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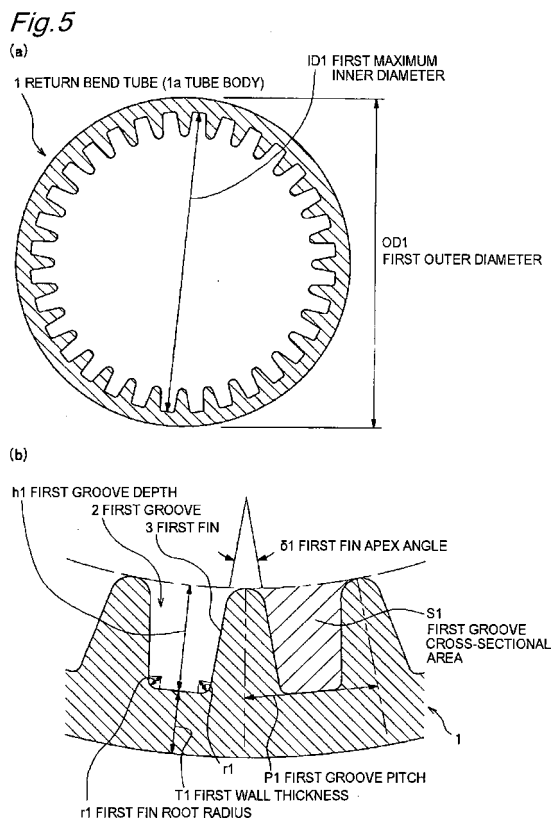
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(54) **FIN-AND-TUBE TYPE HEAT EXCHANGER, AND ITS RETURN BEND PIPE**

(57) The invention provides a fin-and-tube heat exchanger using a return bend tube that allows further enhancement of the evaporative performance of the heat exchanger. The fin-and-tube heat exchanger, where a refrigerant is supplied inside tubing, has a hairpin tube portion where a plurality of hairpin tubes are arranged, a return bend tube portion where there are arranged a plurality of return bend tubes joined to respective hairpin tube ends of the hairpin tube portion, and a fin portion comprising a plurality of fins arranged at a predetermined spacing on the outer surface of the hairpin tubes. The heat exchanger comprises first grooves formed on the tube inner surface of the return bend tube. A first groove pitch (P1) of the first grooves in a cross section perpendicular to a tube axis, and a second groove pitch (P2) of spiral-shaped second grooves formed on the inner surface of the hairpin tube in a cross section perpendicular to a tube axis, satisfy a groove pitch ratio (P1/P2) of 0.65 to 2.2, while a first groove cross-sectional area (S1) per groove of the first grooves in a cross section perpendicular to the tube axis, and a second groove cross-sectional area (S2) per groove of the second grooves in a cross section perpendicular to the tube axis satisfy a groove cross-sectional area ratio (S1/S2) of 0.3 to 3.6.



Description

BACKGROUND OF THE INVENTION

5 TECHNICAL FIELD

[0001] The present invention relates to a heat exchanger used in air-conditioners, in particular to a fin-and-tube heat exchanger in which a refrigerant such as a Freon-type refrigerant and a natural refrigerant flows inside tubes and a plurality of fins formed of aluminum or the like are arranged on the outer face of the tubes, and relates also to a return bend tube connected to a hairpin tube of the fin-and-tube heat exchanger.

BACKGROUND ART

[0002] JP-UM-A-63-154986 (Examples, Figs. 1 to 4) or JP-A-11-190597 (paragraphs 0022 to 0026, Fig. 1) describe conventional fin-and-tube heat exchangers using smooth tubes having a smooth inner surface as return bend tubes, and using inner surface grooved tubes as hairpin tubes. JP-UM-A-63-154986 (embodiments, Figs. 1 to 4) describes that the return bend tube is a U-bend tube, and the hairpin tube is a seam-welded tube, while JP-A-11-190597 (paragraphs 0022 to 0026, Fig. 1) describes that the return bend tube is a U-bend tube, and the hairpin tube is a heat-transfer tube.

[0003] JP-UM-A-04-122986 (paragraphs 0007 to 0008, Fig. 1) proposes a fin-and-tube heat exchanger for use in an evaporator, using an inner surface grooved tube as a return bend tube, and a smooth tube as a hairpin pipe. JP-UM-A-04-122986 describes that the return bend tube is a U-bend tube and the hairpin pipe is a tube.

JP-A-2006-98033 (claim 1, Fig. 4) describes a fin-and-tube heat exchanger using inner surface grooved tubes for both the return bend tube and the hairpin tube.

[0004] Meanwhile, the use of hydrochlorofluorocarbon refrigerants such as R22 (chlorodifluoromethane), conventionally employed as refrigerants for fin-and-tube heat exchangers, has been banned on environmental grounds, as they deplete the ozone layer. Hydrofluorocarbon refrigerants such as R410A, in which all chlorine is replaced by hydrogen, have thus begun to be extensively used as refrigerants for air conditioners.

PATENT DOCUMENT 1 JP-UM-A-63-154986 (Examples, Figs. 1-4)

PATENT DOCUMENT 2 JP-A-11-190597 (para. 0022-0026, Fig. 1)

PATENT DOCUMENT 3 JP-UM-A-04-122986 (para. 0007-0008, Fig. 1)

PATENT DOCUMENT 4 JP-A-2006-98033 (claim 1, Fig. 4)

PROBLEMS TO BE SOLVED BY THE INVENTION

[0005] In the heat exchangers described in JP-UM-A-63-154986 and JP-A-11-190597, the refrigerant flowing through the hairpin tubes develops a swirling flow along the grooves formed on the tube inner surface. This swirling flow persists for a while when the refrigerant flows into the return bend tube. Since the inner surface of the return bend tube is smooth, however, the swirling flow can be maintained only with difficulty at the outlet of the return bend tube, while there occurs droplet (refrigerant film) splashing at the bent portion of the return bend tube, which destabilizes the flow of the liquid film. Thus, such heat exchangers are problematic in that, after inflow into the next hairpin tube, some time is lost until swirling flow is created again in the refrigerant, and in that the flow of refrigerant becomes unstable over that section, while there form also thicker portions in the refrigerant film, all of which tends to decrease the inside-tube heat transfer coefficient and to preclude achieving sufficient evaporative performance.

[0006] In the heat exchanger of JP-UM-A-04-122986, grooves are formed inside the return bend tube but not inside the hairpin tubes, and hence there is a substantial inner-tube shape difference between the two tubes. This is problematic in that the heat exchanger experiences as a result a larger pressure loss of the refrigerant circulating inside the heat exchanger, and a decrease in the flow rate of the refrigerant, all of which lead to a dramatic loss of heat-transfer performance in the heat exchanger, in particular loss of evaporative performance.

[0007] When the wall thickness of the tubes is made thicker in light of the strength loss associated with the formation of grooves in the return bend tube, as in JP-UM-A-04-122986, there forms a bump at the inner surface of the joint between the return bend tube and the hairpin tube that hinders the flow of refrigerant and that is likely to increase refrigerant pressure loss.

[0008] The heat exchanger of JP-A-2006-98033 was also problematic in that the groove lead angle formed between the tube axis and the grooves formed on the return bend tube and the hairpin tube was limited to a predetermined lead angle, but no restrictions were set for the groove pitch and the groove cross-sectional area. Hence, refrigerant film disturbances were apt to occur inside the tubes, with the refrigerant film becoming uneven at the straight-tube portion of the hairpin tube, and with portions of the refrigerant film becoming thicker. As a result, sufficient evaporative performance

could not be achieved.

[0009] More specifically, an uneven refrigerant film means that the liquid film thickness is uneven. When the liquid film thickness becomes uneven there arises a state difference (function of the surface tension of the refrigerant film and the curvature of the liquid film) among portions where the liquid film is thick and portions where it is thin. When such a state difference arises, the thin refrigerant film is stretched in principle by the thick refrigerant film, as a result of which the thin liquid refrigerant film portions become even thinner, thereby promoting evaporation in such portions, while the portions where the refrigerant film is thick persist. Such persisting refrigerant film has the effect of bringing about a dry-out state outside the refrigerant-film persisting portions, which reduces the effective heat transfer surface and impairs evaporative performance.

[0010] In light of the above problems, it is an object of the present invention to provide a fin-and-tube heat exchanger and a return bend tube thereof that allow further enhancement of the evaporative performance of a heat exchanger.

MEANS TO SOLVE THE PROBLEMS

[0011] In a first aspect of the invention, there is provided a fin-and-tube heat exchanger in which a refrigerant is supplied inside tubing and which has: a hairpin tube portion where a plurality of hairpin tubes are arranged; a return bend tube portion where there are arranged a plurality of return bend tubes joined to respective hairpin tube ends of the hairpin tube portion; and a fin portion comprising a plurality of fins arranged at a predetermined spacing on the outer surface of the hairpin tubes, the fin-and-tube heat exchanger further comprising:

first grooves formed on a tube inner surface of the return bend tube,

wherein a first groove pitch (P1) of the first grooves in a cross section perpendicular to a tube axis, and a second groove pitch (P2) of spiral-shaped second grooves formed on the inner surface of the hairpin tube in a cross section perpendicular to a tube axis, satisfy a groove pitch ratio (P1/P2) of 0.65 to 2.2,

and wherein a first groove cross-sectional area (S1) per groove of the first grooves in a cross section perpendicular to the tube axis, and a second groove cross-sectional area (S2) per groove of the second grooves in a cross section perpendicular to the tube axis satisfy a groove cross-sectional area ratio (S1/S2) of 0.3 to 3.6.

[0012] In such a constitution, the predetermined first grooves formed in the inner surface of the return bend tubes in the fin-and-tube heat exchanger allow flattening the refrigerant film at the return bend tube inlet side, and allow forming "annular flow" in the refrigerant film inside the tubes, thus reducing refrigerant film disturbance in the return bend tube. During inflow of liquid refrigerant from the return bend tube outlet side into the next hairpin tube, there forms thus a more homogeneous "annular flow", so that the refrigerant film becomes uniform at the straight-tube portion of the hairpin tubes, stabilizing thus heat exchange with the exterior of the tube and further enhancing evaporative performance.

