heat transfer tube.

2. The refrigeration cycle apparatus according to claim 1, wherein

the heat transfer tube include:

a first part; and
a second part located downstream of the first part,

the groove is formed such that the inner surface area in the second part is smaller than the inner surface area in the first part, and
a boundary between the first part and the second part is located in a region where flow regime of refrigerant flowing through the heat transfer tube changes.

3. The refrigeration cycle apparatus according to claim 2, wherein

the first part includes at least one first tube connected in series,
the second part includes at least one second tube connected in series, and
the at least one first tube and the at least one second tube are formed such that a connecting part that connects the at least one first tube and the at least one second tube to each other is included in the region.

4. The refrigeration cycle apparatus according to claim 2 or 3, further comprising:

a temperature sensor configured to detect a state of the refrigerant; and
a detector configured to detect the flow regime of the refrigerant at the boundary.

5. The refrigeration cycle apparatus according to claim 4, wherein the temperature sensor detects

an inlet-side refrigerant temperature of the first heat exchanger, an outlet-side refrigerant temperature of the first heat exchanger, a heat transfer tube temperature of the first heat exchanger, and an ambient temperature of the first heat ex-

changer, and
an ambient temperature of the second heat exchanger and a heat transfer tube temperature of the second heat exchanger.

6. The refrigeration cycle apparatus according to claim 2 or 3, further comprising a sensor placed at the boundary and configured to detect the flow regime of the refrigerant at the boundary.

7. The refrigeration cycle apparatus according to claim 6, wherein the sensor includes a luminosity sensor configured to detect a state of the refrigerant at the boundary.

8. The refrigeration cycle apparatus according to claim 6, wherein the sensor includes an acoustic wave sensor configured to detect a state of the refrigerant at the boundary.

9. The refrigeration cycle apparatus according to any one of claims 4 to 8, wherein

the pressure reducing device includes a pressure reducing valve,
the refrigeration cycle apparatus further comprises a controller configured to control an opening of the pressure reducing valve, and
the controller is configured to

increase the opening when the detected flow regime is annular flow or annular mist flow, and
decrease the opening when the detected flow regime is not annular flow or annular mist flow.

10. The refrigeration cycle apparatus according to any one of claims 4 to 8, further comprising a controller configured to control an operating frequency of the compressor, wherein
the controller is configured to

decrease the operating frequency when the detected flow regime is annular flow or annular mist flow, and
increase the operating frequency when the detected flow regime is not annular or annular mist flow.

11. The refrigeration cycle apparatus according to any one of claims 4 to 8, further comprising:

a fan provided for the second heat exchanger; and
a controller configured to control a rotational speed of the fan, wherein
the controller is configured to

decrease the rotational speed when the detected flow regime is annular flow or annular mist flow, and
increase the rotational speed when the detected flow regime is not annular flow or annular mist flow.

12. An air conditioner comprising the refrigeration cycle apparatus according to any one of claims 1 to 11.

13. A heat exchanger for a refrigeration cycle apparatus in which incompatible oil is used as refrigerating machine oil, wherein

the heat exchanger comprises a heat transfer tube having a groove formed on an inner surface of the heat transfer tube,
the heat transfer tube includes:

a first part; and
a second part located downstream of the first part,

the groove is formed such that an inner surface area per unit length of the second part is smaller than an inner surface area per unit length of the first part, and
a boundary between the first part and the second part is located in a region where flow regime of refrigerant flowing through the heat transfer tube changes.

14. The heat exchanger according to claim 13, wherein

the first part includes at least one first tube connected in series,
the second part includes at least one second tube connected in series, and
the at least one first tube and the at least one second tube are formed such that a connecting part that connects the at least one first tube and the at least one second tube to each other is included in the region.

