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(71) Applicant: **Mitsubishi Electric Corporation**
Tokyo 100-8310 (JP)

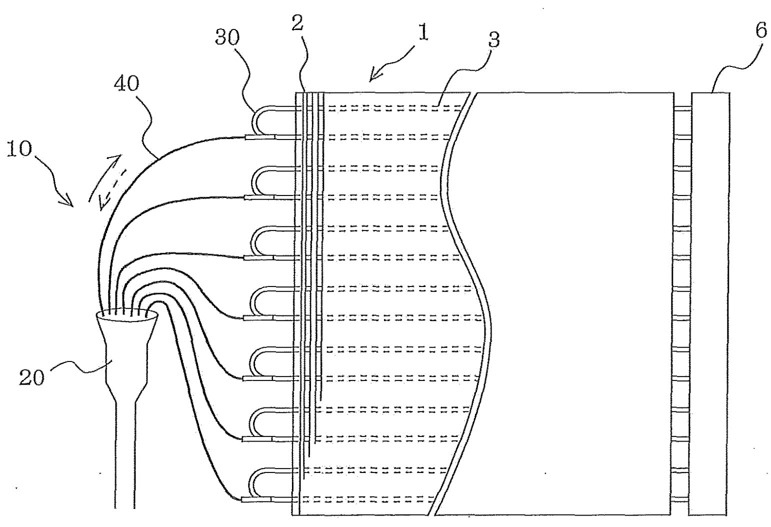
(72) Inventors:
- **MATSUDA, Takuya**
  Tokyo 100-8310 (JP)
- **HOKAZONO, Keisuke**
  Tokyo 100-8310 (JP)
- **OKAZAWA, Hiroki**
  Tokyo 100-8310 (JP)
- **SUZUKI, Wataru**
  Tokyo 100-8310 (JP)
- **NAKAJIMA, Takashi**
  Tokyo 100-8310 (JP)
- **MOCHIZUKI, Atsushi**
  Tokyo 102-0073 (JP)

(74) Representative: **Pfenning, Meinig & Partner GbR**
Patent- und Rechtsanwälte
Theresienhöhe 11a
80339 München (DE)

(54) REFRIGERANT DISTRIBUTOR AND HEAT PUMP DEVICE USING REFRIGERANT DISTRIBUTOR

(57) A refrigerant distribution device 10 is provided for distributing refrigerant to a plurality of heat transfer tubes 3 that constitute a heat exchanger 1. The refrigerant distribution device 10 includes a first distribution device 20 dividing refrigerant into a plurality of portions, and a plurality of two-way branch pipes 30 each dividing refrigerant divided by the first distribution device 20 into two portions to flow into two of the plurality of heat transfer tubes 3.

F I G. 1



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Description

Technical Field

[0001] The present invention relates to refrigerant distribution devices.

Background Art

[0002] When a plurality of refrigerant flow paths are provided in a heat exchanger, which serves as a condensor or an evaporator of a heat pump apparatus such as an air conditioner and a refrigerator, a refrigerant distribution device that distributes the refrigerant to the respective paths is necessary on an inlet side of the refrigerant.

[0003] A distributor is conventionally used as a refrigerant distribution device so that refrigerant distributed by the distributor flows into the respective heat transfer tubes of the heat exchanger by capillary tubes (for example, see Patent Literature 1).

Citation List

Patent Literature

[0004] Patent Literature 1: Japanese Unexamined Patent Application Publication No. 2008-121984 (page 4, Fig. 1)

Summary of Invention

Technical Problem

[0005] As a technique of improving heat exchange efficiency in a heat exchanger, using a circular tube having a small diameter or a flat tube in which a plurality of refrigerant flow paths are formed as a heat transfer tube is known. As the diameter of the circular tube decreases or the flat tube is used, the number of heat transfer tubes used in one heat exchanger increases.

[0006] In Patent Literature 1, a plurality of heat transfer tubes are connected to a distributor via capillary tubes that are individually disposed at the ends of the heat transfer tubes, which serve as inlet ports for refrigerant flowing from the outside, and the number of paths of the heat exchanger and the number of the capillary tubes are the same. As a result, as the number of heat transfer tubes increases and thus the number of paths increases, the number of capillary tubes also increases.

[0007] When the number of capillary tubes increases, there are problems that the capillary tubes are not easily routed and an installation space of the capillary tubes mounted on an actual machine increases. In addition, when the number of capillary tubes increases, the cost also increases. Accordingly, it is required to prevent the cost increase due to use of heat transfer tubes having small diameter or flat heat transfer tubes.

[0008] The present invention is made to overcome such problems and provides a refrigerant distribution device that enables a reduction in the number of connected capillary tubes, a compact installation space of the capillary tubes mounted on an actual machine, and a cost reduction, and a heat pump apparatus that uses this refrigerant distribution device.

Solution to Problem

[0009] A refrigerant distribution device according to the present invention distributes refrigerant to a plurality of heat transfer tubes that constitute a heat exchanger. The refrigerant distribution device includes a first distribution device dividing refrigerant into a plurality of portions, and a plurality of second distribution devices each dividing refrigerant divided by the first distribution device into two portions to flow into two of the plurality of heat transfer tubes. Advantageous Effects of Invention

[0010] According to the present invention, the number of connected capillary tubes can be reduced, an installation space of the capillary tubes mounted on an actual machine can be compact, and the cost of the capillary tubes can be reduced.

Brief Description of Drawings

[0011]

[Fig. 1] Fig. 1 is a schematic configuration diagram showing a state in which a refrigerant distribution device according to Embodiment 1 of the present invention is connected to a heat exchanger.

[Fig. 2] Fig. 2 is perspective views of a two-way branch pipe of Fig. 1.

[Fig. 3] Fig. 3 is a perspective view of a circular tube - flat tube joint.

[Fig. 4] Fig. 4 is configuration explanatory views of a two-way branch pipe of the refrigerant distribution device according to Embodiment 1 of the present invention.

[Fig. 5] Fig. 5 is a view showing a refrigerant circuit of a heat pump apparatus that uses the refrigerant distribution device according to Embodiment 1 of the present invention.

[Fig. 6] Fig. 6 is a view showing a configuration example in which the refrigerant distribution device according to Embodiment 1 of the present invention is connected to a heat exchanger for an outdoor unit of an air conditioning apparatus that uses a flat tube as a heat transfer tube.

[Fig. 7] Fig. 7 is an enlarged perspective view seen from the backside of a connecting portion of the heat exchanger and the refrigerant distribution device of Fig. 5.

[Fig. 8] Fig. 8 is a perspective view of the flat tube of Fig. 6.

[Fig. 9] Fig. 9 is a view showing a conventional example that does not use a two-way branch pipe in a heat exchanger of three-line arrangement configuration.

[Fig. 10] Fig. 10 is a configuration diagram of the refrigerant distribution device according to Embodiment 2 of the present invention.

[Fig. 11] Fig. 11 is views showing a third distribution device in the refrigerant distribution device of Fig. 10.

[Fig. 12] Fig. 12 is configuration diagrams of the second distribution device in the refrigerant distribution device according to Embodiment 3 of the present invention.

[Fig. 13] Fig. 13 is configuration diagrams of the second distribution device in the refrigerant distribution device according to Embodiment 4 of the present invention.

[Fig. 14] Fig. 14 is an enlarged view of an essential part of Fig. 13(A).

[Fig. 15] Fig. 15 is a configuration example in which a header is used for a first distribution device.

Description of Embodiments

Embodiment 1

[0012] Fig. 1 is a schematic configuration diagram showing a state in which a refrigerant distribution device according to Embodiment 1 of the present invention is connected to a heat exchanger. In Fig. 1 and the drawings described later, the same reference signs refer to the same or corresponding elements throughout the whole description. Further, the forms of the elements shown in the description are for exemplary purposes only and the invention is not limited to the description.

[0013] A heat exchanger 1 is a fin and tube heat exchanger that includes a plurality of plate shaped fins 2 being stacked with spaces therebetween and a plurality of heat transfer tubes 3 that penetrate the plate shaped fins 2 in a stack direction and allow refrigerant to flow therein. The heat transfer tubes 3 may be circular tubes or flat tubes made of copper or aluminum. A plurality of heat transfer tubes 3 are connected to a refrigerant distribution device 10 at one end and a gas header 6 at the other end.

[0014] Next, the refrigerant distribution device 10 connected to this heat exchanger 1 will be described.

[0015] The refrigerant distribution device 10 includes a first distribution device 20 and a plurality of two-way branch pipes 30 as a second distribution device. The first distribution device 20 is to uniformly distribute refrigerant that flows into the first distribution device 20 in a state of gas-liquid two-phase into the respective heat transfer tubes 3 of the heat exchanger 1. Accordingly, in Fig. 1, a distributor is used as the first distribution device 20.

[0016] A throttle mechanism such as an orifice is inserted in the distributor such that the two-phase flow that flows therein becomes a spray flow while passing through the orifice to facilitate uniform distribution. The spray flow of refrigerant is uniformly distributed to the respective capillary tubes 40. A throttle mechanism such as an orifice may not be used in the distributor. It is essential that a distributor capable of uniform distribution is used as the first distribution device 20. The distributor may be made of a material such as copper, aluminum and brass.