[0013] Preferably, a second groove lead angle (θ_2) formed between the tube axis and the second grooves of the hairpin tube is 15° or more.

[0014] In such a constitution, a more homogeneous "annular flow" forms during inflow of liquid refrigerant from the return bend tube outlet side into the next hairpin tube, so that the refrigerant film becomes uniform at the straight-tube portion of the hairpin tubes, stabilizing thus heat exchange with the exterior of the tube and further enhancing evaporative performance.

[0015] Preferably, a refrigerant flow channel comprising the hairpin tube and the return bend tube is at least partially branched, forming a plurality of refrigerant flow channels.

[0016] In such a constitution, the refrigerant flow channel of the fin-and-tube heat exchanger is branched, whereby the refrigerant mass rate per branching decreases, and in particular the refrigerant velocity decreases at the return bend tube inlet side, which stabilizes further the "annular flow" of the refrigerant film formed inside the tubes. During inflow of liquid refrigerant from the return bend tube outlet side into the next hairpin tube, thus, there forms amore homogeneous "annular flow", so that the refrigerant film becomes uniform at the straight-tube portion of the hairpin tube, stabilizing thus heat exchange with the exterior of the tube and enhancing evaporative performance.

[0017] Preferably, the refrigerant is a hydrofluorocarbon-type non-azeotropic blend refrigerant.

Such a constitution further enhances evaporative performance in the heat exchanger while reducing refrigerant pressure loss.

[0018] In a second aspect of the invention, there is provided a return bend tube which is used in a fin-and-tube heat exchanger where a refrigerant is supplied inside tubing, and is joined to the tube end of a hairpin tube comprising a plurality of fins arranged at a predetermined spacing on the outer surface thereof, the return bend tube comprising:

first grooves formed on a tube inner surface of the return bend tube,

wherein a first groove pitch (P1) of the first grooves in a cross section perpendicular to a tube axis, and a second groove

pitch (P2) of spiral-shaped second grooves formed on the inner surface of the hairpin tube in a cross section perpendicular to a tube axis, satisfy a groove pitch ratio ($P1/P2$) of 0.65 to 2.2,

and wherein a first groove cross-sectional area ($S1$) per groove of the first grooves in a cross section perpendicular to the tube axis, and a second groove cross-sectional area ($S2$) per groove of the second grooves in a cross section perpendicular to the tube axis satisfy a groove cross-sectional area ratio ($S1/S2$) of 0.3 to 3.6.

[0019] In such a constitution, the liquid refrigerant "swirling flow" formed in the return bend tubes and the hairpin tubes is maintained by setting a predetermined range for the groove pitch ratio ($P1/P2$) and the groove cross-sectional area ratio ($S1/S2$). At the same time, this allows flattening the refrigerant film at the return bend tube inlet side during refrigerant inflow from the hairpin tube into the return bend tube, and allows the refrigerant film to form a uniform "annular flow" inside the tube. Refrigerant liquid disturbance inside the return bend tube is thus reduced as a result. During inflow of liquid refrigerant from the return bend tube outlet side into the next hairpin tube, there forms a more homogeneous "annular flow", so that the refrigerant film becomes uniform at the straight-tube portion of the hairpin tube, stabilizing thus heat exchange with the exterior of the tube (atmosphere) and enhancing evaporative performance.

[0020] Preferably, a first groove lead angle ($\theta1$) formed between the tube axis and the first grooves and a second groove lead angle ($\theta2$) formed between the tube axis and the second grooves satisfy an angle difference ($\theta1-\theta2$) of -15° to $+15^\circ$, and a first groove depth ($h1$) of the first grooves in a cross section perpendicular to the tube axis, and a second groove depth ($h2$) of the second grooves in a cross section perpendicular to the tube axis, satisfy a groove depth ratio ($h1/h2$) of 0.47 to 1.5.

[0021] By setting a predetermined range for the angle difference ($\theta1-\theta2$) of the groove lead angles, such a constitution allows curbing refrigerant film splashing during refrigerant inflow from the hairpin tube into the return bend tube. During inflow of liquid refrigerant from the return bend tube outlet side into the next hairpin tube, also, there forms a more homogeneous "annular flow", so that the refrigerant film becomes uniform at the straight-tube portion of the hairpin tube, stabilizing thus heat exchange with the exterior of the tube and enhancing evaporative performance.

Also, setting a predetermined range for the groove depth ratio ($h1/h2$) hampers separation of the refrigerant from the inner surface of the tubes, thus reducing refrigerant film disturbance. During inflow of liquid refrigerant from the return bend tube outlet side into the next hairpin tube, also, there forms a more homogeneous "annular flow", so that the refrigerant film becomes uniform at the straight-tube portion of the hairpin tube, stabilizing thus heat exchange with the exterior of the tube and enhancing evaporative performance.

[0022] Preferably, a length (L) of the return bend tube is 1.0 to 1.5 times a pitch (P).

[0023] When the return bend tube is joined to the straight-tube section of the hairpin tube, setting the length (L) of the return bend tube to be a predetermined multiple of the bending pitch (P) in accordance with the above constitution has the effect of allowing sufficient "annular flow" to form in the refrigerant film at the straight-tube portion, from the return bend tube inlet to the bent portion. As a result, no refrigerant film disturbance (separated flow) occurs in the bent portion of the return bend tube. When flowing thus into the next hairpin tube, the refrigerant flows with "annular flow" formed therein, so that the refrigerant film becomes uniform at the straight-tube portion of the hairpin tube, stabilizing thus heat exchange with the exterior of the tube and further enhancing evaporative performance.

[0024] Preferably, a material of the return bend tube comprises a material having a lower thermal conductivity than a material of the hairpin tube.

[0025] Heat loss at the return bend tube is suppressed in such a constitution by making the thermal conductivity of the tube body (return bend tube) lower than that of the hairpin tube. Suppressing heat loss at the return bend tube allows preventing refrigerant film disturbance (separated flow) caused by refrigerant film splashing, while forestalling refrigerant evaporation inside the return bend tube and collapse of the "annular flow" of the refrigerant film. When flowing thus into the next hairpin tube, the refrigerant flows with "annular flow" formed therein, whereby the refrigerant film becomes uniform at the straight-tube portion of the hairpin tube, stabilizing thus heat exchange with the exterior of the tube and further enhancing evaporative performance.

[0026] Preferably, a material of the return bend tube comprises a copper alloy more heat resistant than a material of the hairpin tube.

[0027] Since in such a constitution the return bend tube comprises a heat-resistant copper alloy, there is less tube strength loss of the return bend tube after joining (brazing) of the return bend tube and the hairpin tube. As a result, the pressure inside the tubes in use of a heat exchanger makes no break of the return bend tubes at the heat affected portions by the brazing.

This makes thickening of the return bend tube walls unnecessary.

[0028] Preferably, a relationship between a first maximum inner diameter (ID1) of the return bend tube and a second maximum inner diameter (ID2) of the hairpin tube is $ID1 \geq ID2$.

[0029] Such a constitution allows the "annular flow" state to be preserved even more homogeneously during inflow of liquid refrigerant from the return bend tube into the hairpin tube, while spreading the refrigerant film, in the circumferential direction, in the vicinity of the hairpin tube inlet side, thus affording a thinner refrigerant film. Evaporative performance is further enhanced thereby at the straight-tube portion of the hairpin tube.

EFFECT OF THE INVENTION

[0030] By using the above return bend tube, the fin-and-tube heat exchanger according to the first aspect of the present invention allows further enhancement of the evaporative performance of a heat exchanger. The evaporative performance of the heat exchanger can also be further enhanced by using a hairpin tube having a groove lead angle within a predetermined range, and by using a branched refrigerant flow channel and a predetermined refrigerant.

[0031] By setting predetermined ranges for the groove pitch and the groove cross-sectional area of the first grooves of a return bend tube, the return bend tube according to the second aspect of the present invention allows forming "annular flow" in the refrigerant film inside the tubes while uniformizing the thickness of the refrigerant film at the straight-tube portion of a hairpin tube, thereby enhancing the evaporative performance of a heat exchanger. Also, setting a predetermined range for the groove lead angle, groove depth, length, thermal conductivity and maximum inner diameter of the first grooves of the return bend tube allows further enhancement of the evaporative performance of the heat exchanger. Moreover, building the return bend tube using a heat-resistant copper alloy has the effect of increasing the reliability of joints with hairpin tubes, making it thus possible to achieve more light-weight constitutions.

BRIEF DESCRIPTION OF THE DRAWINGS

[0032]

Fig. 1 is a perspective view illustrating the constitution of a return bend tube according to the present invention; Fig. 2 is a partially cut-away front view illustrating an example of a fin-and-tube heat exchanger that incorporates the return bend tube according to the present invention; Fig. 3 (a) is a perspective view of the heat exchanger of Fig. 2 viewed from the return bend tube, Fig. 3(b) is a perspective view of the heat exchanger viewed from a hairpin tube, and Fig. 3(c) is a schematic view illustrating schematically the flow of refrigerant inside the heat exchanger; Fig. 4 is an enlarged end cross-sectional view illustrating an example of a joint between a hairpin tube and a return bend tube, cut along the axial direction of the tube; Fig. 5(a) is an end cross-sectional view, perpendicular to the tube axis, of the return bend tube, and Fig. 5(b) is a partial enlarged end cross-sectional view of Fig. 5(a); Fig. 6(a) is an end view, perpendicular to the tube axis, of the hairpin bend, and Fig. 6(b) is a partial enlarged end view of Fig. 6(a); Figs. 7(a) and 7(b) are schematic views illustrating schematically the flow of refrigerant inside a heat exchanger in another embodiment according to the present invention; and Fig. 8(a) is a schematic view of a suction-type wind tunnel used for measuring the evaporative performance of a heat exchanger, and Fig. 8 (b) is a schematic view of a refrigerant supply apparatus for supplying refrigerant to the suction-type wind tunnel of Fig. 8(a).