FIG.1

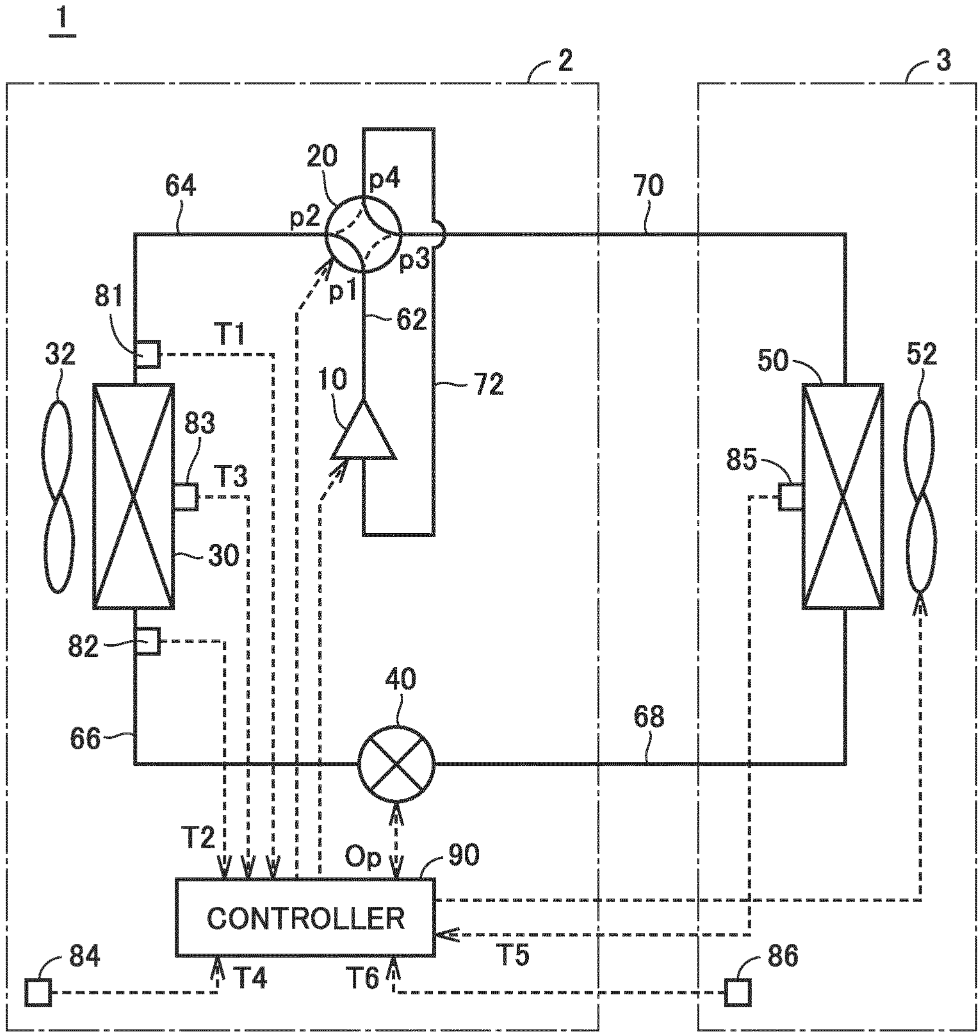

1
2
3
20
p4
64
70
p2
p3
p1
81
62
T1
32
50
52
10
72
83
T3
85
30
82
40
66
68
T2
Op
90
CONTROLLER
T5
84
86
T4
T6