[0017] The capillary tube 40 has an inner diameter of approximately 3.5 mm and a length of approximately 1000 mm. These dimensions of the capillary tube 40 are merely an example. The capillary tube 40 may be curved in a circular shape depending on the relationship between the length and the installation space.

[0018] Further, an in-pipe pressure loss can be adjusted based on the specification (inner diameter, length) of the capillary tubes 40, thus a branch flow rate from the first distribution device 20 to each of the two-way branch pipes 30 can be adjusted. An air speed of an air-sending fan (not shown in the figure) that sends air to the heat exchanger 1 is not necessarily uniform across the entire surface of the heat exchanger 1, and an air speed distribution may be present. For example, when an air-sending fan is disposed in the upper part of the heat exchanger 1, the air speed is faster in the upper part of the heat exchanger 1 than in the lower part.

[0019] When the heat exchanger 1 is used as an evaporator, the refrigerant passing through a part in which the air speed is fast tends to be easily evaporated and dried compared with the refrigerant passing through a part in which the air speed is slow. Accordingly, when the same amount of refrigerant flows into the respective heat transfer tubes 3, the refrigerant passing through a part in which the air speed is fast has a quality higher than the refrigerant passing through a part in which the air speed is slow. This causes variation in refrigerant state at the exit of the heat exchanger, and the refrigerant state fails to be stabilized. Therefore, it is required that a larger amount of refrigerant flows into the heat transfer tube located in a part in which the air speed is fast. As described above, when adjustment of branch flow rate is necessary, the branch flow rate can be adjusted by adjusting the specification of the capillary tubes 40.

[0020] The two-way branch pipes 30 are connected to the end of the capillary tubes 40 on the opposite side to the first distribution device 20. Using the two-way branch pipes 30 allows the refrigerant to be distributed to the number of branches (the number of paths) of 2 x A, where A is the number of branches of the first distribution device 20 and the number of capillary tubes 40. A configuration of the two-way branch pipe 30 will be described below.

[0021] Fig. 2 is a perspective view of the two-way branch pipe of Fig. 1.

[0022] The two-way branch pipe 30 includes a U-bend section 31 formed by bending a circular tube in U-shape and a straight inflow section 35. The U-bend section 31 includes a connection 32 and two arms 33 and 34 extending from each end of the connection 32 in parallel to each other. The straight inflow section 35 is formed by performing bulge-forming process on one of the two arms

33 and 34 (in this example, the arm 34) of the U-bend section 31. In this example, the straight inflow section 35 is disposed on the arm 34. Bulge-forming process is a method for forming a hollow shape, in which a pipe-shaped material is set in a mold of a press and then clamped, and after that, while the material is filled with a liquid of high pressure, the material is compressed in the axial direction so that the both ends are forced toward each other, thereby allowing the material to be expanded into the shape of the mold cavity.

[0023] The straight inflow section 35 of the two-way branch pipe 30 constituted as described above is connected to one end of an L-shaped pipe 36, which is bent in L-shape, while the other end of the L-shaped pipe 36 is connected to the capillary tube 40. The opening ports A and B of the two arms 33 and 34 of the two-way branch pipe 30 are connected to the heat transfer tubes 3 of the heat exchanger 1. In connecting the two arms 33 and 34 and the heat transfer tubes 3, the heat transfer tubes 3 are directly connected when they are circular tubes, and the heat transfer tubes 3 are connected via circular tube - flat tube joints 4 of Fig. 3 when they are flat tubes. Further, the two-way branch pipe 30 is connected to the heat exchanger 1 so that the two arms 33 and 34 extend in a parallel direction and the refrigerant flowing from the straight inflow section 35 to the arm 34 is branched and flows in the horizontal direction. The reason will be described later.

[0024] In the right figure of Fig. 2, the dotted line arrow indicates the flow of refrigerant in the case where the heat exchanger 1 is used as an evaporator, and in the two-way branch pipe 30, the refrigerant flowing from the capillary tube 40 flows into the straight inflow section 35 via the L-shaped pipe 36. The refrigerant flowing into the straight inflow section 35 is branched into the arm 33 and the arm 34, each of which flows into the respective heat transfer tubes 3.

[0025] As described above, as the two-way branch pipe 30 allows the introduced refrigerant to be branched into two flows to the respective heat transfer tubes 3, the number of capillary tubes 40 can be reduced to half compared with the configuration in which the capillary tubes 40 are directly connected to the respective heat transfer tubes 3. Accordingly, using the refrigerant distribution device 10 of Embodiment 1 can improve installability of the capillary tubes 40.

[0026] In addition to the improvement in instability of the capillary tubes 40, the two-way branch pipe 30 is required to have a function of uniformly distributing the refrigerant flowing from the capillary tubes 40 and allowing the refrigerant to flow into the respective heat transfer tubes 3. In the refrigerant distribution device 10, the first distribution device 20 can uniformly distribute the refrigerant into the respective capillary tubes 40, and it is also required for the two-way branch pipes 30 to uniformly distribute the refrigerant so that the refrigerant flows into the respective heat transfer tubes of the heat exchanger 1 while remaining in the uniformly distributed state.

[0027] The refrigerant that has passed an expansion valve of the heat pump apparatus and the refrigerant at an inlet of the evaporator is generally in a state of gas-liquid two-phase flow made up of gas refrigerant and liquid refrigerant, causing density distribution in a cross section of the refrigerant flowing in pipes. For example, when a pipe has a curve, a biased flow occurs due to an effect of centrifugal force, which causes the flowing liquid refrigerant to be biased to one side of the inner surface of the pipe. That is, two-phase refrigerant is separated into gas and liquid phases.

[0028] Therefore, when the heat exchanger is used as an evaporator, it is required for the refrigerant distribution device located on the inflow side of the refrigerant to have a function of preventing the above biased flow from being generated and gas and liquid phases from being separated. Further, it is required for the refrigerant distribution device to have a function of distributing the refrigerant in a state of being uniformly mixed while the ratio of the gas and liquid mass flow rate at the inlet of the refrigerant distribution device remains to be equal to the ratio of the gas and liquid mass flow rate at the outlet of the refrigerant distribution device. Moreover, as described above, as the number of paths increases by decreasing the diameter of the circular tube or flattening the tube, distributing uniform flow rate of refrigerant to the respective paths becomes a more important matter.

[0029] A configuration used in the two-way branch pipes 30 for uniformly distributing the refrigerant will be described below.

[0030] Fig. 4 is configuration explanatory views of the two-way branch pipe of the refrigerant distribution device according to Embodiment 1 of the present invention. Fig. 4(a) is a front view of the two-way branch pipe 30 and Fig. 4(b) is a side view of Fig. 4(a).

[0031] In the two-way branch pipe 30, the straight inflow section 35 is formed by bulge-forming such that an angle θ_1 between a tube axis X1 of the straight inflow section 35 and a tube axis X2 of the arm 34 is 90 degrees.

[0032] When the angle θ_1 deviates from 90 degrees by 10 degrees or more, the refrigerant flowing from the straight inflow section 35 obliquely collides against a portion of the arm 34 that faces the straight inflow section 35 and generates a biased flow, leading to deterioration of heat exchange efficiency. Therefore, the angle θ_1 is formed to be 90 degrees. It should be noted the angle is not limited to be exact 90 degrees, and may be slightly deviated.

[0033] It is further effective for uniform distribution that the straight inflow section 35 has the length of 5 mm or more. This point will be described below. The two-phase flow of refrigerant becomes a state in which the liquid phase of the refrigerant is biased to one side in the capillary tube 40. Providing an approach section that extends until a position at which the refrigerant collides against the two-way branch and has 20 times or more of the inner diameter allows for a sufficiently stable flow. However, in Embodiment 1, ensuring a configuration with the approach section having 20 times or more of the inner diameter is difficult. Accordingly, Embodiment 1 ensures the length of 5 mm or more for the straight inflow section 35 and the length of 15 mm or more (two times or more of the inner diameter) together with the length of the horizontal section of the capillary tube 40 or the L-shaped pipe 36.

[0034] An experiment revealed that ensuring these dimensions allows a liquid film of the two-phase flow of refrigerant flowing into the straight inflow section 35 to be uniformly distributed in the pipe so that a stable annular flow is provided. This annular flow vertically collides against a collision wall 34a that faces the straight inflow section 35 and then can be uniformly distributed regardless of the amount of circulation because the branching directions after collision are horizontal, being free from the effect of gravity.

[0035] When the straight inflow section 35 is less than 5 mm, the gas-liquid two-phase flow is affected by a curve of the capillary tube 40 and becomes a state in which the liquid surface is biased in the pipe of the straight inflow section 35. In this case, the gas-liquid two-phase flow into the straight inflow section 35 is biased by an effect of the biased liquid surface and fails to be uniformly distributed in the two-way branch pipe 30, thereby compromising heat exchange efficiency.

[0036] As described above, the two-way branch pipe 30 preferably has the angle θ_1 of 90 degrees between the arm 34 and the straight inflow section 35 and the length of the straight inflow section 35 of 5 mm or more.