[0033]

1	return bend tube
1a	tube body
2	first groove
3	first fin
11	hair pin tube
12	second groove
13	second fin
20, 20A,	20B heat exchanger
21	fin portion
21a	fin
22	return bend tube portion
23	hairpin tube portion
P1	first groove pitch
P2	second groove pitch
S1	first groove cross-sectional area
S2	second groove cross-sectional area
θ1	first groove lead angle
θ2	second groove lead angle
h1	first groove depth

h2	second groove depth
L	length
P	pitch
ID1	first maximum inner diameter
ID2	second maximum inner diameter
OD1	first outer diameter
OD2	second outer diameter

BEST MODE FOR CARRYING OUT THE INVENTION

[0034] The present invention is explained in detail next with reference to accompanying drawings. Fig. 1 is a perspective view illustrating the constitution of a return bend tube according to the present invention; Fig. 2 is a partially cut-away front view illustrating an example of a fin-and-tube heat exchanger that incorporates the return bend tube according to the present invention; Fig. 3(a) is a perspective view of the heat exchanger of Fig. 2 viewed from the return bend tube, Fig. 3(b) is a perspective view of the heat exchanger viewed from a hairpin tube, and Fig. 3(c) is a schematic view illustrating schematically the flow of refrigerant inside the heat exchanger; Fig. 4 is an enlarged end cross-sectional view illustrating an example of a joint between a hairpin tube and a return bend tube, cut along the axial direction of the tube; Fig. 5(a) is an end cross-sectional view, perpendicular to the tube axis, of the return bend tube, and Fig. 5 (b) is a partial enlarged end cross-sectional view of Fig. 5(a); Fig. 6(a) is an end cross-sectional view, perpendicular to the tube axis, of the hairpin bend, and Fig. 6 (b) is a partial enlarged end cross-sectional view of Fig. 6 (a) ; Figs. 7 (a) and 7 (b) are schematic views illustrating schematically the flow of refrigerant inside a heat exchanger in another embodiment according to the present invention; and Fig. 8(a) is a schematic view of a suction-type wind tunnel used for measuring the evaporative performance of a heat exchanger, and Fig. 8(b) is a schematic view of a refrigerant supply apparatus for supplying refrigerant to the suction-type wind tunnel of Fig. 8(a).

(1) Return bend tube

[0035] The return bend tube of the present invention is explained first. As illustrated in Figs. 1 through 3, the return bend tube 1 of the present invention, which is used in a fin-and-tube heat exchanger 20 (hereinafter "heat exchanger" for short), is joined to the tube end of a hairpin tube 11 through which refrigerant is supplied. The return bend tube 1 comprises a U-shaped tube body 1a, a tube end 1b for connecting the tube end of the tube body 1a with the hairpin tube 11, and a plurality of first grooves 2 formed on the inner surface of the tube body 1a (the first grooves have been omitted in Fig. 1, refer to Fig. 4). The return bend tube 1 is interposed between two hairpin tubes 11, to connect the respective hairpin tubes 11. As illustrated in Fig. 2, a long-stretch refrigerant flow channel can thus be achieved by connecting in series the plurality of hairpin tubes 11, 11

[0036] The evaporative performance of the heat exchanger 20 (Figs. 2 and 3) into which the return bend tube 1 is built can be enhanced by controlling as described below the inner surface groove shape of the first grooves 2 plurally formed on the tube inner surface of the return bend tube 1, as illustrated in Figs. 5 and 6. Since the outer diameter (second outer diameter OD2) of the hairpin tube 11 joined to the return bend tube 1 ranges from 3 to 10 mm, the outer diameter (first outer diameter OD1) of the return bend tube 1 ranges preferably from 3 to 10 mm.

<Inner surface groove shape>

[0037] The first grooves 2 of the return bend tube 1 must satisfy a groove pitch ratio ($P1/P2$) of 0.65 to 2.2, wherein ($P1$) is a first groove pitch of the return bend tube 1 in a cross section perpendicular to the tube axis, and ($P2$) is a second groove pitch of spiral-shaped second grooves 12 formed on the inner surface of the hairpin tube 11, in a cross section perpendicular to the tube axis. Also, a first groove cross-sectional area ($S1$) per groove of the first grooves 2 in a cross section perpendicular to the tube axis, and a second groove cross-sectional area ($S2$) per groove of the second grooves 12 in a cross section perpendicular to the tube axis, must satisfy a groove cross-sectional area ratio ($S1/S2$) of 0.3 to 3.6. More preferably, the groove cross-sectional area ratio ($S1/S2$) ranges from 0.54 to 2.7. The rationale for setting such numerical value limits for the groove pitch ratio ($P1/P2$) and the groove cross-sectional area ratio ($S1/S2$) are explained next.

(Groove pitch ratio ($P1/P2$): 0.65 to 2.2)

[0038] When the groove pitch ratio ($P1/P2$) is less than 0.65, the number of grooves in the return bend tube 1 increases with respect to one groove in the hairpin tube 11, so that when liquid refrigerant flows from the hairpin tube 11 into the return bend tube 1, contracted flow occurs in the refrigerant film inside the tube (first grooves 2) at the return bend tube

inlet side, thereby disrupting the refrigerant film. When flowing thus into the next hairpin tube 11, the refrigerant film does so in a disrupted state, whereby portions of the refrigerant film thicken at the straight-tube portion of the hairpin tube, destabilizing thus heat exchange with the exterior of the tube and eventually impairing evaporative performance.

[0039] When the groove pitch ratio ($P1/P2$) exceeds 2.2, the number of grooves in the return bend tube 1 decreases with respect to one groove in the hairpin tube 11. As a result, the holding ability of the refrigerant film becomes greatly reduced in the first grooves 2 in the return bend tube 1 when liquid refrigerant flows from the hairpin tube 11 into the return bend tube 1, the formation of "annular flow" breaks down, and the refrigerant film is disrupted. When flowing thus into the next hairpin tube 11, the refrigerant film does so in a disrupted state, whereby portions of the refrigerant film thicken at the straight-tube portion of the hairpin tube 11, destabilizing thus heat exchange with the exterior of the tube and eventually impairing evaporative performance.

(Groove cross-sectional area ratio ($S1/S2$): 0.3 to 3.6)

[0040] When the groove cross-sectional area ratio ($S1/S2$) is less than 0.3, the cross-sectional area of the first grooves 2 is largely reduced, so that when liquid refrigerant flows from the hairpin tube 11 into the return bend tube 1, contracted flow occurs in the refrigerant film at the return bend tube inlet, thereby disrupting the refrigerant film. When flowing thus into the next hairpin tube 11, the refrigerant film does so in a disrupted state, whereby portions of the refrigerant film thicken at the straight-tube portion of the hairpin tube 11, destabilizing thus heat exchange with the exterior of the tube and eventually impairing evaporative performance.

[0041] When the groove cross-sectional area ratio ($S1/S2$) exceeds 3.6, although flowing resistance of the refrigerant drops thanks to the increased cross-sectional area of the first grooves 2, the holding ability of the refrigerant film of the first groove 2 becomes greatly reduced by contrast when liquid refrigerant flows from the hairpin tube 11 into the return bend tube 1. As a result, the formation of "annular flow" breaks down, and the refrigerant film is disrupted. When flowing thus into the next hairpin tube 11, the refrigerant film does so in a disrupted state, whereby portions of the refrigerant film thicken at the straight-tube portion of the hairpin tube 11, destabilizing thus heat exchange with the exterior of the tube and eventually impairing evaporative performance.

[0042] As illustrated in Figs. 4 to 6, in the first grooves 2 of the return bend tube 1, preferably, an angle difference ($\theta1-\theta2$) satisfies -15 to $+15^\circ$, wherein ($\theta1$) is a first groove lead angle formed between the first grooves 2 and the tube axis, and ($\theta2$) is a second groove lead angle formed between the second grooves 12 provided on the inner surface of the hairpin tube 11 and the tube axis, while a groove depth ratio ($h1/h2$) satisfies 0.47 to 1.5, wherein ($h1$) is a first groove depth of the first grooves 2 in a cross section perpendicular to the tube axis, and ($h2$) is a second groove depth of the second grooves 12 in a cross section perpendicular to the tube axis. The first grooves 2 may have a first groove lead angle ($\theta1$) of 0° , i.e., the first grooves 2 may be parallel to the tube axis. The rationale for setting such numerical value limits for the angle difference ($\theta1-\theta2$) and the groove depth ratio ($h1/h2$) is explained next.

(Angle difference ($\theta1-\theta2$): -15 to $+15^\circ$)

[0043] When the angle difference ($\theta1-\theta2$) is less than -15° , i.e. when the first groove lead angle ($\theta1$) is smaller than (second groove lead angle ($\theta2$)- 15°), the refrigerant film splashes at the apex of first fins 3 formed between the first grooves 2, whereby the refrigerant film becomes disrupted (separated flow) in the return bend tube inlet side. When flowing thus into the next hairpin tube 11, the refrigerant film does so in a disrupted state, whereby portions of the refrigerant film thicken at the straight-tube portion of the hairpin tube 11, destabilizing thus heat exchange with the exterior of the tube and eventually impairing evaporative performance.

[0044] If the angle difference ($\theta1-\theta2$) exceeds $+15^\circ$, i.e. if the first groove lead angle ($\theta1$) is greater than (second groove lead angle ($\theta2$) $+15^\circ$), pressure loss on the return bend tube side becomes greater when the liquid refrigerant flows from the hairpin tube 11 into the return bend tube 1, thereby giving rise to contracted flow in the refrigerant film at the return bend tube inlet side and disrupting the refrigerant film. When flowing thus into the next hairpin tube 11, the refrigerant film does so in a disrupted state, whereby portions of the refrigerant film thicken at the straight-tube portion of the hairpin tube 11, destabilizing thus heat exchange with the exterior of the tube and eventually impairing evaporative performance.