FIG.2

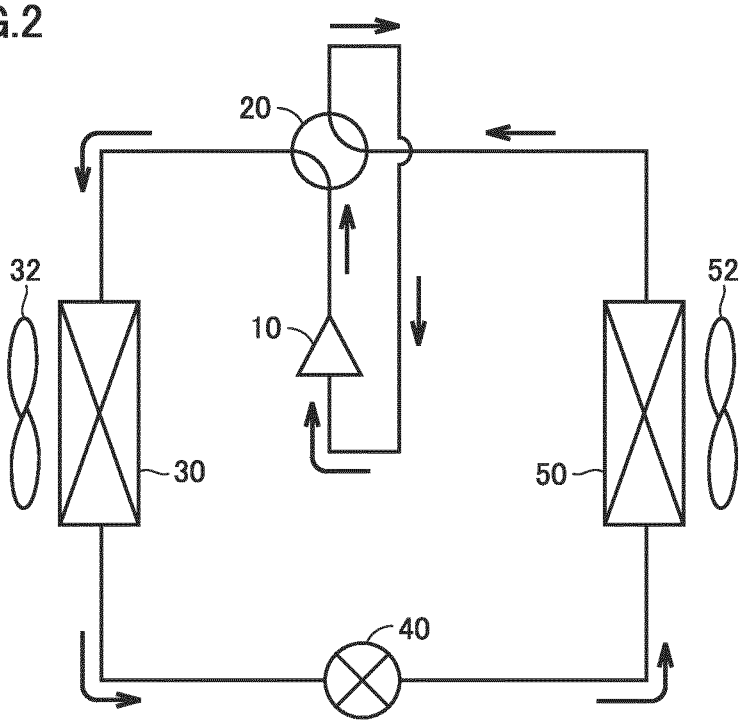


FIG.3

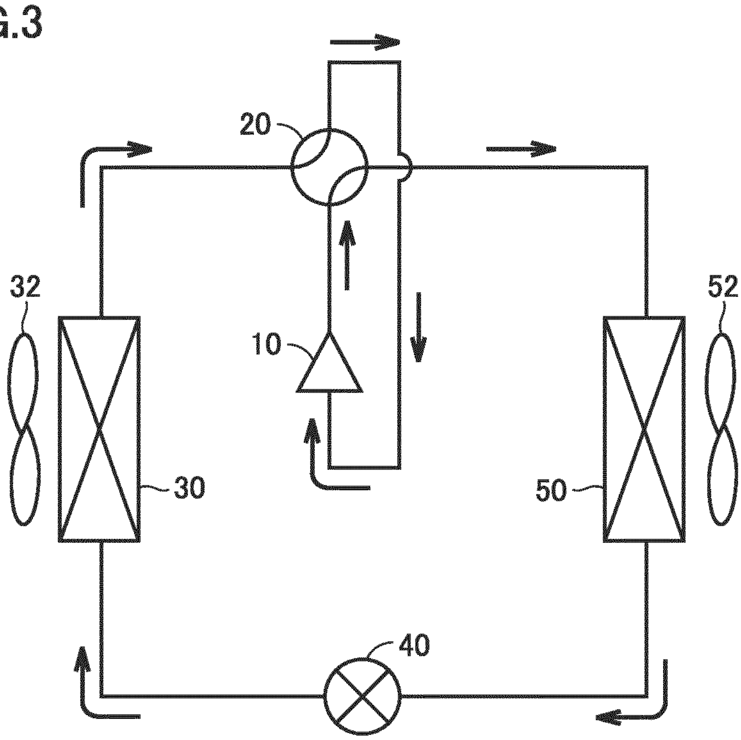


FIG.4

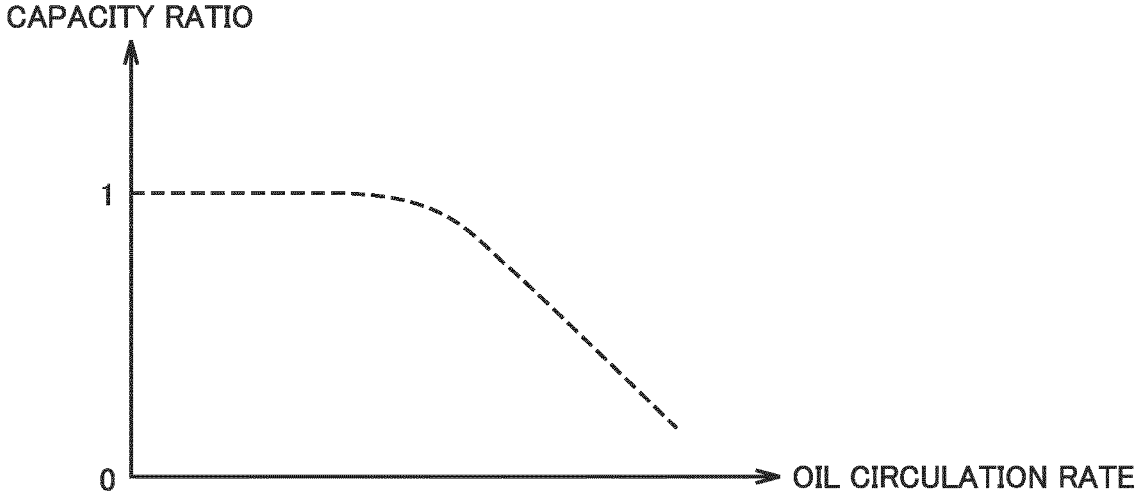

CAPACITY RATIO
1
0
OIL CIRCULATION RATE

FIG.5

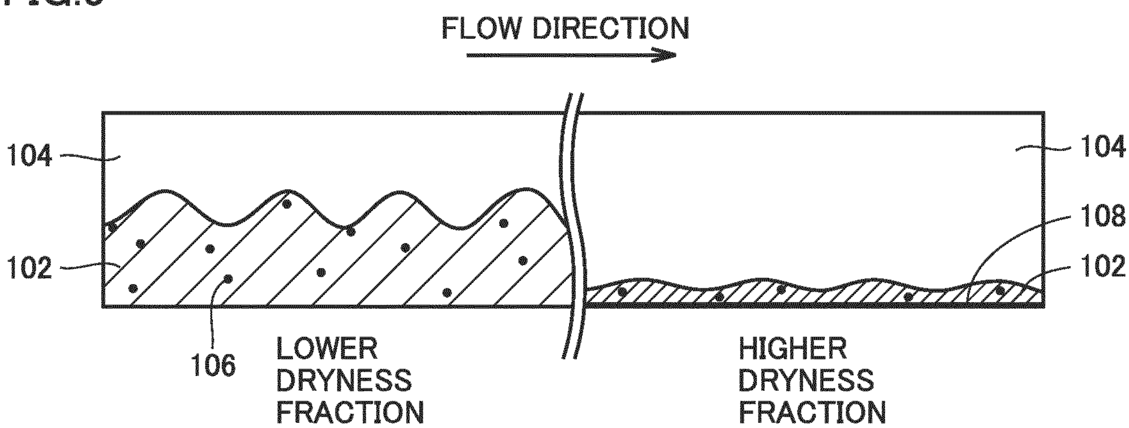

FLOW DIRECTION