[0037] As described above, the straight inflow section 35 of the two-way branch pipe 30 is formed by bulge-forming. Using the bulge-forming process allows for a stable processing of the shape of 90 degrees for the angle θ_1 and ensures the length of 5 mm or more for the straight inflow section 35. Bulge-forming process can be applied to not only a straight pipe but also a curved pipe, and further, the straight inflow section 35 can be formed to have a long length while preventing the wall thickness from being decreased.

[0038] The two-way branch pipe 30 and the capillary tube 40 can be connected by inserting the L-shaped pipe 36 or the capillary tube 40 having an outer diameter smaller than an inner diameter of an inlet port of the bulge-formed straight inflow section 35 in the inlet port of the bulge-formed straight inflow section 35 and brazing thereto. This provides stable manufacture without defective. On the other hand, when the straight inflow section 35 is formed without using bulge-forming, the straight inflow section 35 has a small length and a small wall thickness. Consequently, when the L-shaped pipe 36 or the capillary tube 40 is connected to the straight inflow section 35, the angle θ_1 deviates from 90 degrees by a large amount. Further, in brazing connection, a part of the straight inflow section 35 is to be brazed, and stable brazing cannot be performed due to a small wall thickness and small length of the connection.

[0039] Although an angle θ_2 between the straight inflow section 35 and the tube axis X2 of the arm 34 may be any degrees, the angle θ_2 herein is 90 degrees as shown in Fig. 4. The reason will be described below. As the diameter of the heat transfer tube 3 decreases and the heat transfer tube 3 is flattened, flow resistance of air decreases. Accordingly, the heat transfer tubes 3 are designed to have narrow arrangement pitch, and the mounting density of the heat transfer tubes 3 is increased. As described above, using the two-way branch pipe 30 can improve instability of the capillary tubes 40. In addition to that, providing the angle θ_2 of the straight inflow section 35 of the two-way branch pipe 30 of 90 degrees (shown in Fig. 4) can further improve connectivity of the capillary tubes 40.

[0040] To curve a pipe without making deformation and creases, a curvature R that is twice to three times of the inner diameter is necessary, and in the case where the angle θ_2 is 0 degree, the L-shaped pipe 36 or the capillary tube 40 interferes with the two-way branch pipe 30. Accordingly, in the case where the angle θ_2 is 0 degree, when the mounting density of the heat transfer tubes 3 is increased, it is difficult to connect the capillary tube 40 or the L-shaped pipe 36 to the straight inflow section 35. With the angle θ_2 of 90 degrees, the connection with the capillary tube 40 or the L-shaped pipe 36 comes to the direction of air flow, which does not interfere with the two-way branch pipe 30, thereby improving connectivity.

[0041] The advantage of the L-shaped pipe 36 will be described below. The L-shaped pipe 36 and the straight inflow section 35 of the two-way branch pipe 30 are initially connected by brazing, then each of the opening ports A and B are connected to the circular tube - flat tube joints 4 by brazing, and then the L-shaped pipe 36 and the capillary tube 40 are connected by brazing. This procedure allows for a stable manufacturing process having a low defective rate. The reason will be as described below. That is, the above connection by brazing is performed by a burner, and since the L-shaped pipe 36 and the capillary tube are connected by brazing at last, other pipes are less likely to be exposed to the flame of burner.

[0042] Further, by using the L-shaped pipe 36, it becomes easy to provide the angle θ_1, at which refrigerant flowing into the two-way branch pipe, of 90 degrees and provide a long approach distance for flowing into the two-way branch pipe, thereby improving uniform distribution. In the case where the L-shaped pipe 36 is not used, the capillary tube 40 is directly connected to the straight inflow section 35 of the two-way branch pipe 30. Consequently, as the capillary tube 40 is long, unstable, and non-easily routed, the angles θ_1, at which refrigerant flowing into the two-way branch pipe 30, tends to largely vary.

[0043] In the above description, the straight inflow section 35 of the two-way branch pipe 30 is 5 mm or more. However, since it is essential to provide a straight flow path of 10 mm or more, it is also possible that a sum of the L-shaped pipe 36 and the straight inflow section 35 is 10 mm or more.

[0044] Fig. 5 is a view showing a refrigerant circuit of a heat pump apparatus that uses the refrigerant distribution device according to Embodiment 1 of the present invention.

[0045] The heat pump apparatus 60 includes a compressor 61, a condensor 62 (heat exchanger 1), an expansion valve 63 as a depressurizing device and an evaporator 64 (heat exchanger 1). The gas refrigerant ejected from the compressor 61 flows into the condensor 62, exchanges heat with air passing through the condensor 62, becomes a high pressure liquid refrigerant and flows out from the condensor 62. The high pressure liquid refrigerant out of the condensor 62 is depressurized by the expansion valve 63, becomes a low pressure gas-liquid two-phase refrigerant, and flows into the evaporator 64. The low pressure gas-liquid two-phase refrigerant flowing into the evaporator 64 exchanges heat with air passing through the evaporator 64, becomes a low pressure gas refrigerant, and is suctioned back into the compressor 61.

[0046] Referring to Figs. 1 and 5, an operation in the case where the heat exchanger 1 is used as an evaporator will be described. In Fig. 1, the solid arrow indicates a flow of refrigerant in the case where the heat exchanger 1 is used as an evaporator.

[0047] First, the gas-liquid two-phase refrigerant flow out of the expansion valve 63 flows into the first distribution device 20 and becomes a spray flow. The spray flow of the refrigerant is uniformly distributed to flow into the respective capillary tubes 40. After passing through the respective capillary tubes 40, the refrigerant flows into the two-way branch pipes 30 and flows out while being uniformly distributed into two branched flows as described above, each of which flows into the heat transfer tube 3. The refrigerant flowing into the respective heat transfer tubes 3 exchanges heat with air and becomes a gas state, and is then collected in the gas header 6. The refrigerant collected in the gas header 6 is suctioned into the compressor 61.

[0048] Referring to Figs. 3 and 5, an operation in the case where the heat exchanger 1 is used as a condensor will be described. In Fig. 1, the dotted arrow indicates a flow of refrigerant in the case where the heat exchanger 1 is used as a condensor.

[0049] In the case of condensor, the refrigerant flows in the direction opposite to the case of the evaporator, and the gas refrigerant flow out of the compressor 61 flows into the gas header 6. The refrigerant flowing into the gas header 6 flows into the respective heat transfer tubes 3 while being uniformly distributed in the gas header 6. A distributor or the like is not necessary since the refrigerant can be uniformly distributed without difficulty when the refrigerant is in a gas state, and the gas header 6 constituted by a cylindrical hollow tube is used.

[0050] After exchanging heat with air, the refrigerant flowing into the respective heat transfer tubes 3 flows through the two-way branch pipes 30, the capillary tubes 40 and the first distribution device 20 in sequence. Then, the refrigerant is collected in the first distribution device 20, and flows into the expansion valve 63.

[0051] Next, a specific example of pipe connection using the refrigerant distribution device 10 according to Embodiment 1 will be described.

[0052] Fig. 6 is a view showing a configuration example in which the refrigerant distribution device according to Embodiment 1 of the present invention is connected to a heat exchanger for an outdoor unit of an air conditioning apparatus that uses a flat tube as a heat transfer tube. Fig. 7 is an enlarged perspective view seen from the backside of a connecting portion of the heat exchanger and the refrigerant distribution device of Fig. 5. In Fig. 7, shaded portions are the two-way branch pipes 30. Fig. 8 is a perspective view of the flat tube of Fig. 6.

[0053] The heat exchanger 100 has a configuration in which heat exchangers are arranged in three lines of an air flow direction, and includes a plurality of plate shaped fins 2 being stacked with spaces therebetween and a plurality of flat tube 3 that penetrate the plate shaped fins 2 in a stack direction and allow refrigerant to flow therein. The ends of a plurality of flat tubes 3 are connected to bends 5, which are bent in a hairpin shape, and the gas header 6 in addition to the two-way branch pipes 30. Further, in this heat exchanger 100, since the flat tubes 3 are used as heat transfer tubes, the flat tubes 3 are connected to the two-way branch pipes 30 and the bends 5 via the circular tube - flat tube joints 4.

[0054] Further, when the heat exchanger 100 is used as an evaporator, a path configuration in the heat exchanger 100 is constituted so that the refrigerant flows in a so-called parallel flow in which refrigerant flows while turning back from upstream to downstream with respect to an air flow direction. On the other hand, when the heat exchanger 100 is used as a condensor, paths are constituted so that the refrigerant flows in a so-called opposite flow in which refrigerant flows while turning back from downstream to upstream with respect to an air flow direction. Since the refrigerant is sub-cooled in the condensor, the refrigerant temperature decreases. Accordingly, the heat exchange efficiency in the case where the heat exchanger 100 is used as a condensor can be improved by constituting paths so that the refrigerant flows as an opposite flow.

[0055] The plate shaped fin 2 is made of aluminum, and the flat tube 3 and the plate shaped fin 2 are connected by in-furnace brazing. In the case where the heat exchanger uses the heat transfer tube of a circular tube, the heat transfer tube and the plate shaped fin are connected by a mechanical pipe expansion method, which causes a problem that heat exchange efficiency is compromised because an air layer is present between the heat transfer tube and the plate shaped fin. However, since the flat tube 3 and the plate shaped fin 2 are connected by in-furnace brazing, heat resistance between the flat tube 3 and the plate shaped fin 2 becomes zero, thereby improving heat exchange efficiency.