[0045] The direction of the first groove lead angle ($\theta1$) formed between the first grooves 2 and the tube axis, and the direction of the second groove lead angle ($\theta2$) formed between the second grooves 12 provided on the inner surface of the hairpin tube 11 and the tube axis, are preferably the same direction. If the direction of the first groove lead angle ($\theta1$) and the direction of the second groove lead angle ($\theta2$) are different, refrigerant pressure loss at the return bend tube 1 becomes greater, which impairs evaporative performance.

(Groove depth ratio (h_1/h_2): 0.47 to 1.5)

[0046] If the groove depth ratio (h_1/h_2) is smaller than 0.47, the refrigerant film of the first grooves 2 tends to separate from the inner surface at the return bend tube inlet side, so that the refrigerant film splashes and becomes disrupted (separated flow). When flowing thus into the next hairpin tube 11, the refrigerant film does so in a disrupted state, whereby portions of the refrigerant film thicken at the straight-tube portion of the hairpin tube 11, destabilizing thus heat exchange with the exterior of the tube and eventually impairing evaporative performance.

[0047] If the groove depth ratio (h_1/h_2) is greater than 1.5, the first fins 3 of the return bend tube 1 offer resistance when the liquid refrigerant flows from the hairpin tube 11 into the return bend tube 1, thereby giving rise to contracted flow in the refrigerant film at the return bend tube inlet side and disrupting the refrigerant film. When flowing thus into the next hairpin tube, the refrigerant film does so in a disrupted state, whereby portions of the refrigerant film thicken at the straight-tube portion of the hairpin tube 11, destabilizing thus heat exchange with the exterior of the tube and eventually impairing evaporative performance.

[0048] Preferably, a first fin apex angle (δ_1) and a first fin root radius (r_1) of the first fins 3 formed between first grooves 2 of the return bend tube 1 are identical to a second fin apex angle (δ_2) and a second fin root radius (r_2) of the second fins 13 formed between second grooves 12 of the hairpin tube 11. More preferably, the first fin apex angle (δ_1) ranges from 4.5 to 45°, and the first fin root radius (r_1) ranges from 1/12 to 1/2 of the first groove depth (h_1). Ideally, the first fin apex angle (δ_1) ranges from 4.5 to 28.5°, and the first fin root radius (r_1) ranges from 1/12 to 1/4 of the first groove depth (h_1). Formation of "annular flow" by the refrigerant film at the return bend tube 1 is further maintained thereby.

This enhances even more, as a result, the evaporative performance of the heat exchanger 20 (Figs. 2 and 3). The rationale for setting such numerical value limits for the first fin apex angle (δ_1) and the first fin root radius (r_1) is explained next.

(First fin apex angle (δ_1): 4.5 to 45°)

[0049] When the first fin apex angle (δ_1) is smaller than 4.5°, flowing resistance of the refrigerant drops thanks to the increased cross-sectional area of the first grooves 2, whereas the holding ability of the refrigerant film becomes greatly reduced owing to the widening of the groove bottom of the first grooves 2, when liquid refrigerant flows from the hairpin tube 11 into the return bend tube 1. As a result, the formation of "annular flow" breaks down, and the refrigerant film is disrupted. When flowing thus into the next hairpin tube 11, the refrigerant film does so in a disrupted state, whereby portions of the refrigerant film thicken at the straight-tube portion of the hairpin tube 11, destabilizing thus heat exchange with the exterior of the tube and eventually impairing evaporative performance.

[0050] When the first fin apex angle (δ_1) exceeds 45°, the reduced cross-sectional area of the first grooves 2 is likely to give rise to contracted flow of the refrigerant film at the return bend tube inlet side during inflow of refrigerant from the hairpin tube 11 into the return bend tube 1, thereby disrupting the refrigerant film. When flowing thus into the next hairpin tube 11, the refrigerant film does so in a disrupted state, whereby portions of the refrigerant film thicken at the straight-tube portion of the hairpin tube 11, destabilizing thus heat exchange with the exterior of the tube and eventually impairing evaporative performance.

(First fin root radius (r_1): 1/12 to 1/2 of the first groove depth (h_1))

[0051] If the first fin root radius (r_1) is smaller than 1/12 of the first groove depth (h_1), flowing resistance of the refrigerant drops thanks to the increased cross-sectional area of the first grooves 2, whereas the holding ability of the refrigerant film becomes greatly reduced owing to the widening of the groove bottom of the first grooves 2 when liquid refrigerant flows from the hairpin tube 11 into the return bend tube 1. As a result, the formation of "annular flow" breaks down, and the refrigerant film is disrupted. When flowing thus into the next hairpin tube 11, the refrigerant film does so in a disrupted state, whereby portions of the refrigerant film thicken at the straight-tube portion of the hairpin tube 11, destabilizing thus heat exchange with the exterior of the tube and eventually impairing evaporative performance.

[0052] If the first fin root radius (r_1) is greater than 1/2 of the first groove depth (h_1), the reduced cross-sectional area of the first grooves 2 is likely to give rise to contracted flow of the refrigerant film at the return bend tube inlet side during inflow of refrigerant from the hairpin tube 11 into the return bend tube 1, thereby disrupting the refrigerant film. When flowing thus into the next hairpin tube 11, the refrigerant film does so in a disrupted state, whereby portions of the refrigerant film thicken at the straight-tube portion of the hairpin tube 11, destabilizing thus heat exchange with the exterior of the tube and eventually impairing evaporative performance.

[0053] As illustrated in Fig. 1, the evaporative performance of the heat exchanger into which the return bend tube 1 is built can be enhanced by restricting the tube body 1a of the return bend tube 1 as described below.

<Tube body>

[0054] (Length (L): 1.0 to 1.5 times the pitch (P))

The length (L) of the return bend tube 1 (tube body 1a) measures preferably 1.0 to 1.5 times the pitch (P) thereof. Herein, the length (L) is the distance between the tube end 1b and the outer face of the bending apex of the U-shaped tube body 1a. The pitch (P) is the distance between the centers of both tube ends of the U-shaped tube body 1a.

[0055] If the Length (L) is smaller than 1.0 times the bending pitch (P), the resulting shorter length from the entrance of the return bend tube to the point where bending starts precludes sufficient formation of "annular flow" and gives rise to splashing of the refrigerant film on the inner side of the bending portion, which disrupts the refrigerant film (separated flow). When flowing thus into the next hairpin tube, the refrigerant film does so in a disrupted state, whereby portions of the refrigerant film thicken at the straight-tube portion of the hairpin tube, destabilizing thus heat exchange with the exterior of the tube and eventually impairing evaporative performance.

[0056] If the length (L) is greater than 1.5 times the bending pitch (P), the resulting longer length from the inlet side of the return bend tube to the point where bending starts facilitates formation of "annular flow", whereas it increases pressure loss of the flowing refrigerant in return bend tube 1, whereby evaporative performance may be impaired.

(Materials)

[0057] The return bend tube 1 (tube body 1a) comprises preferably a material having a lower thermal conductivity than the material of the hairpin tube. When the return bend tube 1 is used in a heat exchanger 20 (Figs. 2 and 3), in particular in an air heat exchanger, the return bend tube 1 is used outside the heat exchange portion. When the material of the return bend tube 1 has a higher thermal conductivity than the material of the hairpin tube, therefore, there occurs heat loss at the portion of the return bend tube 1. When heat loss occurs at the portion of the return bend tube 1, refrigerant evaporates at the portion of the return bend tube 1, as a result of which formation of the "annular flow" of the refrigerant film collapses and the refrigerant film splashes around, thereby disrupting the refrigerant film (separated flow). When flowing thus into the next hairpin tube, the refrigerant film does so in a disrupted state, whereby portions of the refrigerant film thicken at the straight-tube portion of the hairpin tube, destabilizing thus heat exchange with the exterior of the tube and eventually impairing evaporative performance.

[0058] Phosphorus deoxidized copper has been often used conventionally as the material of the hairpin tube and of the return bend tube 1 (tube body 1a), with brazing as the method employed for connecting the tubes. During brazing, the tube ends of both tubes are heated to about 800 to 900°C by means of a gas burner or the like. When phosphorus deoxidized copper is used in the return bend tube 1 (tube body 1a), such brazing heat lowers the strength of the return bend tube 1 (heat-affected portion), and breaking of the tube tends to occur due to the internal pressure of the tube in use of the heat exchanger. To avoid the breaking of the tube, a first tube wall thickness (T1) (Fig. 4) of the return bend tube 1 (tube body 1a) must be made thicker. This strength loss caused by heating can be avoided, however, by making the return bend tube 1 (tube body 1a) with a heat resistant copper alloy having a greater heat resistance than the hairpin tube. This allows also further enhancement of compression strength while avoiding wall thickening. The return bend tube 1 (tube body 1a) can be made more lightweight as a result. Preferred heat-resistant copper alloys include, for instance, Cu-Sn-P alloys, Cu-Sn-Zn-P alloys and the like, having a compression strength of 10 MPa or more at room temperature even after heating at 850°C. A heat-resistant copper alloy identical to that of the return bend tube 1 may be used also in the hairpin tube.

(First maximum inner diameter (ID1))

[0059] As illustrated in Figs. 5 and 6, the first maximum inner diameter (ID1) of the return bend tube 1 (tube body 1a) and the second maximum inner diameter (ID2) of the hairpin tube 11 satisfy the relationship $ID1 \geq ID2$. If $ID1 < ID2$, "annular flow" of the refrigerant film formed inside the return bend tube 1 becomes spreaded flow of the refrigerant film of the inlet portion of the hair pin tube 11, and the thickness of the refrigerant film becomes uneven, which disrupts the refrigerant film. Thus, the refrigerant film flows in a disrupted state in the vicinity of the inlet of the next hairpin tube, whereby part of the refrigerant film thickens, destabilizing thus heat exchange with the exterior of the tube and eventually impairing evaporative performance.