104
102
106
LOWER
DRYNESS
FRACTION
104
108
102
HIGHER
DRYNESS
FRACTION

FIG.6

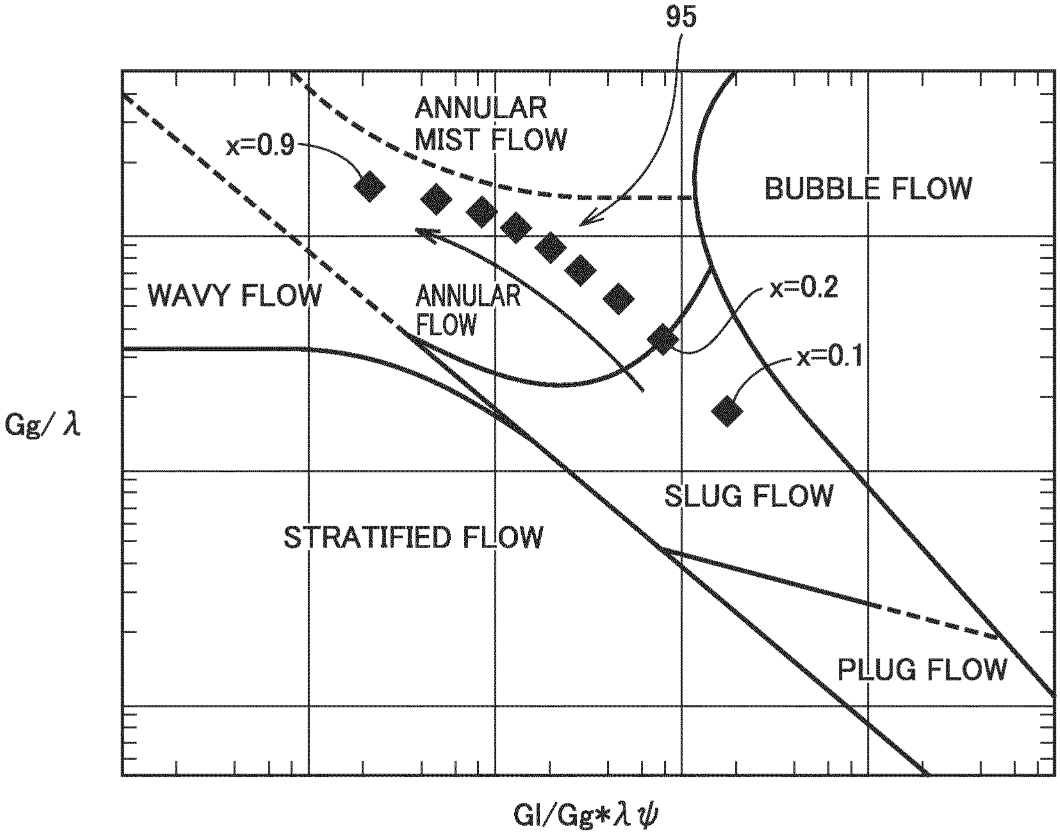

95
ANNULAR
MIST FLOW
x=0.9
BUBBLE FLOW
WAVY FLOW
ANNULAR
FLOW
x=0.2
x=0.1
Gg/ λ
SLUG FLOW
STRATIFIED FLOW
PLUG FLOW
Gl/Gg*λψ