[0056] The flat tube 3 is made of aluminum and the inside of the flat tube 3 is separated to form a plurality of passages 31 a as shown in Fig. 8 so that the refrigerant flows through the respective passages 31 a. The flat tube 3 can increase the amount of refrigerant and a contact heat transfer area three times of those of the circular tube.

[0057] Next, the case where the refrigerant distribution device 10 according to Embodiment 1 is used in the heat exchanger of three-line arrangement is compared with the case where the refrigerant distribution device 10 according to Embodiment 1 is not used with reference to Fig. 7 and Fig. 9, respectively. Fig. 9 is a view showing a conventional example that does not use a two-way branch pipe in a heat exchanger of three-line arrangement configuration.

[0058] As seen from the comparison of Fig. 7 and Fig. 9, using the two-way branch pipe 30 can decrease the number of the capillary tubes 40 to half of that in the conventional example.

[0059] According to Embodiment 1, as described above, the number of capillary tubes 40 can be reduced by combining the capillary tube 40, which is an outflow pipe of the first distribution device 20, with the two-way branch pipe 30 in the refrigerant distribution device 10. As a result, when the capillary tube 40 is mounted on an actual machine, installation space of the capillary tube 40 becomes compact. Further, cost can be reduced by reducing the number of the capillary tubes 40, and the actual machine can be inexpensively configured.

[0060] Since the angle θ_1 between the arm 34 and the straight inflow section 35 is 90 degrees, the refrigerant two-phase flow vertically collides against the collision wall 34a, which faces the inflow section. Further, branch directions after collision are horizontal, and the refrigerant is not under the effect of gravity and can be uniformly distributed regardless of the amount of circulation. Since the straight inflow section 35 is formed by bulge-forming process, stable processing of the shape of 90 degrees can be performed for the angle θ_1.

[0061] Further, since the straight inflow section 35 is 10 mm or more, the refrigerant two-phase flow into the straight inflow section 35 can be allowed to vertically collide against the collision wall 34a as a stable annular flow, thereby preventing variation in refrigerant distribution and allowing for uniform distribution.

[0062] As described above, using the refrigerant distribution device 10 allows for uniform distribution of the two-phase flow of gas-liquid. Accordingly, specific heat exchange efficiency can be sufficiently performed in the heat exchanger 1.

Embodiment 2

[0063] Embodiment 2 is provided for further reducing the number of the capillary tubes 40. The following description focuses on a part of Embodiment 2 that differs from Embodiment 1. Further, a modification example that is applied to the same configuration as that of Embodiment 1 is also applied to Embodiment 2.

[0064] Fig. 10 is a configuration diagram of the refrigerant distribution device according to Embodiment 2 of the present invention. In Fig. 10, (a) is a front view, and (b) is a cross sectional view taken along the line A-A of (a). Fig. 11 is views showing a third distribution device in the refrigerant distribution device of Fig. 10. In Fig. 11, (a) is a front view of the refrigerant distribution device 10A, (b) is a side view of (a), and (c) is a bottom view of (a).

[0065] The refrigerant distribution device 10A of Embodiment 2 has a configuration that includes a plurality of two-way branch pipes 50 as a third distribution device in addition to the first distribution device 20 and the two-way branch pipes 30 as a second distribution device of Embodiment 1. The two-way branch pipe 50 has the same structural characteristics as those of the two-way branch pipe 30 of Embodiment 1, and includes a U-bend section 51, which is formed by bending the circular tube in U-shape, and the straight inflow section 55. The U-bend section 51 includes a connection 52 and two arms 53 and 54 extending from each end of the connection 52 in parallel to each other. The straight inflow section 55 is formed by performing bulge-forming process on one of the two arms 53 and 54 (in this example, the arm 54) of the U-bend section 51.

[0066] The straight inflow section 55 of the two-way branch pipe 50 is connected to the capillary tube 40, and the two arms 53 and 54 are each connected to the straight inflow section 35 of the adjacent two-way branch pipe 30. Further, the two-way branch pipe 50 is also disposed so that the two arms 53 and 54 extend in a parallel direction as similar to the two-way branch pipe 30 of Embodiment 1.

[0067] The refrigerant distribution device 10A of Embodiment 2, which is configured as described above, can obtain the same effect as that of Embodiment 1 and the following effect. That is, since the two-way branch pipes 50 are disposed as a third distribution device, the number of capillary tubes 40 connected to the first distribution device 20 can be reduced to half of that of

Embodiment 1.

[0068] Since the two-way branch pipe 50 serves as an inflow pipe for the refrigerant flowing out from the capillary tube 40, the same function as that of the two-way branch pipe 30 of Embodiment 1 is required. That is, the function of uniformly distributing the refrigerant out of the capillary tube 40 and allowing it to flow into the respective heat transfer tubes 3 is required. Since the two-way branch pipe 50 has the same configuration as that of the two-way branch pipe 30, the refrigerant out of the straight inflow section 35 of the two-way branch pipe 50 vertically collides against a collision wall that is a wall surface facing the straight inflow section 55 in the arm 54. Since branch directions after collision are horizontal, the refrigerant is not under the effect of gravity and can be uniformly distributed regardless of the amount of circulation.

Embodiment 3

[0069] Embodiment 3 is directed to a refrigerant distribution device that can adjust a distribution ratio of refrigerant. The following description focuses on a part of Embodiment 3 that differs from Embodiment 1. Further, in Embodiment 3, a modification example that is applied to the same configuration as that of Embodiment 1 is also applied to Embodiment 3.

[0070] Fig. 12 is configuration diagrams of the second distribution device in the refrigerant distribution device according to Embodiment 3 of the present invention. In Fig. 12, (a) is a front view of the second distribution device, and (b) is a side view of (a).

[0071] The refrigerant distribution device of Embodiment 3 has the same configuration as that of Embodiment 1 except that a two-way branch pipe 30A is disposed instead of the two-way branch pipe 30.

[0072] The two-way branch pipe 30 of Embodiment 1 forms the angle θ_1 of 90 degrees with respect to the straight inflow section 35 and the arm 34. On the other hand, the two-way branch pipe 30A of Embodiment 3 has a configuration in which the angle θ_1 is adjusted depending on a desired distribution ratio. When the angle θ_1 is formed as the angle smaller than 90 degrees, the flow rate of the refrigerant flowing in the opening port A becomes larger than that of the opening port B, while when the angle θ_1 is formed as the angle larger than 90 degrees, the flow rate of the refrigerant flowing in the opening port B becomes larger than that of the opening port A. Accordingly, the straight inflow section 35 that requires angle adjustment is formed by bulge-forming process as similar to Embodiment 1. As a result of forming by bulge-forming process, variation of the angles θ_1 can be reduced and the angle θ_1 can be precisely formed at a predetermined angle. Although the angle θ_2 may be any angle as similar to Embodiment 1, 90 degrees is preferable in consideration of connection to the capillary tubes 40.

[0073] The two-way branch pipe 30A is connected to the first distribution device 20 at one end and the other end of the capillary tube 40 at the other end.

[0074] As described above, an air speed of an air sending fan that sends air to the heat exchanger 1 is not necessarily uniform across the entire surface of the heat exchanger 1, and air speed distribution is present. Such air speed distribution or difference in the lengths of the refrigerant flow path for each of thermal paths may cause difference in thermal load of the paths. Accordingly, it is necessary that a larger amount of refrigerant flows into one of the paths connected to the opening port A and the opening port B that is, for example, having a faster air speed or a larger number of heat transfer tubes and a longer length of flow path. Therefore, the distribution ratio of refrigerant to the opening port A and the opening port B is determined based on the thermal load distribution of the paths in the heat exchanger 1, and thus the angle θ_1 is determined.

[0075] By determining the angle θ_1 as described above, the refrigerant flowing from the straight inflow section 35 collides against a surface facing the straight inflow section 35 with the angle θ_1 and is then distributed into the arm 33 and the arm 34 while the flow rate is adjusted depending on an inclination angle θ. Then, the refrigerant of distributed flow rate flows into the heat transfer tubes 3 of the heat exchanger 1 via the opening port A and the opening port B.

[0076] As described above, according to Embodiment 3, the number of capillary tubes 40 can be reduced as similar to Embodiment 1, and a refrigerant distribution device that can adjust a distribution ratio of refrigerant can be provided.

Embodiment 4

[0077] Embodiment 4 is also directed to a refrigerant distribution device that can adjust a distribution ratio of refrigerant as similar to Embodiment 3. The following description focuses on a part of Embodiment 4 that differs from Embodiment 1. Further, a modification example that is applied to the same configuration as that of Embodiment 1 in Embodiment 3 is also applied to Embodiment 4.

[0078] Fig. 13 is configuration diagrams of the second distribution device in the refrigerant distribution device according to Embodiment 4 of the present invention. In Fig. 13, (A) shows a configuration example of the two-way branch pipe 30B in which the distribution ratio at the opening port B is larger than that at the opening port A, and (B) shows a configuration example of the two-way branch pipe 30B in which the distribution ratio at the opening port A is larger than that at the opening port B. Further, in Figs. 13(A) and (B), (a) is a front view of the two-way branch pipe 30B, and (b) is a side view of (a). Fig. 14 is an enlarged view of an essential part of Fig. 13(A).