(2) Hairpin tube

[0060] Next are explained the hairpin tubes 11 that, as illustrated in Figs. 2 and 3, make up the heat exchanger 20 together with the return bend tubes 1 according to the present invention. As illustrated in Fig. 6, the hairpin tube 11 has the plurality of spiral second grooves 12 inside the tube, wherein the inner groove shape of the second grooves 12 is preferably restricted as described below. In heat transfer tubes used in air-conditioners, 3 to 10 mm tubes are ordinarily

used, and hence tubes having an outer diameter (second outer diameter OD2) ranging from 3 to 10 mm are preferably used as the hairpin tubes 11. Owing to its excellent formability, phosphorus deoxidized copper is preferably used as the material of the hairpin tubes 11. A heat-resistant copper alloy, which has better heat resistance than phosphorus deoxidized copper, may also be used herein.

(Second groove pitch (P2), second groove cross-sectional area (S2))

[0061] Preferably, the second groove pitch (P2) ranges from 0.37 to 0.42 mm and the second groove cross-sectional area (S2) from 0.04 to 0.06 mm². When the second groove pitch (P2) is smaller than 0.37 mm and the second groove cross-sectional area (S2) smaller than 0.04 mm², the fluidity of the tube material into the groove portions of the groove forming tool (for instance, a grooved plug) decreases during formation of the second grooves 12 on the tube inner surface, which entails a greater press force from the exterior of the tube. As a result, the grooving tool becomes prone to break, while the second grooves 12 become harder to be shaped stably on the tube inner surface. When the second groove pitch (P2) exceeds 0.42 mm and the second groove cross-sectional area (S2) exceeds 0.06 mm², the liquid refrigerant film is hard to form the thin layer in the second grooves 12 inside the tube. As a result, the refrigerant film inside the tube turns resistance to heat exchange with exterior of the tube, and evaporative performance is eventually impaired.

(Second groove lead angle (θ_2): Fig. 4)

[0062] Preferably, the second groove lead angle (θ_2) is 15° or more. When the second groove lead angle (θ_2) is smaller than 15°, formation of "swirling flow" by the refrigerant film inside the tube is insufficient, which is likely to impair evaporative performance. During inflow of liquid refrigerant from the return bend tube outlet side into the next hairpin tube 11, lack of the second groove lead angle reduces formation of homogeneous "annular flow" of the refrigerant film on the second grooves 12, so that the refrigerant film becomes uneven at the straight-tube portion of the hairpin tube 11, destabilizing thus heat exchange with the exterior of the tube and eventually impairing evaporative performance. When the second groove lead angle (θ_2) exceeds 45°, the rolling speed of formation of the second grooves 12 on the tube inner side tends to decrease sharply, which makes it more difficult to manufacture stably a long hairpin tube 11. Accordingly, the second groove lead angle (θ_2) is preferably of 45° or less.

(Second groove depth (h2))

[0063] The second groove depth (h2) ranges preferably from 0.10 to 0.28 mm. When the second groove depth (h2) is smaller than 0.10 mm, the second fins 13 formed between the second grooves 12 on the tube inner side drop below the level of the refrigerant inside the tube, and hence the fins become buried by the refrigerant film. The effective heat transfer area inside the tube decreases dramatically as a result, and evaporative performance is impaired. When the second groove depth (h2) is greater than 0.28 mm, the groove forming tool (for instance, a grooved plug) becomes prone to break during formation of the second grooves 12 on the tube inner surface, and the second grooves 12 become harder to be shaped stably on the tube inner surface.

(Second fin apex angle (δ_2))

[0064] The second fin apex angle (δ_2) ranges preferably from 5 to 45°. When the second fin apex angle (δ_2) is smaller than 5°, the second fins 13 are likelier to collapse or break during mechanical tube expansion (not shown in the figure) to incorporate the hairpin tubes 11 into a heat exchanger 20 for air-conditioners. Also, the groove forming tool becomes prone to get chipped during shaping on the second grooves 12 and the second fins 13 on the tube inner surface, so that the second grooves 12 become harder to shape stably on the tube inner surface. When the second fin apex angle (δ_2) exceeds 45°, the cross-sectional area of the second grooves 12 shrinks dramatically, thereby impairing heat-transfer performance. Also, the cross-sectional area of the second fins 13 (second wall thickness (T2) of the hairpin tube 11) increases, thereby increasing the weight of the hairpin tube 11 and making it harder to build a light-weight heat exchanger 20.

(Second fin root radius (r2))

[0065] Preferably, the second fin root radius (r2) ranges from 1/10 to 1/3 of the second groove depth (h2). When the second fin root radius (r2) is smaller than 1/10 of the second groove depth (h2) and the second fins 13 are high, formability of the second fins 13 (second grooves 12) worsens, making it more difficult to achieve second fins 13 of a predetermined shape, and increasing the likelihood of damage in the groove forming tool that abuts the root of the second grooves 12

on the tube inner surface. When the second fin root radius (r_2) is larger than $1/3$ of the second groove depth (h_2), the cross-sectional area of the second fins 13 increases, the second wall thickness (T_2) of the hairpin tube 11 increases, and the hairpin tube 11 becomes heavier.

5 (Second maximum inner diameter (ID2))

10 **[0066]** The second maximum inner diameter (ID2) of the hairpin tube 11 is preferably 0.80 to 0.96 of the outer diameter (OD2) of the hairpin tube 11. When the second maximum inner diameter (ID2) is smaller than 0.80 of the outer diameter (OD2) of the hairpin tube 11, the second wall thickness (T_2) becomes thicker, thereby increasing the weight of the hairpin tube 11 and making it harder to build a light-weight heat exchanger 20 (Figs. 2 and 3). When the second maximum inner diameter (ID2) exceeds 0.96 of the outer diameter (OD2) of the hairpin tube 11, the second wall thickness (T_2) becomes thinner, thereby reducing the tube strength of the hairpin tube 11 and increasing the likelihood of tube breakage in use of the heat exchanger 20.

15 (3) Method for manufacturing the return bend tube and the hairpin tube

[0067] A method for manufacturing the return bend tube and the hairpin tube is explained next. The return bend tube and the hairpin tube are manufactured, for instance, in accordance with the following conventional manufacturing method. A soft material is ordinarily used as the tube stock employed in the below-described first step. The below-described first through third steps are carried out sequentially using tube rolling machine provided with a diameter-reducing apparatus at a preliminary state and a final stage. After the third diameter-reducing process of the third step, the inner surface grooved tube is wound as a level wound coil, is annealed into a soft material in an annealing furnace, and is used in a fourth step to manufacture a return bend tube and a hairpin tube.

25 (First step)

[0068] Tube stock made of a base material such as phosphorus deoxidized copper or a heat-resistant copper alloy is drawn by passing between a diameter-reducing die and a diameter-reducing plug, to subject thereby the tube stock to a first diameter-reducing process.

30 (Second step)

[0069] A grooved plug is inserted into the tube stock that was reduced in the first step, and then outer surface of the tube stock is rolled at the portion inside which the grooved plug is located by a plurality of rolling balls or rolling rolls, to subject thereby the tube stock to a second diameter-reducing process. Simultaneously therewith, the groove shape of the grooved plug is transferred to the inner surface of the reduced tube stock, to form thereby the first grooves 2 or the second grooves 12 (Fig. 4). The grooved plug has herein a groove shape that corresponds to the above-described inner surface groove shapes (Figs. 5 and 6).

40 (Third step)

[0070] The tube stock, onto the inner surface of which the first grooves 2 or the second grooves 12 have been formed in the second step, is then drawn using a forming die, to carry out a third diameter-reducing step and manufacture thereby an inner-surface grooved heat transfer tube having a first outer diameter (OD1) or a second outer diameter (OD2).

45 (Fourth step)

[0071] The inner-surface grooved tube manufactured in the third step is then bent using a predetermined jig, to manufacture thereby a return bend tube 1 and a hairpin tube 11 having a predetermined shape (Figs. 1 and 2).

50 (4) Fin-and-tube heat exchanger

[0072] The heat exchanger of the present invention is explained next. As illustrated in Figs. 2 and Figs. 3(a), 3(b) and 3(c), the heat exchanger 20, wherein refrigerant is supplied through tubing, comprises a hairpin tube portion 23, in which a plurality hairpin tubes 11, 11... are arranged at a predetermined bending pitch P_a ; a return bend tube portion 22 having a plurality of return bend tubes 1, 1... joined by tube ends 1b, 1b (Fig. 1) to the tube end portions of respective hairpin tubes 11, 11... of the hairpin tube portion 23; and a fin portion 21 comprising a plurality of fins 21a, 21a ... arranged at a predetermined spacing (fin pitch P_b) on the outer surface of the hairpin tubes 11. Thanks to such a constitution, the

plurality of hairpin tubes 11, 11... are coupled in series over multiple stages via the return bend tubes 1, 1..., and thus the heat exchanger 20 has a long effective heat-transfer tube length (refrigerant flow channel). As illustrated in Fig. 3 (b), the hairpin tubes 11 may also be arranged in a plurality of columns with a predetermined column-direction pitch P_c . As illustrated in Fig. 3(c), the refrigerant supplied inside the tubes of the heat exchanger 20 flows in the same direction as that of the flow of the air with which the heat exchanger 20 is blown, during refrigerant condensation, and in the reverse direction, during refrigerant evaporation.

[0073] At least part of the return bend tube portion 22 comprises the return bend tube 1 on the inner surface of which there are formed the above-described plurality of first grooves 2 (Fig. 5). Such a constitution allows reducing evaporative performance loss by the heat exchanger 20. The inner-surface groove shape of the return bend tube 1, for instance, the groove pitch ratio ($P1/P2$), the groove cross-sectional area ratio ($S1/S2$), the groove depth ratio ($h1/h2$) (Figs. 5 and 6), the angle difference between groove lead angles ($\theta1-\theta2$) (Fig. 4), or the first maximum inner diameter ($ID1$), may vary depending on the location of the return bend tube portion 22, in consideration of the flow of refrigerant (upstream, downstream) in the heat exchanger 20. On account of refrigerant pressure loss, inner-surface smooth return bend tubes may also be used in at least part of the return bend tube portion 22.