FIG.7

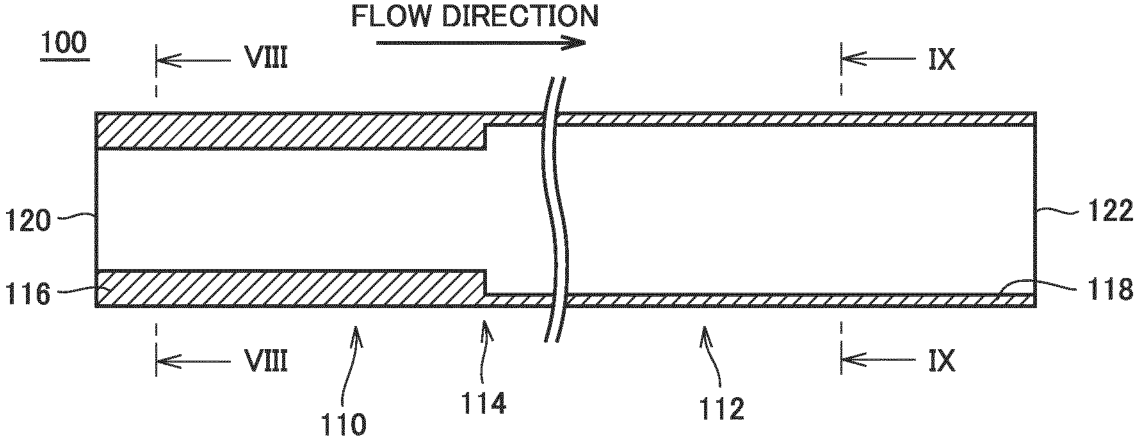

FLOW DIRECTION
100
VIII
IX
120
122
116
118
VIII
IX
110
114
112

FIG.8

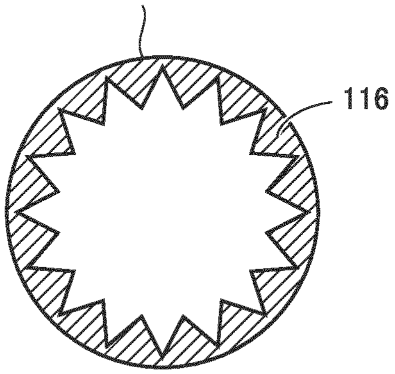


FIG.9

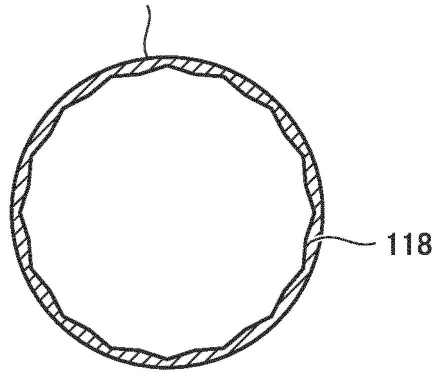


FIG.10

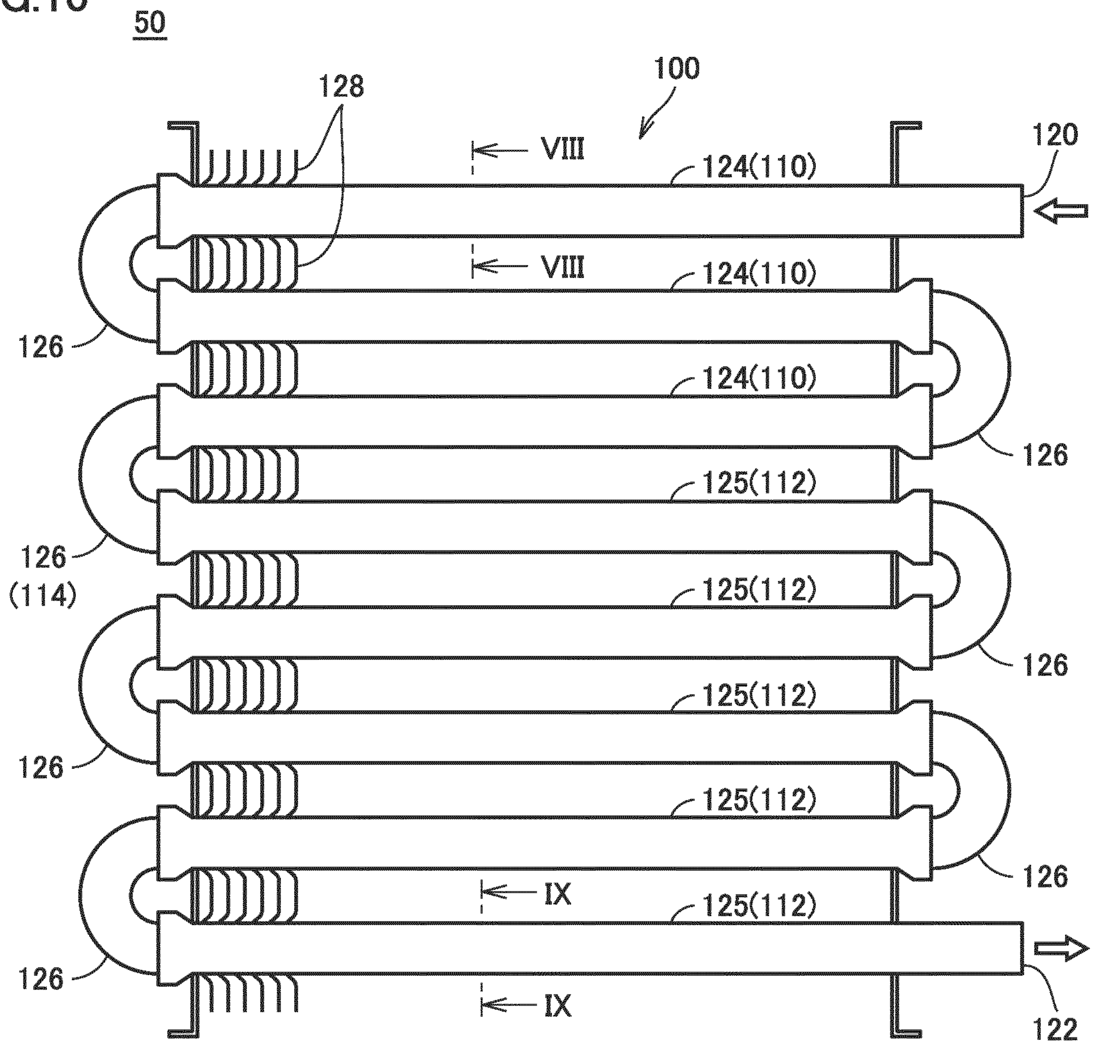

50
100
128
VIII
VIII
120
124(110)
124(110)
124(110)
125(112)
125(112)
125(112)
125(112)
125(112)
126
126
(114)
126
126
126
126
126
126
IX
IX
122