[0079] The refrigerant distribution device of Embodiment 4 has the same configuration as that of Embodiment 1 except that the two-way branch pipe 30B is disposed instead of the two-way branch pipe 30. The two-way branch pipe 30B has an inwardly depressed recess 37 at a position adjacent to the collision wall 34a. The remaining is the same as Embodiment 1 including the facts that the straight inflow section 35 is formed by bulge-forming process and that the angle θ_2 can be any angle.

[0080] When the thermal load at the opening port B is higher, that is, when the refrigerant flow rate should be larger at the opening port B than at the opening port A, the recess 37 is formed at a position close to the opening port A with respect to the collision wall 34a in a tube axis direction of the U-bend section 31 as shown in Fig. 13(A). On the other hand, when the thermal load at the opening port A is higher, that is, when refrigerant flow rate should be larger at the opening port A than at the opening port B, the recess 37 is formed at a position close to the opening port B with respect to the collision wall 34a in a tube axis direction of the U-bend section 31 as shown in Fig. 13(B).

[0081] Since the recess 37 is formed adjacent to the collision wall 34a, a cross sectional area of the refrigerant becomes small and the refrigerant after colliding against the collision wall 34a becomes difficult to flow. Accordingly, in the case of Fig. 13(A), the distribution ratio at the opening port B becomes larger than that at the opening port A, and in the case of Fig. 13(B), the distribution ratio at the opening port A becomes larger than that at the opening port B.

[0082] Further, the recess 37 is not formed on the collision wall 34a, which faces the straight inflow section 35. Since the recess 37 is formed by using a press or punch, forming the recess 37 on the collision wall 34a may deform the straight inflow section 35. Accordingly, the two-way branch pipe 30B can be manufactured in a stable manner while preventing disadvantage of deformation by not providing the recess 37 on the collision wall 34a, which faces the straight inflow section 35.

[0083] As described above, according to Embodiment 4, the number of capillary tubes 40 can be reduced as similar to Embodiment 1, and a refrigerant distribution device that can adjust a distribution ratio of refrigerant can be provided.

[0084] Although the first distribution device 20 is described as a distributor in the above Embodiments 1 to 4, the invention is not limited thereto, and the first distribution device 20 may be a header 70 as shown in Fig. 15. In that case, the same effect can be obtained.

[0085] The heat exchanger described in the above Embodiments and an air conditioning apparatus that uses the heat exchanger can achieve the above effects with any refrigerating machine oil such as mineral oil, alkylbenzene oil, ester oil, ether oil, and fluorine oil regardless of whether the refrigerant dissolves in the oil or not.

[0086] Although Embodiments 1 to 4 are described as separate Embodiments, combination of Embodiments can be used as appropriate when Embodiments can be combined. For example, the two-way branch pipe 30 of Embodiment 2 shown in Fig. 10 can be replaced with the two-way branch pipe 30A of Embodiment 3 by combining Embodiment 2 and Embodiment 3.

Industrial Applicability

[0087] As an application example, the present invention can be used for a heat exchanger of a heat pump apparatus that is required to improve heat exchange efficiency and performance.

Reference Signs List

[0088] 1 heat exchanger 2 plate shaped fin 3 heat transfer tube (flat tube) 4 circular tube-flat tube joint 5 bend 6 gas header 10 refrigerant distribution device 10A refrigerant distribution device 20 first distribution device 30 two-way branch pipe (second distribution device) 30A two-way branch pipe (second distribution device) 30B two-way branch pipe (second distribution device) 31 U-bend section 31 a passage 32 connection 33 arm 34 arm 34a collision wall 35 straight inflow section 36 L-shaped pipe 37 recess 40 capillary tube 50 two-way branch pipe (third distribution device) 51 U-bend section 52 connection 53 arm 54 arm 55 straight inflow section 60 heat pump apparatus 61 compressor 62 condensor 63 expansion valve 64 evaporator 70 header 100 heat exchanger

Claims

1. A refrigerant distribution device for distributing refrigerant to a plurality of heat transfer tubes that constitute a heat exchanger, the refrigerant distribution device comprising:

   a first distribution device dividing refrigerant into a plurality of portions; and
   a plurality of second distribution devices each dividing refrigerant divided by the first distribution device into two portions to flow into two of the plurality of heat transfer tubes.

2. The refrigerant distribution device of claim 1, wherein each of the plurality of second distribution devices is constituted by a two-way branch pipe, and the two-way branch pipe includes a U-bend section formed in U-shape having a connecting section and two arms extending from each end of the connecting section in parallel to each other and a straight inflow section formed by performing bulge-forming process on one of the two arms of the U-bend section.

3. A refrigerant distribution device for distributing refrigerant to a plurality of heat transfer tubes that constitute a heat exchanger, the refrigerant distribution device comprising:

   a first distribution device dividing refrigerant into a plurality of portions;
   a plurality of second distribution devices each dividing refrigerant into two portions to flow into two of the plurality of heat transfer tubes; and
   a plurality of third distribution devices provided between the first distribution device and the plurality of second distribution devices to each divide refrigerant divided by the first distribution device into two portions to flow into adjacent two of the plurality of second distribution devices.

4. The refrigerant distribution device of claim 3, wherein each of the plurality of second distribution devices and each of the plurality of third distribution devices are each constituted by a two-way branch pipe, and the two-way branch pipe includes a U-bend section formed in U-shape having a connecting section and two arms extending from each end of the connecting section in parallel to each other and a straight inflow

section formed by performing bulge-forming process on one of the two arms of the U-bend section.

5. The refrigerant distribution device of claim 2 or claim 4, wherein the two-way branch pipe is formed such that an angle of 90 degrees is provided between the straight inflow section and the one of the two arms on which the straight inflow section is formed.

6. The refrigerant distribution device of claim 2 or claim 4, wherein the two-way branch pipe is formed such that the straight inflow section is formed to be inclined to the one of the two arms.

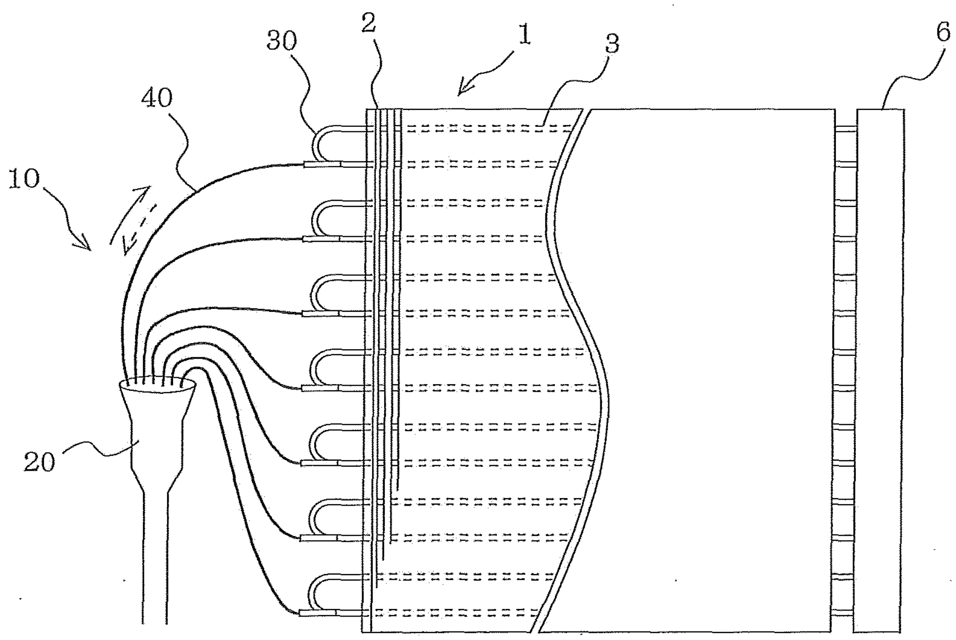
7. The refrigerant distribution device of claim 6, wherein an inclined angle of the straight inflow section to the one of the two arms is determined depending on thermal load distribution of each path of the heat exchanger.

8. The refrigerant distribution device of any one of claim 2 and claims 4 to 7, wherein each of the plurality of second distribution devices includes an inwardly depressed recess adjacent to a collision wall that is a wall surface that faces the straight inflow section in the one of the two arms.

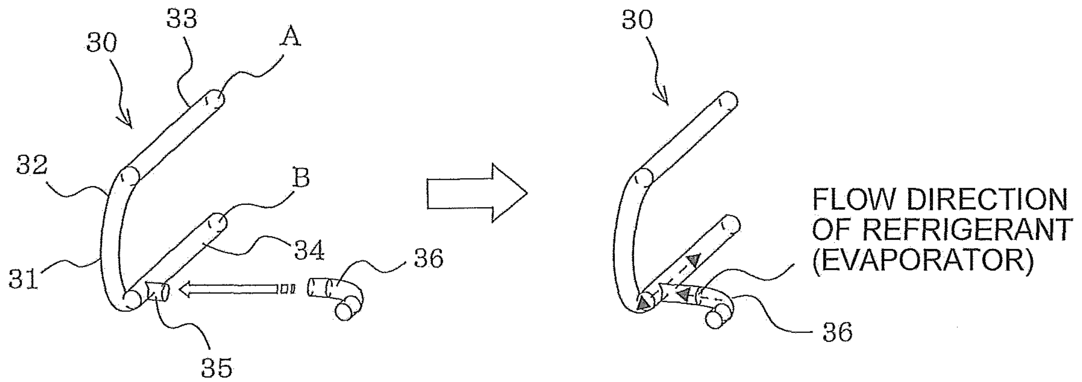
9. The refrigerant distribution device of any one of claims 1 to 8, wherein the first distribution device is constituted by a distributor or a header.