[0074] In the heat exchanger of the present invention, at least one part of the refrigerant flow channel constituted by the hairpin tubes and the return bend tubes may be branched, forming thus a plurality of refrigerant flow channels. As illustrated in Figs. 7(a) and 7(b), for instance, the heat exchanger of the present invention may be a two-pass heat exchanger 20A where the refrigerant flow channel as a whole is branched, and a partial two-pass heat exchanger 20B in which part of the refrigerant flow channel is branched. Although in Fig. 7 (a) and 7 (b) the refrigerant flow channel is branched into two flow channels (refrigerant flow channel A and refrigerant flow channel B), branching is not limited thereto, and the refrigerant may be branched into three or more flow channels. Also, a branched refrigerant flow channel (refrigerant flow channel A and refrigerant flow channel B) may in turn be branched into the plurality of refrigerant flow channels. In the partial two-pass heat exchanger 20B of Fig. 7(b) there is one branching location, but there may be two or more such locations. That is, the one-pass heat exchanger 20 having no branched refrigerant flow channel, as illustrated in Fig. 3(c), may be joined to the plurality of two-pass heat exchangers 20A.

[0075] As in the above one-pass heat exchanger 20 (Fig. 3(c)), maintaining the swirling flow of the refrigerant enhances evaporative performance also in the heat exchangers 20A (two-pass heat exchanger) and 20B (partial two-pass heat exchanger) illustrated in Fig. 7. In the heat exchangers 20A and 20B, where the refrigerant flow channel is branched, the refrigerant mass velocity per branching decreases, and in particular the refrigerant velocity decreases at the return bend tube inlet side, which stabilizes further the "annular flow" of the refrigerant film formed inside the tubes. During inflow of liquid refrigerant from the return bend tube outlet side into the next hairpin tube, there forms a more homogeneous "annular flow", so that the refrigerant film becomes uniform at the straight-tube portion of the hairpin tube, stabilizing thus heat exchange with the exterior of the tube (atmosphere) and further enhancing evaporative performance. Also, forming the plurality of refrigerant flow channels (refrigerant flow channel A and refrigerant flow channel B) has the effect of reducing the number of hairpin tubes and return bend tubes constituting one refrigerant flow channel (refrigerant flow channel A or refrigerant flow channel B) compared with number in the above-described one-pass heat exchanger 20 (from 11 stages to 6 stages in Fig. 3(c) and Fig. 7).

As a result, this reduces refrigerant pressure loss and further enhances evaporative performance.

[0076] The refrigerant used in the heat exchanger 20 of the present invention is a hydrofluorocarbon (HFC) refrigerant, preferably, for instance, of R410 type, and more preferably R410A, which is a 50/50% mixture of difluoroethane (R32) and pentafluoroethane (R125). Using a non-azeotropic HFC mixed refrigerant has the effect of increasing the evaporative performance of the heat exchanger 20 and of reducing refrigerant pressure loss. Although R410 refrigerants have excellent evaporative performance, they also have a high working pressure, which tends to result in large compressors. Thus an R407 type, having a slightly lower evaporative performance but also a lower working pressure than R410 type, may be used as the refrigerant of the present invention.

EXAMPLES

<Examples 1 to 20 (excluding Example 9)>

[0077] Examples of the present invention are explained in detail next.

Firstly, phosphorus deoxidized copper having an alloy number C1220 or oxygen-free copper having an alloy number C1020, as per JISH3300, was melted, cast, hot-extruded, cold-rolled and cold-drawn to yield a tube stock in Examples 1 to 6 and 8 to 20, while a Cu-Sn-P (0.65wt%, 0.03wt%, balance Cu) heat-resistant alloy was similarly processed to yield a tube stock in Example 7. After subsequent annealing, the tube stock was subjected to a first diameter-reducing process, then the reduced tube stock was subjected to a second diameter-reducing process while forming thereon spiral grooves (or parallel grooves) as inner-surface groove shapes given in Table 1 and Table 2. The grooved tube stock was then subjected to a third diameter-reducing process and was annealed to manufacture thereby a test tube (for return

bend tubes) having a first outer diameter (OD1) of 7 mm. Test tubes (for hairpin tubing) having a second outer diameter (OD2) of 7 mm were manufactured in accordance with the same manufacturing method, using herein a phosphorus deoxidized copper having an alloy number C1220 as per JISH3300.

[0078] A fin-and-tube heat exchanger (one-pass heat exchanger) 20 as illustrated in Fig. 2 and Figs. 3(a) and 3(b) was manufactured then using the respective test tubes. The test tubes (for hairpin tubes) were first bent, by the middle portion thereof, into a hairpin shape with a predetermined bending pitch (Pa), to manufacture a plurality of hairpin tubes 11. The plurality of hairpin tubes 11 were then passed through the plurality of fins 21a arranged parallel to one another at a predetermined spacing (fin pitch (Pb)). A bullet for yielding an expansion rate of 105.5% with respect to the outer diameter of the a copper tube (hairpin tube 11) was the inserted into the hairpin tubes 11, then the tubes were expanded using a shrinkage-type tube expander, and the hairpin tubes 11 were joined to the fins 21a. The test tubes (for return bend tubes) were then bent to a predetermined length L and pitch (P) (Fig. 1), to manufacture the plurality of return bend tubes 1. To manufacture the heat exchanger 20, as illustrated in Fig. 4, the tube ends of the adjacent hairpin tube 11 were further expanded, the return bend tubes 1 provided with a ring of phosphorus copper brazing alloy (BCuP-2) were fitted to the ends of the hairpin tube 11, and then both tubes were heat-brazed together (850°C, 1 minute) using a burner, while nitrogen gas was streamed through the interior of the tubes to prevent oxidation. The specifications of the heat exchanger 20 were as follows.

(Heat exchanger 20)

[0079] Outer dimensions: length 500 mm x height 250 mm x width 25.4 mm.

(Hairpin tubes 11)

[0080] Arranged in 2 columns, 12 stages (bending pitch (Pa) 21 mm, column-direction pitch (Pc) 13.4 mm (length (La) prior to tube expansion about 535 mm).

(Return bend tube 1)

[0081] Length (L) = 20.0 mm, 21.2 mm, 22.5 mm, 31.4 mm, 33.0 mm

Pitch (P) = 21.0 mm (Fig. 1).

(Fins 21a)

For the fins 21a there was used a plate material comprising aluminum of alloy number 1N30 according to JIS H4000, the surface of the plate material being covered with resin. The thickness of the fins 21a was 110 μm. There were 410 fins 21a arranged in parallel with a fin pitch (Pb) of 1.25 mm.

[0082] The same test tubes (hairpin tube, return bend tube) as in Example 1 were used in Example 9, and a fin-and-tube heat exchanger (two-pass heat exchanger) 20A such as the one illustrated in Fig. 7 (a) was manufactured in the same way as in Example 1. Herein the hairpin tubes 11 of refrigerant flow channels A and B comprised 2 columns and 6 stages.

<Comparative examples 1 to 5>

[0083] As illustrated in Table 3, Comparative example 1 was identical to Example 1 except that a smooth tube, without grooves formed on the inner surface, was used herein as the test tube (return bend tube). Comparative examples 2 to 5 were identical to Example 1 except that herein there were used inner surface grooved tubes in which the groove pitch ratio (P1/P2) and/or the groove cross-sectional area ratio (S1/S2) lay outside the ranges in the claims of the present invention. A heat exchanger (one-pass heat exchanger) 20 was manufactured in the same way as in Example 1.

[0084] The evaporative performance of the heat exchangers of Examples 1 to 20 and Comparative examples 1 to 5 was measured in accordance with JIS C 9612. The results are given in Table 1, Table 2 and Table 3. Evaporative performance is based on measured heat-transfer rates and is expressed as a ratio relative to Comparative example 1, which is taken as 1.

[0085] Fig. 8(a) is a schematic view illustrating a measurement apparatus for manufacturing evaporative performance. As illustrated in Fig. 8(a), the measurement apparatus comprises a suction-type wind tunnel 100 having a thermo-hygrostatic function, a refrigerant supply apparatus 110 (Fig. 8(b)), and an air-conditioner (not shown). In the suction-type wind tunnel 100, a heat exchanger 20 (20A) is arranged in the flow path of air that flows in through an air flow inlet 108 and is discharged through an air discharge outlet 109, with air samplers 101, 102 arranged respectively upstream and downstream of the heat exchanger 20 (20A). The air samplers 101, 102 are coupled to respective thermohygrometer boxes 103, 104. The thermohygrometer boxes 103, 104 measure the dry-bulb temperature and the wet-bulb temperature of air sampled by the air samplers 101, 102, to measure the temperature and the humidity of the air. An induced draft

fan 105 for discharging air to the air discharge outlet 109 is arranged downstream of the air sampler 102. Flow regulators 106, 106 for adjusting the airflow passing through the heat exchanger 20(20A) are provided between the heat exchanger 20(20A) and the air sampler 102, and between the air sampler 102 and the induced draft fan 105.

[0086] Fig. 8(a) illustrates a schematic view of the refrigerant supply apparatus 110. In Fig. 8(b), the reference numeral 107 denotes refrigerant piping, 111 a sight glass, 112 a heat exchanger for heating and cooling a liquid (refrigerant), 113 a dryer, 114 a liquid (refrigerant) receiver, 115 a fusible plug, 116 a condenser, 117 an oil separator, 118 a compressor, 119 an accumulator, 120 an evaporator, 121 an expansion valve and 122 a flow meter. Pressure and temperature-adjusted refrigerant is supplied via the refrigerant piping 107 to the hairpin tubes 11 (Fig. 2) of the heat exchanger 20 (20A) provided in the suction-type wind tunnel 100. Pressure gauges 123 for measuring the temperature and the pressure of the refrigerant (the temperature is taken as the measured pressure-equivalent saturation temperature) are provided also at the inlet and the outlet of the heat exchanger 20(20A). The air-conditioner (not shown) supplies air of controlled temperature and humidity to the air flow inlet 108 of the suction-type wind tunnel 100.

[0087] The measurement conditions were as follows:

<Refrigerant> R22, R410A

<Air side> Dry-bulb temperature 27.0°C, wet-bulb temperature 19.0°C

Face wind velocity of the heat exchanger 0.8 m/s

<Refrigerant side> Evaporation temperature (with respect to outlet) 7.5°C, inlet dryness 0.2°C, outlet superheating 5.0°C.