FIG.11

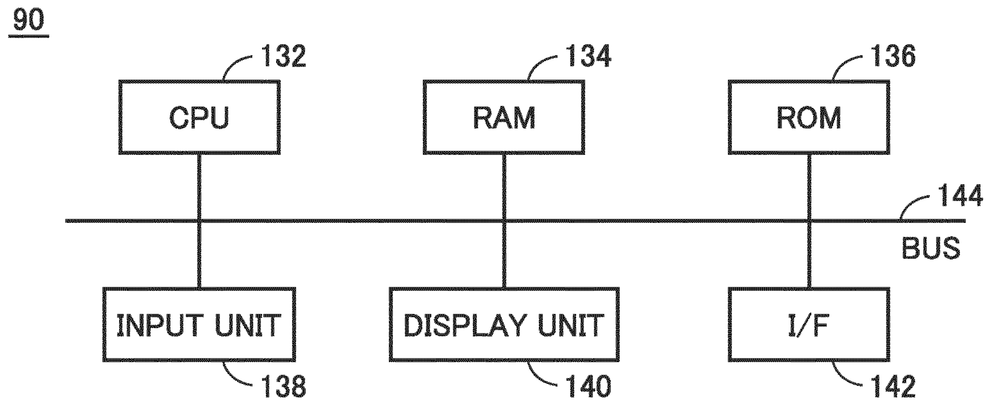

90
132
CPU
134
RAM
136
ROM
144
BUS
INPUT UNIT
DISPLAY UNIT
I/F
138
140
142

FIG.12

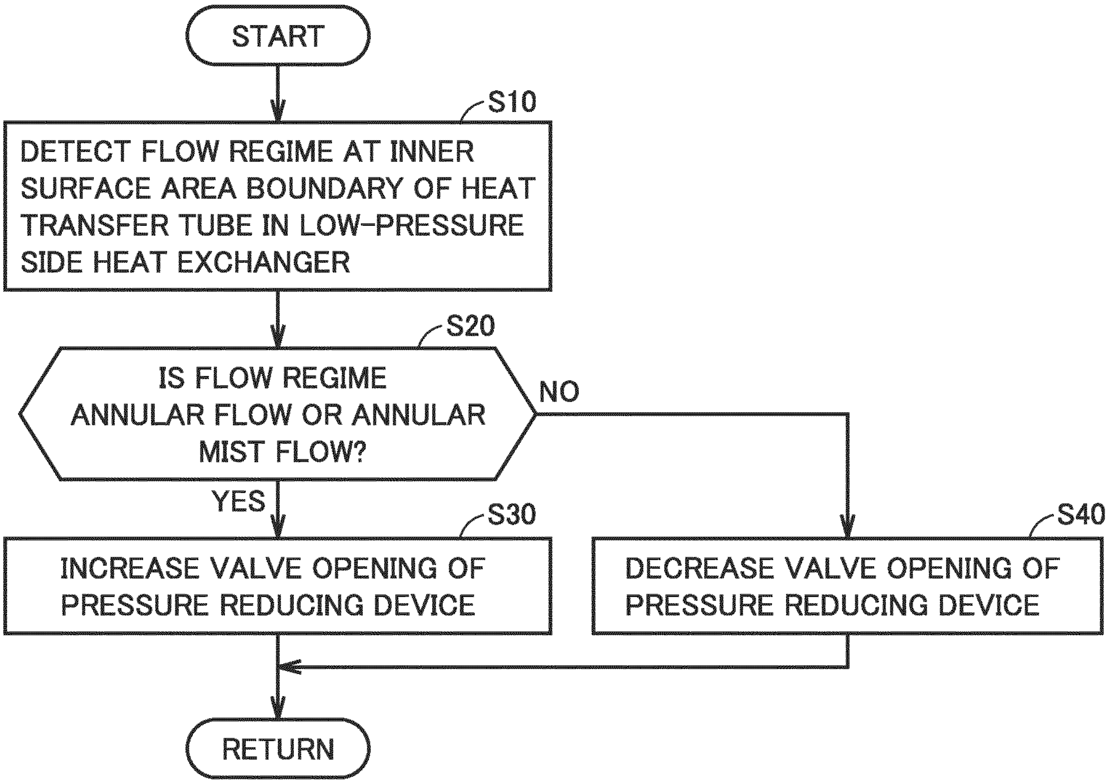

START
S10
DETECT FLOW REGIME AT INNER SURFACE AREA BOUNDARY OF HEAT TRANSFER TUBE IN LOW-PRESSURE SIDE HEAT EXCHANGER
S20
IS FLOW REGIME ANNULAR FLOW OR ANNULAR MIST FLOW?
NO
YES
S30
INCREASE VALVE OPENING OF PRESSURE REDUCING DEVICE
S40
DECREASE VALVE OPENING OF PRESSURE REDUCING DEVICE
RETURN