10. A heat pump apparatus using the refrigerant distribution device of any one of claims 1 to 9.

F I G. 1



F I G. 2



F I G. 3

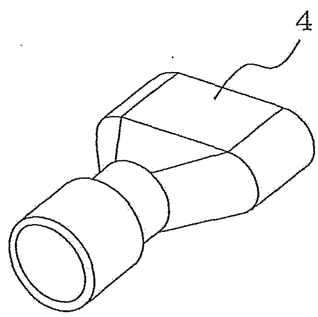


FIG. 4

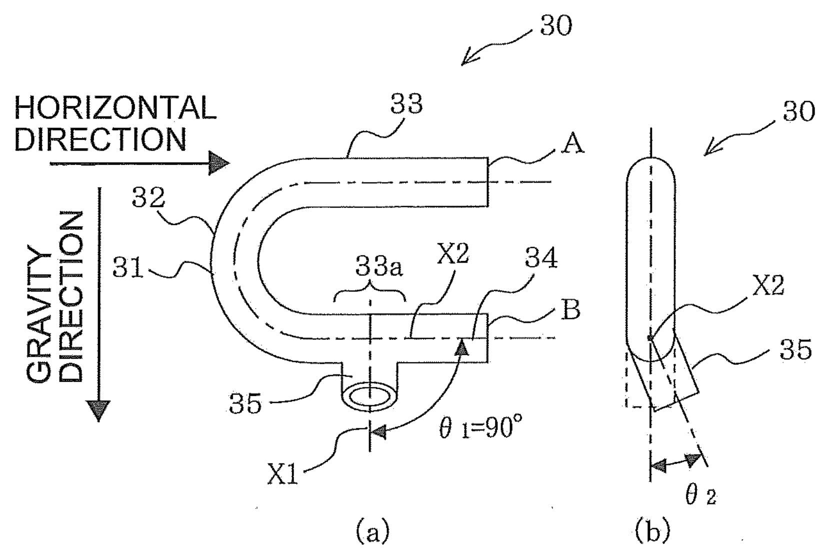
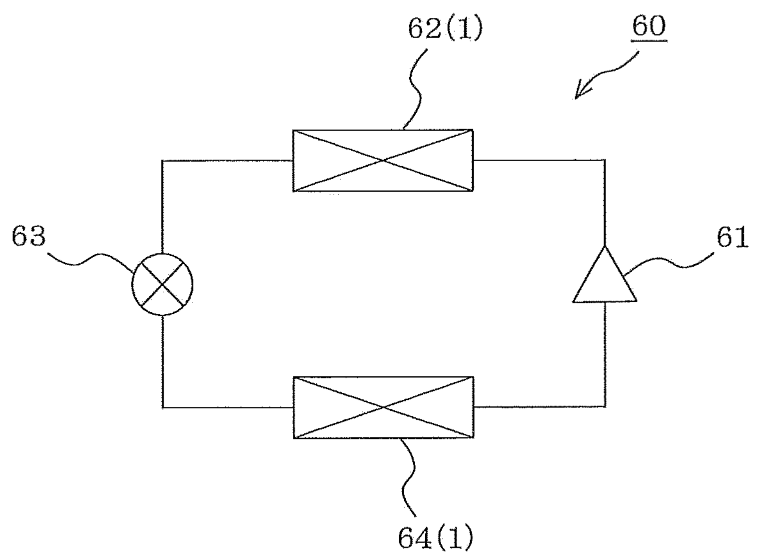
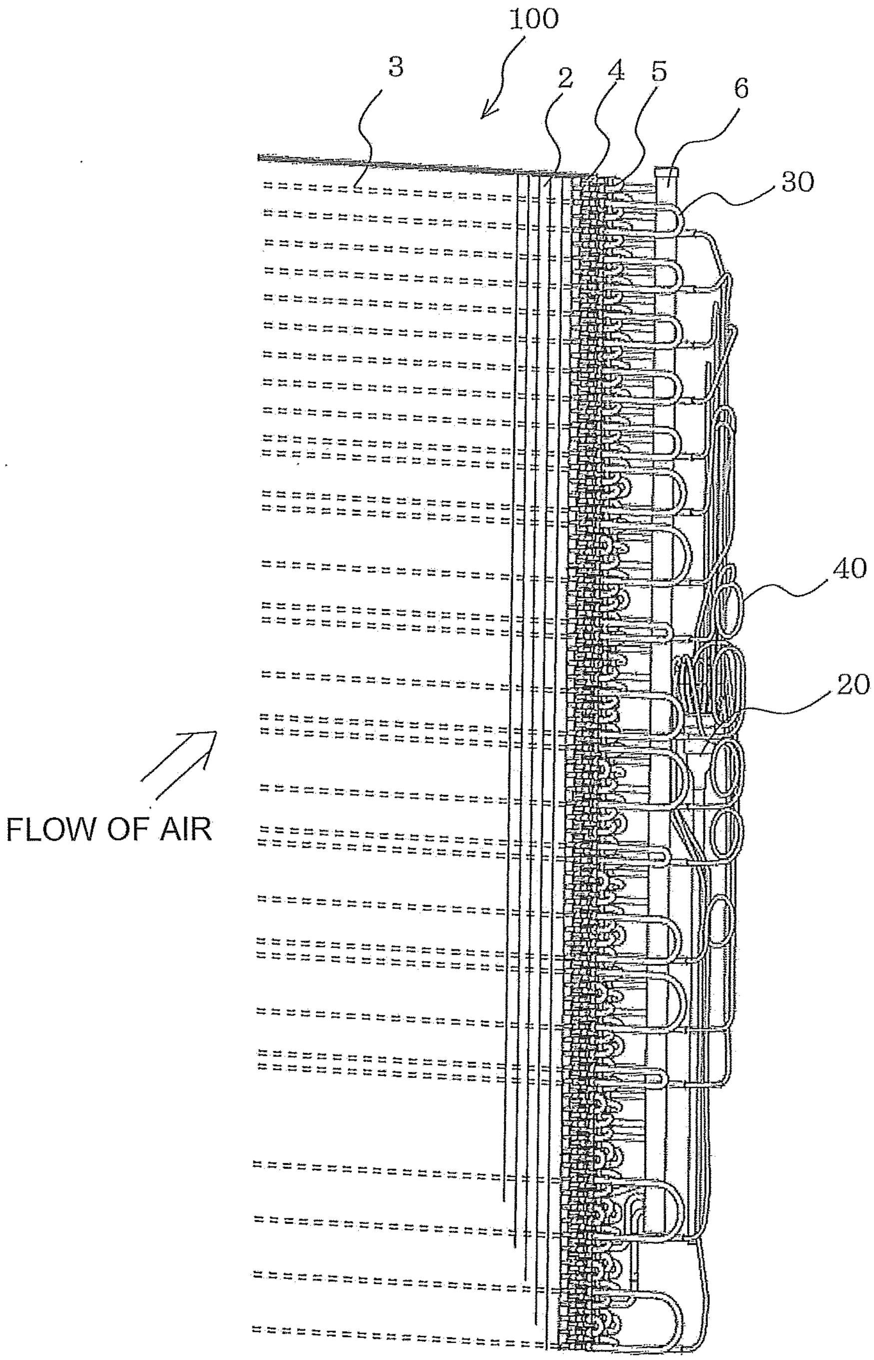

30
HORIZONTAL DIRECTION
GRAVITY DIRECTION
33
A
32
31
33a
X2
34
B
35
θ1=90°
X1
(a)
30
X2
35
θ2
(b)

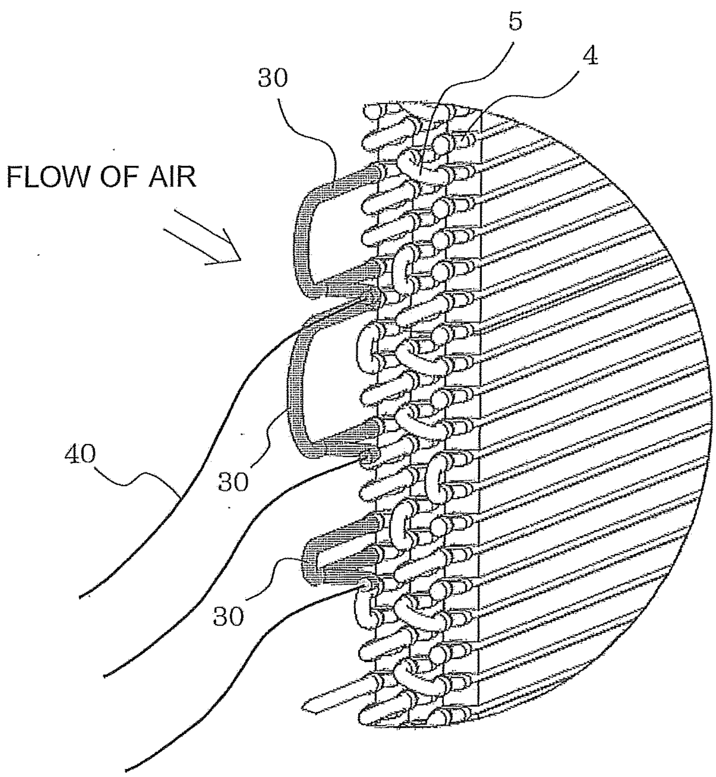
FIG. 5


62(1)
60
63
61
64(1)