[0088]

[Table 1]

		Units	Example 1	Example 2	Example 3	Example 4	Example 5	Example 6	Example 7	Example 8	Example 9	Example 10	Example 11
Hairpin tube	Second outer diameter (OD2)	mm	7	7	7	7	7	7	7	7	7	7	7
	Second wall thickness (T2)	mm	0.24	0.24	0.24	0.24	0.24	0.24	0.24	0.24	0.24	0.24	0.24
	Second maximum inner diameter (ID2)	mm	6.52	6.52	6.52	6.52	6.52	6.52	6.52	6.52	6.52	6.52	6.52
	Groove direction	-	Left- hand spiral	Left- hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral
	Second groove lead angle (θ_2)	°	18	18	18	18	18	15	18	18	18	18	18
	Second groove depth (h2)	mm	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15
	Second fin apex angle (δ_2)	°	40	40	40	40	40	40	40	40	40	40	40
	Second fin root radius (r2)	mm	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03
	Groove count	Grooves	50	50	50	50	50	50	50	50	50	50	50

(continued)													
		Units	Example 1	Example 2	Example 3	Example 4	Example 5	Example 6	Example 7	Example 8	Example 9	Example 10	Example 11
Return bend tube	Second groove pitch (P2)	mm	0.410	0.410	0.410	0.410	0.410	0.410	0.410	0.410	0.410	0.410	0.410
	Second groove cross- sectional area (S2)	mm ²	0.0428	0.0428	0.0428	0.0428	0.0428	0.0428	0.0428	0.0428	0.0428	0.0428	0.0428
	Material	-	C1220	C1220	C1220	C1220	C1220	C1220	C1220	C1220	C1220	C1220	C1220
	Thermal conductivity	W/(m·K)	339	339	339	339	339	339	339	339	339	339	339
	First outer diameter (OD1)	mm	7	7	7	7	7	7	7	7	7	7	7
	First wall thickness	mm	0.24	0.24	0.24	0.24	0.24	0.24	0.24	0.18	0.24	0.24	0.24
	First maximum inner diameter (ID1)	mm	6.52	6.52	6.52	6.52	6.52	6.52	6.52	6.64	6.52	6.52	6.52
	Groove direction	-	Left- hand spiral	Left- hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral	-	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral
	First groove lead angle (θ ₁) lead	°	18	18	18	18	18	0	18	18	18	18	18
	First groove depth (h1)	mm	0.15	0.15	0.1	0.15	0.21	0.15	0.15	0.15	0.15	0.15	0.15

(continued)													
		Units	Example 1	Example 2	Example 3	Example 4	Example 5	Example 6	Example 7	Example 8	Example 9	Example 10	Example 11
	First fin apex angle (δ_1)	°	40	40	40	40	40	40	40	40	40	40	40
	First fin root radius (r1)	mm	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03
	Groove count	Grooves	50	75	75	23	23	50	50	50	50	50	50
	First groove pitch (p1)	mm	0.410	0.273	0.273	0.891	0.891	0.410	0.410	0.417	0.410	0.410	0.410
	First groove cross-sectional area (S1)	mm ²	0.0428	0.0228	0.0173	0.1133	0.1522	0.0428	0.0428	0.044	0.0428	0.0428	0.0428
	Bending pitch (P)	mm	21	21	21	21	21	21	21	21	21	21	21
	Length (L)	mm	22.5	22.5	22.5	22.5	22.5	22.5	22.5	22.5	22.5	21.2	31.4
	Material	-	C1220	C1220	C1220	C1220	C1220	C1220	Cu-Sn-P	C1220	C1220	C1220	C1220
	Thermal conductivity	W/(m·K)	339	339	339	339	339	339	227	339	339	339	339
Heat exchanger structure	Coolantpass count	Pass	1	1	1	1	1	1	1	1	2	1	1
	Angle difference ($\theta_1 - \theta_2$)	°	0	0	0	0	0	-15	0	0	0	0	0
	Groove pitch ratio (P1/P2)	-	1.0000	0.6667	0.6667	2.1739	2.1739	1.0000	1.0000	1.0184	1.0000	1.0000	1.0000

(continued)

		Units	Example 1	Example 2	Example 3	Example 4	Example 5	Example 6	Example 7	Example 8	Example 9	Example 10	Example 11
	Groove cross- sectional area ratio (S1/S2)	-	1.0000	0.5327	0.4042	2.6472	3.5561	1.0000	1.0000	1.0280	1.0000	1.0000	1.0000
	(ID1/ID2)	-	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0184	1.0000	1.0000	1.0000
	Groove depth ratio (h1/h2)	-	1.0000	1.0000	0.6667	1.0000	1.4000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
	(L/P)	-	1.0714	1.0714	1.0714	1.0714	1.0714	1.0714	1.0714	1.0714	1.0714	1.0095	1.4952
	Evaporative performance (R22)	-	1.0130	1.0128	1.0120	1.0127	1.0120	1.0132	1.0131	1.0132	1.0133	1.0134	1.0132
	Evaporative performance (R410A)	-	1.0137	1.0132	1.0130	1.0131	1.0131	1.0136	1.0135	1.0136	1.0135	1.0136	1.0135

[0089]

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[Table 2]

Hairpin tube		Units	Example 12	Example 13	Example 14	Example 15	Example 16	Example 17	Example 18	Example 19	Example 20
	Second outer diameter (OD2)	mm	7	7	7	7	7	7	7	7	7
	Second wall thickness (T2)	mm	0.24	0.24	0.24	0.24	0.24	0.24	0.24	0.24	0.24
	Second maximum inner diameter (ID2)	mm	6.52	6.52	6.52	6.52	6.52	6.52	6.52	6.52	6.52
	Groove direction	-	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral
	Second groove lead angle (θ_2)	°	16	18	18	18	18	18	18	18	14
	Second groove depth (h2)	mm	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15
	Second fin apex angle (δ_2)	°	40	40	40	40	40	40	40	40	40
	Second fin root radius (r2)	mm	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03
	Groove count	Grooves	50	50	50	50	50	50	50	50	50
	Second groove pitch (P2)	mm	0.410	0.410	0.410	0.410	0.410	0.410	0.410	0.410	0.410

(continued)											
		Units	Example 12	Example 13	Example 14	Example 15	Example 16	Example 17	Example 18	Example 19	Example 20
	Second groove cross-sectional area (S2)	mm ²	0.0428	0.0428	0.0428	0.0428	0.0428	0.0428	0.0428	0.0428	0.0428
	Material	-	C1220	C1220	C1220	C1220	C1220	C1220	C1220	C1220	C1220
	Thermal conductivity	W/(m·K)	339	339	339	339	339	339	339	339	339
Return bend tube	First outer diameter (OD1)	mm	7	7	7	7	7	7	7	7	7
	First wall thickness (T1)	mm	0.24	0.24	0.24	0.24	0.24	0.24	0.24	0.36	0.24
	First maximum inner diameter (ID1)	mm	6.52	6.52	6.52	6.52	6.52	6.52	6.52	6.28	6.52
	Groove direction	-	-	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral
	First groove lead angle (θ1)	°	0	35	18	18	18	18	18	18	18
	First groove depth (h1)	mm	0.15	0.15	0.07	0.23	0.15	0.15	0.15	0.15	0.15
	First fin apex angle (δ1)	°	40	40	40	40	40	40	40	40	40
	First fin root radius (r1)	mm	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03

(continued)											
		Units	Example 12	Example 13	Example 14	Example 15	Example 16	Example 17	Example 18	Example 19	Example 20
	Groove count	Grooves	50	50	50	50	50	50	50	50	50
	First groove pitch (P1)	mm	0.410	0.410	0.410	0.410	0.410	0.410	0.410	0.395	0.410
	First groove cross-sectional area (S1)	mm ²	0.0428	0.0428	0.0226	0.0577	0.0428	0.0428	0.0428	0.0406	0.0428
	Bending pitch (P)	mm	21	21	21	21	21	21	21	21	21
	Length (L)	mm	22.5	22.5	22.5	22.5	20	33	22.5	22.5	22.5
	Material	-	C1220	C1220	C1220	C1220	C1220	C1220	C1020	C1220	C1220
	Thermal conductivity	W/(m·K)	339	339	339	339	339	339	391	339	339
Heat exchanger structure	Coolant pass count	Pass	1	1	1	1	1	1	1	1	1
	Angle difference (θ1-θ2)	°	-16	17	0	0	0	0	0	0	4
	Groove pitch ratio (P1/P2)	-	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	0.9632	1.0000
	Groove cross-sectional area ratio (S1/S2)	-	1.0000	1.0000	0.5280	1.3481	1.0000	1.0000	1.0000	0.9486	1.0000
	(ID1/ID2)	-	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	0.9632	1.0000

(continued)

		Units	Example 12	Example 13	Example 14	Example 15	Example 16	Example 17	Example 18	Example 19	Example 20
	Groove depth ratio (h1/h2)	-	1.0000	1.0000	0.4667	1.5333	1.0000	1.0000	1.0000	1.0000	1.0000
	(L/P)	-	1.0714	1.0714	1.0714	1.0714	0.9524	1.5714	1.0714	1.0714	1.0714
	Evaporative performance (R22)	-	1.0114	1.0110	1.0109	1.0108	1.0109	1.0108	1.0107	1.0105	1.0103
	Evaporative performance (R410A)	-	1.0116	1.0114	1.0110	1.0109	1.0110	1.0111	1.0110	1.0108	1.0106

[0090]

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[Table 3]