FIG.13

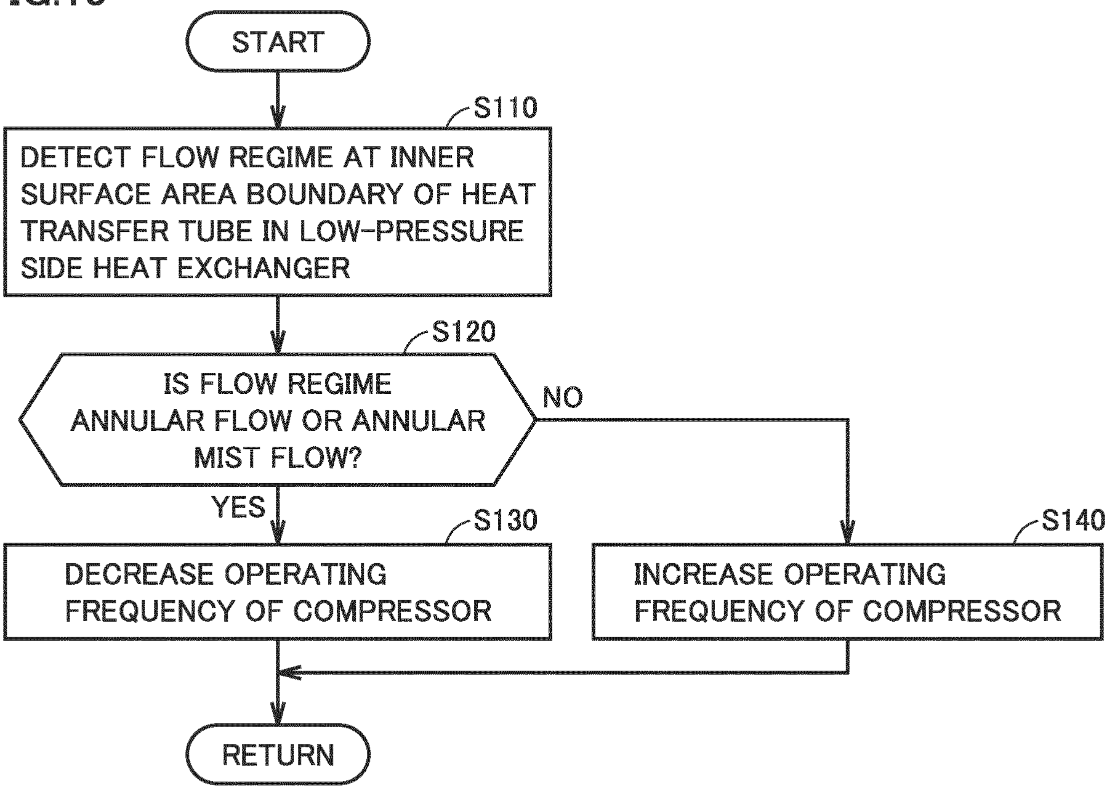

START
S110
DETECT FLOW REGIME AT INNER SURFACE AREA BOUNDARY OF HEAT TRANSFER TUBE IN LOW-PRESSURE SIDE HEAT EXCHANGER
S120
IS FLOW REGIME ANNULAR FLOW OR ANNULAR MIST FLOW?
NO
YES
S130
DECREASE OPERATING FREQUENCY OF COMPRESSOR
S140
INCREASE OPERATING FREQUENCY OF COMPRESSOR
RETURN

FIG.14

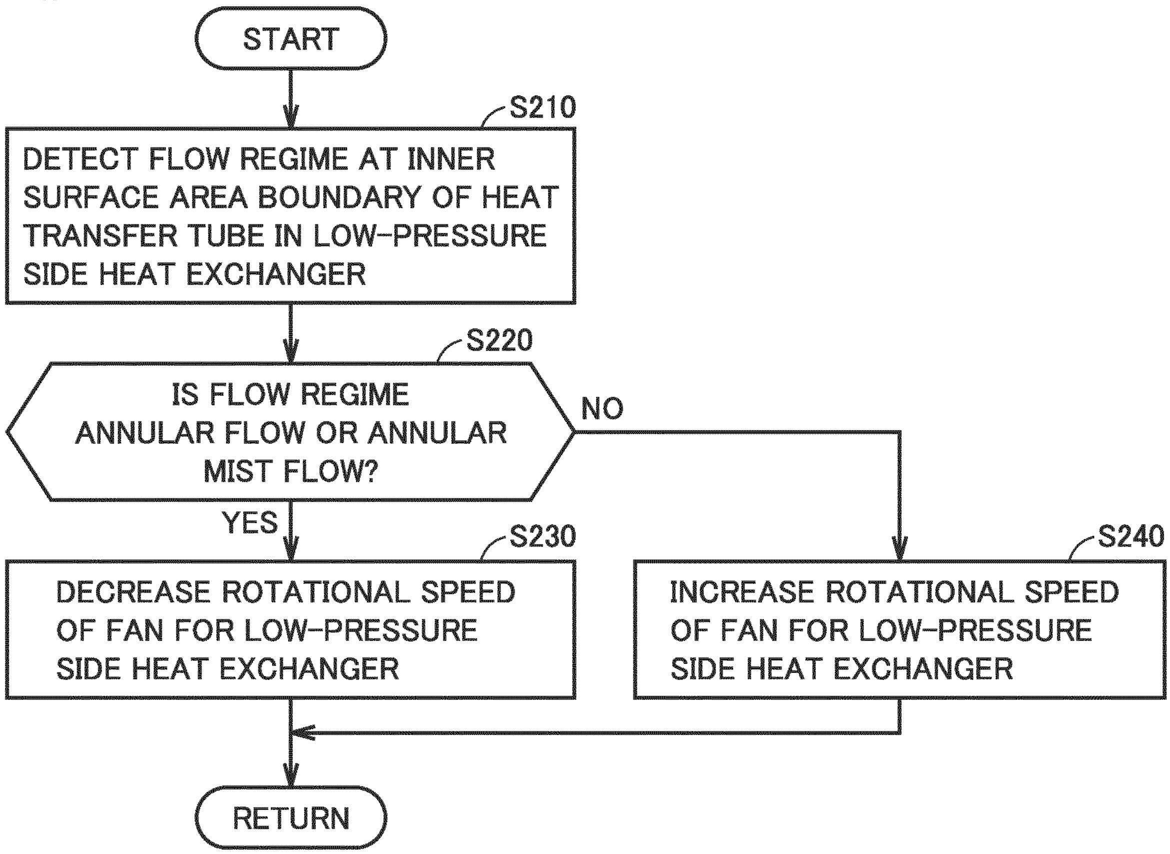
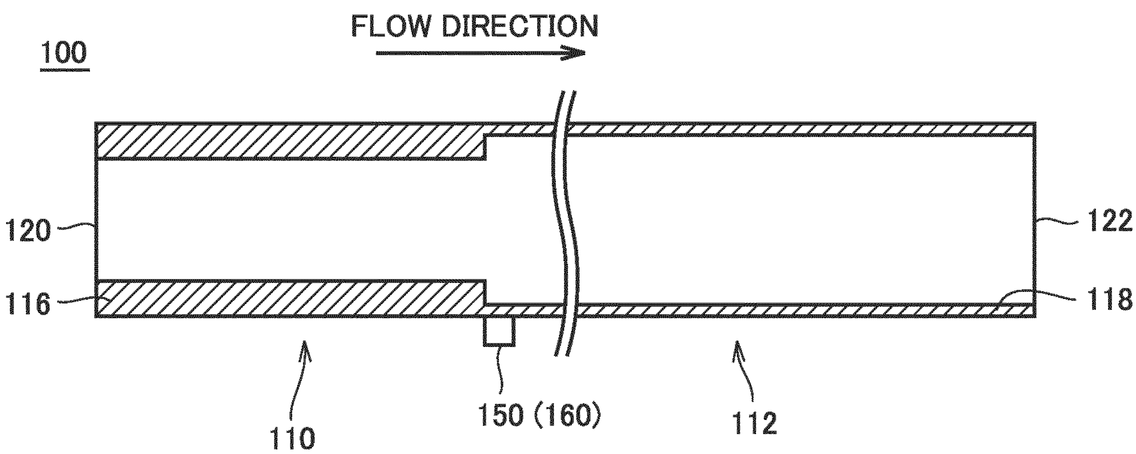