F I G. 6



F I G. 7



F I G. 8

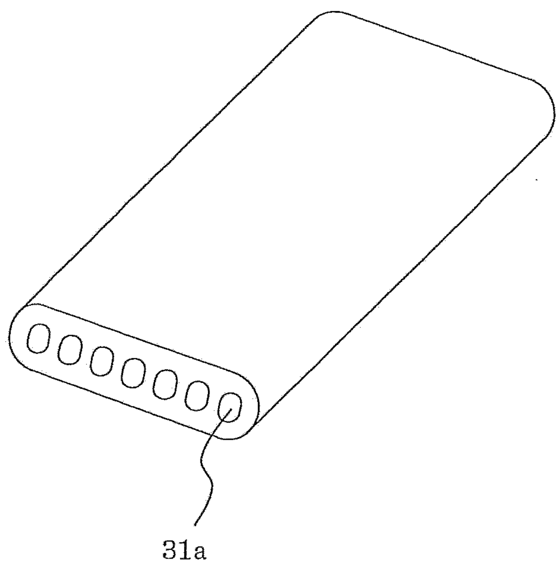


FIG. 9

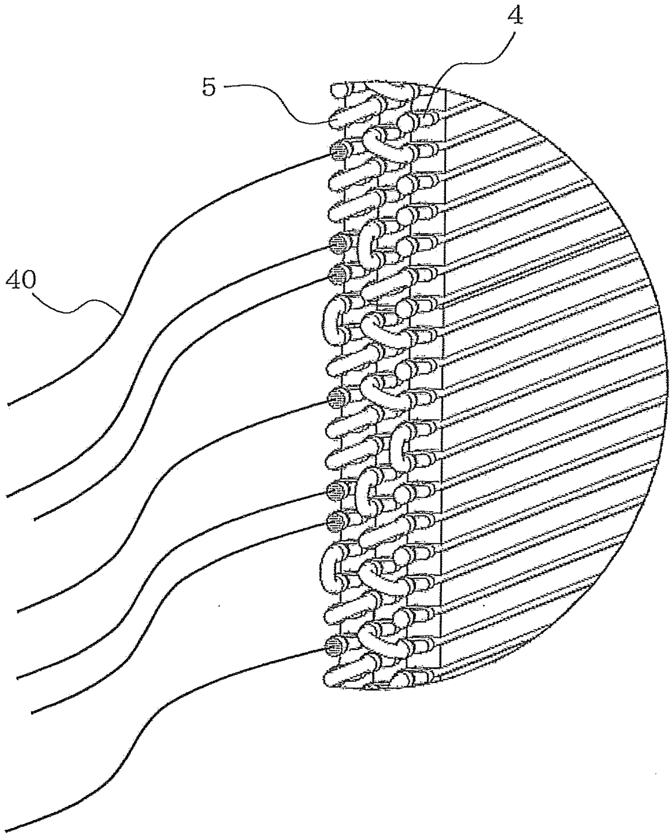
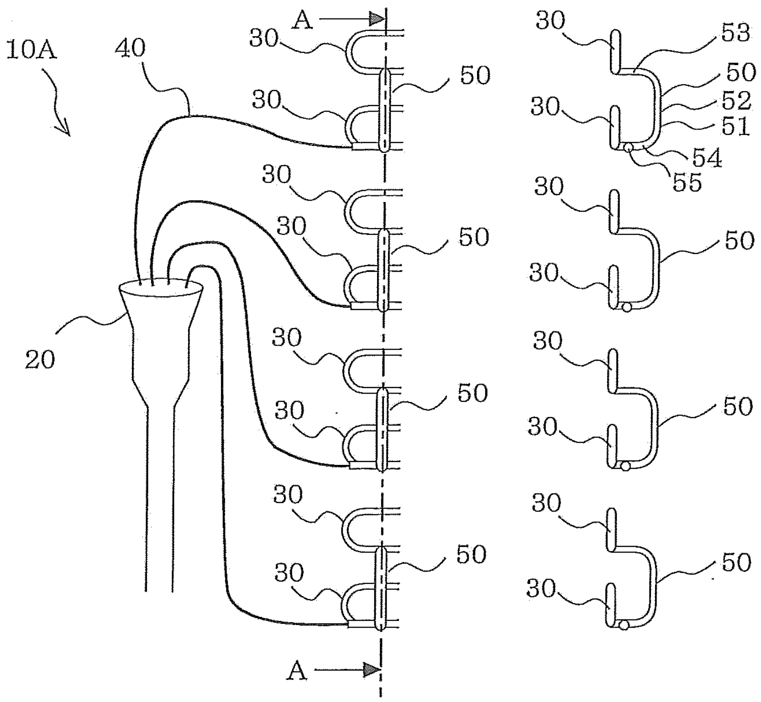
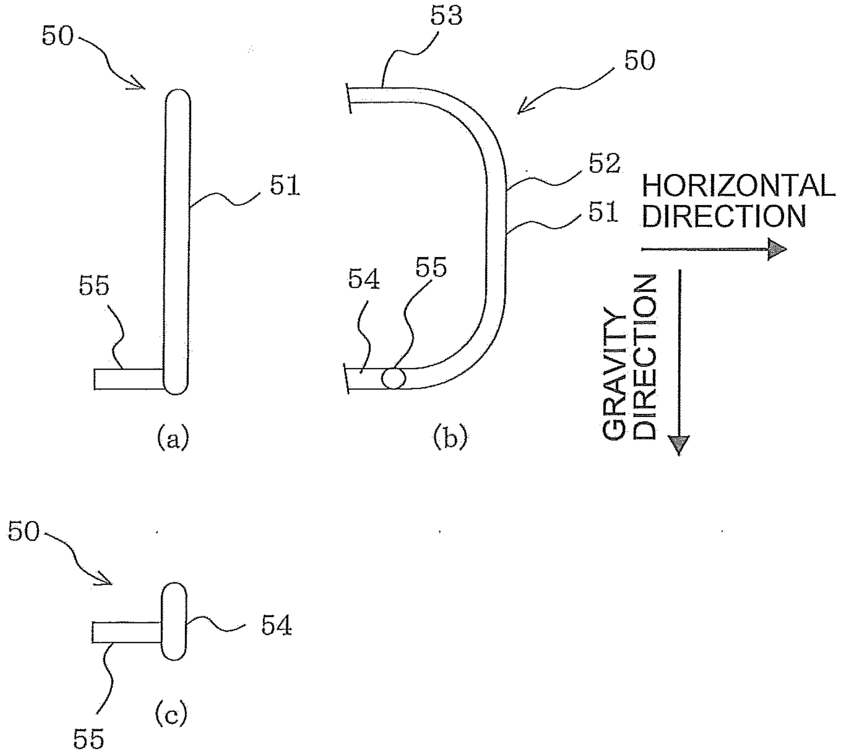


FIG. 10



(a) (b) CROSS SECTION BY A-A

F I G. 1 1


50
53
51
52
55
54
HORIZONTAL DIRECTION
GRAVITY DIRECTION
(a)
(b)
(c)

F I G. 1 2

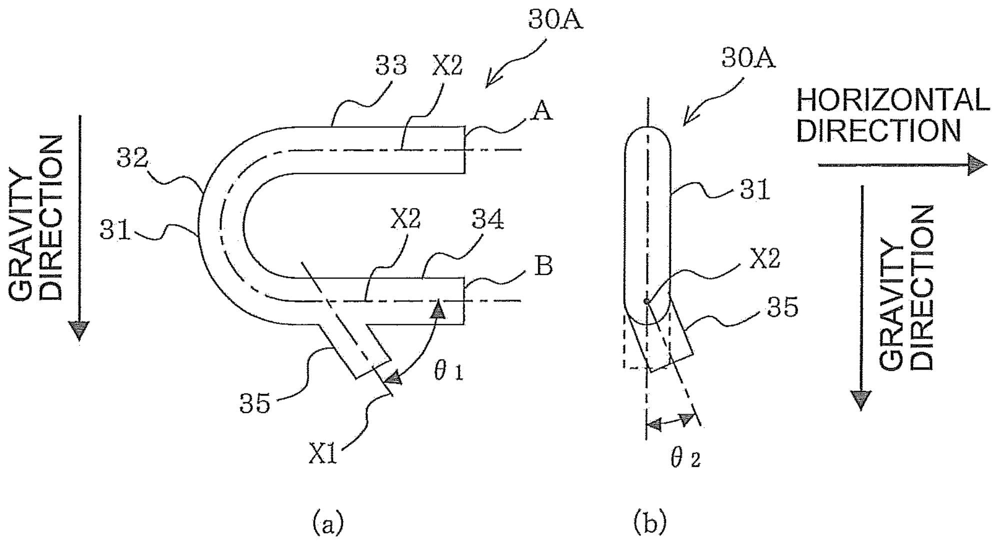
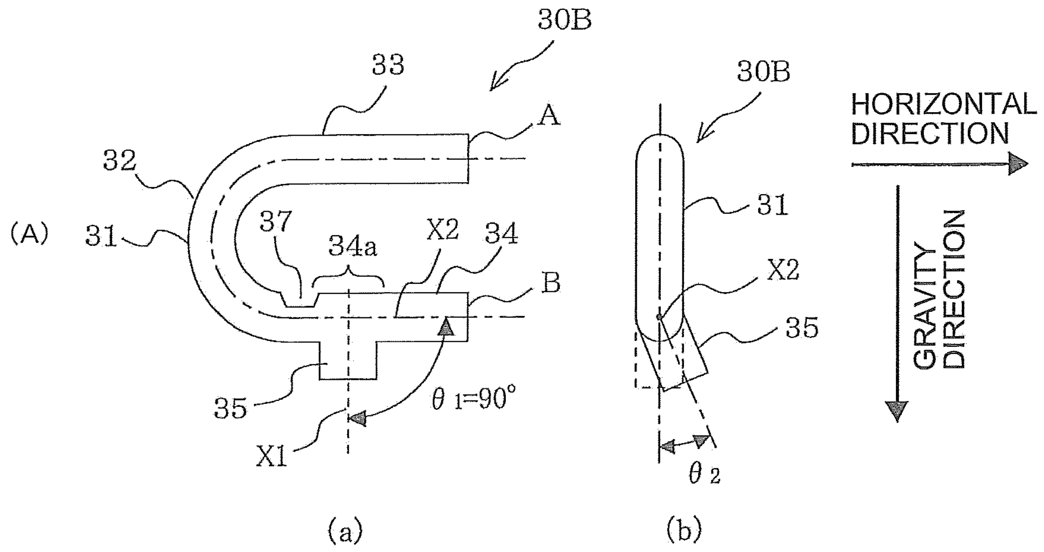

30A
33
X2
A
32
31
34
B
35
X1
θ1
θ2
HORIZONTAL DIRECTION
GRAVITY DIRECTION
(a)
(b)

FIG. 13


(A)
30B
33
A
32
31
37
34a
X2
34
B
35
X1
θ1=90°
(a)
30B
31
X2
35
θ2
(b)
HORIZONTAL DIRECTION
GRAVITY DIRECTION

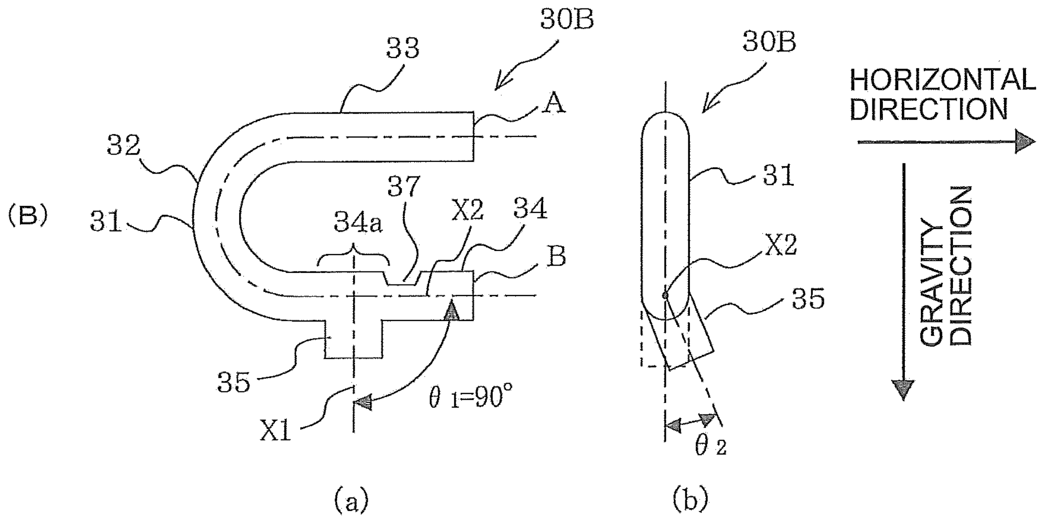
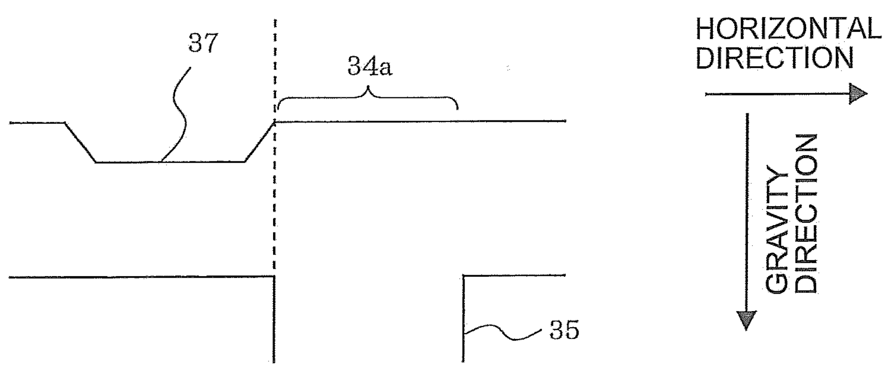
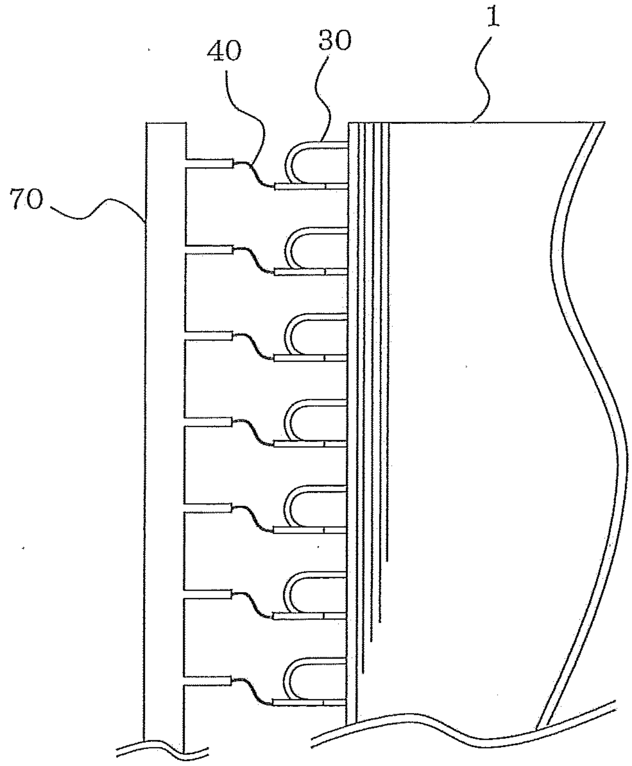

(B)
30B
33
A
32
31
37
34a
X2
34
B
35
X1
θ1=90°
(a)
30B
31
X2
35
θ2
(b)
HORIZONTAL DIRECTION
GRAVITY DIRECTION

FIG. 14


37
34a
35
HORIZONTAL DIRECTION
GRAVITY DIRECTION

ＦＩＧ．１５



INTERNATIONAL SEARCH REPORT

International application No. PCT/JP2013/051146

A. CLASSIFICATION OF SUBJECT MATTER
F25B41/00(2006.01)i, *F28F9/22*(2006.01)i

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)
F25B41/00, F28F9/22

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

Jitsuyo Shinan Koho	1922-1996	Jitsuyo Shinan Toroku Koho	1996-2013
Kokai Jitsuyo Shinan Koho	1971-2013	Toroku Jitsuyo Shinan Koho	1994-2013

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X A	WO 2011/135946 A1 (Daikin Industries, Ltd.), 03 November 2011 (03.11.2011), fig. 1 to 5; paragraphs [0019] to [0032] & JP 2011-247571 A	1,9,10 2-8
X A	JP 2009-186084 A (Hitachi Appliances, Inc.), 20 August 2009 (20.08.2009), fig. 1 to 3; paragraphs [0015] to [0021] (Family: none)	1,9,10 2-8
X A	JP 11-230637 A (Mitsubishi Heavy Industries, Ltd.), 27 August 1999 (27.08.1999), fig. 5; paragraphs [0017] to [0020] (Family: none)	1,9,10 2-8

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

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"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
12 March, 2013 (12.03.13)	26 March, 2013 (26.03.13)

Name and mailing address of the ISA/	Authorized officer
Japanese Patent Office	
Facsimile No.	Telephone No.

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INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2013/051146

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	JP 11-159917 A (Goh Shoji Co., Inc.), 15 June 1999 (15.06.1999), fig. 3 (Family: none)	8

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REFERENCES CITED IN THE DESCRIPTION

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