START
S210
DETECT FLOW REGIME AT INNER SURFACE AREA BOUNDARY OF HEAT TRANSFER TUBE IN LOW-PRESSURE SIDE HEAT EXCHANGER
S220
IS FLOW REGIME ANNULAR FLOW OR ANNULAR MIST FLOW?
NO
YES
S230
DECREASE ROTATIONAL SPEED OF FAN FOR LOW-PRESSURE SIDE HEAT EXCHANGER
S240
INCREASE ROTATIONAL SPEED OF FAN FOR LOW-PRESSURE SIDE HEAT EXCHANGER
RETURN

FIG.15


FLOW DIRECTION
100
120
116
122
118
110
150 (160)
112

INTERNATIONAL SEARCH REPORT

International application No. PCT/JP2020/038481

A. CLASSIFICATION OF SUBJECT MATTER
Int. Cl. F25B39/02(2006.01)i, F28F1/40(2006.01)i
FI: F25B39/02 W, F28F1/40 B

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

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
Int. Cl. F25B39/02, F28F1/40

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched
Published examined utility model applications of Japan 1922-1996
Published unexamined utility model applications of Japan 1971-2020
Registered utility model specifications of Japan 1996-2020
Published registered utility model applications of Japan 1994-2020

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	Microfilm of the specification and drawings annexed to the request of Japanese Utility Model Application No. 070563/1982 (Laid-open No. 174668/1983) (MITSUBISHI HEAVY INDUSTRIES, LTD.) 22 November 1983 (1983-11-22), description, p. 1, line 12, to p. 7, line 18, fig. 1-9	1-14
Y	JP 09-031450 A (DAIKIN INDUSTRIES, LTD.) 04 February 1997 (1997-02-04), paragraph [0004]	1-14
Y	JP 2001-272117 A (MITSUBISHI ELECTRIC CORP.) 05 October 2001 (2001-10-05), paragraph [0041]	1-14

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

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

Date of the actual completion of the international search	Date of mailing of the international search report
26.11.2020	08.12.2020

Name and mailing address of the ISA/	Authorized officer
Japan Patent Office 3-4-3, Kasumigaseki, Chiyoda-ku, Tokyo 100-8915, Japan	Telephone No.

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

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2020/038481

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	JP 2007-248030 A (MATSUSHITA ELECTRIC INDUSTRIAL CO., LTD.) 27 September 2007 (2007-09-27), paragraph [0029]	1-14
Y	Microfilm of the specification and drawings annexed to the request of Japanese Utility Model Application No. 145988/1979 (Laid-open No. 066690/1981) (MITSUBISHI ELECTRIC CORP.) 03 June 1981 (1981-06-03), description, p. 1, line 12, to p. 4, line 11, fig. 1-2	3, 14
Y	WO 2018/173256 A1 (MITSUBISHI ELECTRIC CORP.) 27 September 2018 (2018-09-27), paragraphs [0188-0189], fig. 57	4-5
Y	JP 2008-151380 A (MITSUBISHI ELECTRIC CORP.) 03 July 2008 (2008-07-03), paragraphs [0005], [0095]	4-11
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