		Units	Comparative example 1	Comparative example 2	Comparative example 3	Comparative example 4	Comparative example 5
Hairpin tube	Second outer diameter (OD2)	mm	7	7	7	7	7
	Second wall thickness (T2)	mm	0.24	0.24	0.24	0.24	0.24
	Second maximum inner diameter (ID2)	mm	6.52	6.52	6.52	6.52	6.52
	Groove direction	-	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral
	Second groove lead angle (θ_2)	°	18	18	18	18	18
	Second groove depth (h2)	mm	0.15	0.15	0.15	0.15	0.15
	Second fin apex angle (δ_2)	°	40	40	40	40	40
	Second fin root radius (r2)	mm	0.03	0.03	0.03	0.03	0.03
	Groove count	Grooves	50	50	50	50	50
	Second groove pitch (P2)	mm	0.410	0.410	0.410	0.410	0.410
	Second groove cross-sectional area (S2)	mm ²	0.0428	0.0428	0.0428	0.0428	0.0428
	Material	-	C1220	C1220	C1220	C1220	C1220
	Thermal conductivity	W/(m·K)	339	339	339	339	339

(continued)							
		Units	Comparative example 1	Comparative example 2	Comparative example 3	Comparative example 4	Comparative example 5
Return bend tube	First outer diameter (OD1)	mm	7	7	7	7	7
	First wall thickness (T1)	mm	0.24	0.24	0.24	0.24	0.24
	First maximum inner diameter (ID1)	mm	6.52	6.52	6.52	6.52	6.52
	Groove direction	-	-	Left-hand spiral	Left-hand spiral	Left-hand spiral	Left-hand spiral
	First groove lead angle ($\theta 1$)	°	-	18	18	18	18
	First groove depth (h1)	mm	-	0.1	0.21	0.15	0.15
	First fin apex angle ($\delta 1$)	°	-	40	40	40	40
	First fin root radius (r1)	mm	-	0.03	0.03	0.03	0.03
	Groove count	Grooves	-	76	22	18	78
	First groove pitch (P1)	mm	-	0.270	0.931	1.138	0.263
	First groove cross-sectional area (S1)	mm ²	-	0.0113	0.1604	0.1329	0.0213
	Bending pitch (P)	mm	21	21	21	21	21
	Length (L)	mm	22.5	22.5	22.5	22.5	22.5
	Material	-	C1220	C1220	C1220	C1220	C1220
	Thermal conductivity	W/(m·K)	339	339	339	339	339

(continued)							
		Units	Comparative example 1	Comparative example 2	Comparative example 3	Comparative example 4	Comparative example 5
Heat exchanger structure	Coolant pass count	Pass	1	1	1	1	1
	Angle difference (θ_1 - θ_2)	°	-	0	0	0	0
	Groove pitch ratio (P1/P2)	-	-	0.6579	2.2727	2.7778	0.6410
	Groove cross- sectional area ratio (S1/S2)	-	-	0.2640	3.7477	3.1051	0.4977
	(ID1/ID2)	-	1.0000	1.0000	1.0000	1.0000	1.0000
	Groove depth ratio (h1/h2)	-	-	0.6667	1.4000	1.0000	1.0000
	(L/P)	-	1.0714	1.0714	1.0714	1.0714	1.0714
	Evaporative performance (R22)	-	1.00000	0.9951	0.9953	0.9954	0.9961
	Evaporative performance (R410A)	-	1.00000	0.9964	0.9961	0.9964	0.9964

[0091] The results of Table 1, Table 2 and Table 3 show that the heat exchangers in Examples 1 to 20 have superior evaporative performance as compared with the heat exchanger in Comparative example 1, in which a smooth tube is used as the return bend tube.

In the heat exchanger of Comparative example 2 the groove cross-sectional area ratio ($S1/S2$) is below the lower limit, in the heat exchanger of Comparative example 3 the groove pitch ratio ($P1/P2$) and the groove cross-sectional area ratio ($S1/S2$) exceed the upper limit, in the heat exchanger of Comparative example 4 the groove pitch ratio ($P1/P2$) exceeds the upper limit, while in the heat exchanger of Comparative example 5 the groove pitch ratio ($P1/P2$) is below the lower limit. As a result, the heat exchangers in Comparative examples 1 to 5 exhibit a poorer evaporative performance than the heat exchangers in Examples 1 to 20.

<Examples 21 and 22>

[0092] As indicated in Table 4, Example 21 was identical to Example 1 except that herein an inner surface grooved tube having a first wall thickness ($T1$) of 0.20 mm and comprising a Cu-Sn-P material (heat-resistant alloy of 0.65wt% Sn, 0.03wt% P, balance Cu), was used as the test tube (return bend tube).

Example 22 was identical to Example 1 except that herein an inner surface grooved tube having a first wall thickness ($T1$) of 0.34 mm was used as the test tube (return bend tube). A heat exchanger (one-pass heat exchanger) was manufactured in the same way as in Example 1. The heat exchangers of Example 1, Example 21 and Example 22 were subjected to a pressure resistance test by water pressure. The pressure at which the return bend tube portion (return bend tube) of the heat exchanger ruptures, i.e. the compression strength, was measured using a Bourdon tube pressure gauge. The results are given in Table 4.

[0093]

[Table 4]

	Return bend tube	Hairpin tube	compression strength
Example 1	Material: C1220 Outer diameter (OD1) : 7.00mm First wall thickness ($T1$) : 0.24mm Other groove shapes: Same as Table 1	Material: C1220 Outer diameter (OD2): 7.00mm Second wall thickness ($T2$) : 0.24mm Other groove shapes: Same as Table 1	13.0 MPa
Example 21	Material: Cu-Sn-P Outer diameter (OD1) : 7.00mm First wall thickness($T1$) : 0.20mm Other groove shapes: Same as Example 1	Material: C1220 Outer diameter (OD2) : 7.00mm Second wall thickness ($T2$) : 0.24mm Other groove shapes: Same as Example 1	13.5 MPa
Example 22	Material: C1220 Outer diameter (OD1) : 7.00mm First wall thickness ($T1$) : 0.34mm Other groove shapes: Same as Example 1	Material: C1220 Outer diameter (OD2) : 7.00mm Second wall thickness ($T2$) : 0.24mm Other groove shapes: Same as Example 1	13.5 MPa

[0094] The results of Table 4 show that the heat exchanger of Example 21 has higher compression strength than that of Example 1, thanks to a smaller loss of strength through brazing, even though the first wall thickness ($T1$) of the return bend tube was thinner than that of Example 1. The heat exchanger of Example 22, where the material of the return bend tube was the same as that of Example 1, exhibited compression strength similar to that of Example 21, but with a first wall thickness ($T1$) of the return bend tube 1.7 times thicker than that of Example 1, which implied an increased material usage.

Claims

1. A fin-and-tube heat exchanger in which a refrigerant is supplied inside tubing and which comprises: a hairpin tube portion where a plurality of hairpin tubes are arranged; a return bend tube portion where there are arranged a plurality of return bend tubes joined to respective hairpin tube ends of said hairpin tube portion; and a fin portion comprising

a plurality of fins arranged at a predetermined spacing on the outer surface of said hairpin tubes, the fin-and-tube heat exchanger further comprising:

first grooves formed on a tube inner surface of said return bend tube,

wherein a first groove pitch (P1) of said first grooves in a cross section perpendicular to a tube axis, and a second groove pitch (P2) of spiral-shaped second grooves formed on the inner surface of said hairpin tube in a cross section perpendicular to a tube axis, satisfy a groove pitch ratio (P1/P2) of 0.65 to 2.2, and wherein a first groove cross-sectional area (S1) per groove of said first grooves in a cross section perpendicular to the tube axis, and a second groove cross-sectional area (S2) per groove of said second grooves in a cross section perpendicular to the tube axis satisfy a groove cross-sectional area ratio (S1/S2) of 0.3 to 3.6.

2. The fin-and-tube heat exchanger according to claim 1, wherein a second groove lead angle (θ_2) formed between the tube axis and the second grooves of said hairpin tube is 15° or more.

3. The fin-and-tube heat exchanger according to claim 1, wherein a refrigerant flow channel comprising said hairpin tube and said return bend tube is at least partially branched, forming a plurality of refrigerant flow channels.

4. The fin-and-tube heat exchanger according to claim 1, wherein said refrigerant is a hydrofluorocarbon-type non-azeotropic mixed refrigerant.

5. A return bend tube, which is used in a fin-and-tube heat exchanger where a refrigerant is supplied inside tubing, and is joined to the tube end of a hairpin tube comprising a plurality of fins arranged at a predetermined spacing on the outer surface thereof, comprising:

first grooves formed on a tube inner surface of said return bend tube,

wherein a first groove pitch (P1) of said first grooves in a cross section perpendicular to a tube axis, and a second groove pitch (P2) of spiral-shaped second grooves formed on the inner surface of said hairpin tube in a cross section perpendicular to a tube axis, satisfy a groove pitch ratio (P1/P2) of 0.65 to 2.2, and wherein a first groove cross-sectional area (S1) per groove of said first grooves in a cross section perpendicular to the tube axis, and a second groove cross-sectional area (S2) per groove of said second grooves in a cross section perpendicular to the tube axis satisfy a groove cross-sectional area ratio (S1/S2) of 0.3 to 3.6.

6. The return bend tube according to claim 5, wherein a first groove lead angle (θ_1) formed between the tube axis and said first grooves and a second groove lead angle (θ_2) formed between the tube axis and said second grooves satisfy an angle difference ($\theta_1 - \theta_2$) of -15 to +15°, and wherein a first groove depth (h1) of said first grooves in a cross section perpendicular to the tube axis, and a second groove depth (h2) of said second grooves in a cross section perpendicular to the tube axis, satisfy a groove depth ratio (h1/h2) of 0.47 to 1.5.

7. The return bend tube according to claim 5, wherein a length (L) of said return bend tube is 1.0 to 1.5 times a pitch (P).

8. The return bend tube according to claim 5, wherein a material of said return bend tube comprises a material having a lower thermal conductivity than a material of said hairpin tube.

9. The return bend tube according to claim 5, wherein a material of said return bend tube comprises a copper alloy more heat resistant than a material of said hairpin tube.

10. The return bend tube according to claim 5, wherein a relationship between a first maximum inner diameter (ID1) of said return bend tube and a second maximum inner diameter (ID2) of said hairpin tube is (ID1) $\geq$ (ID2).

Fig. 1

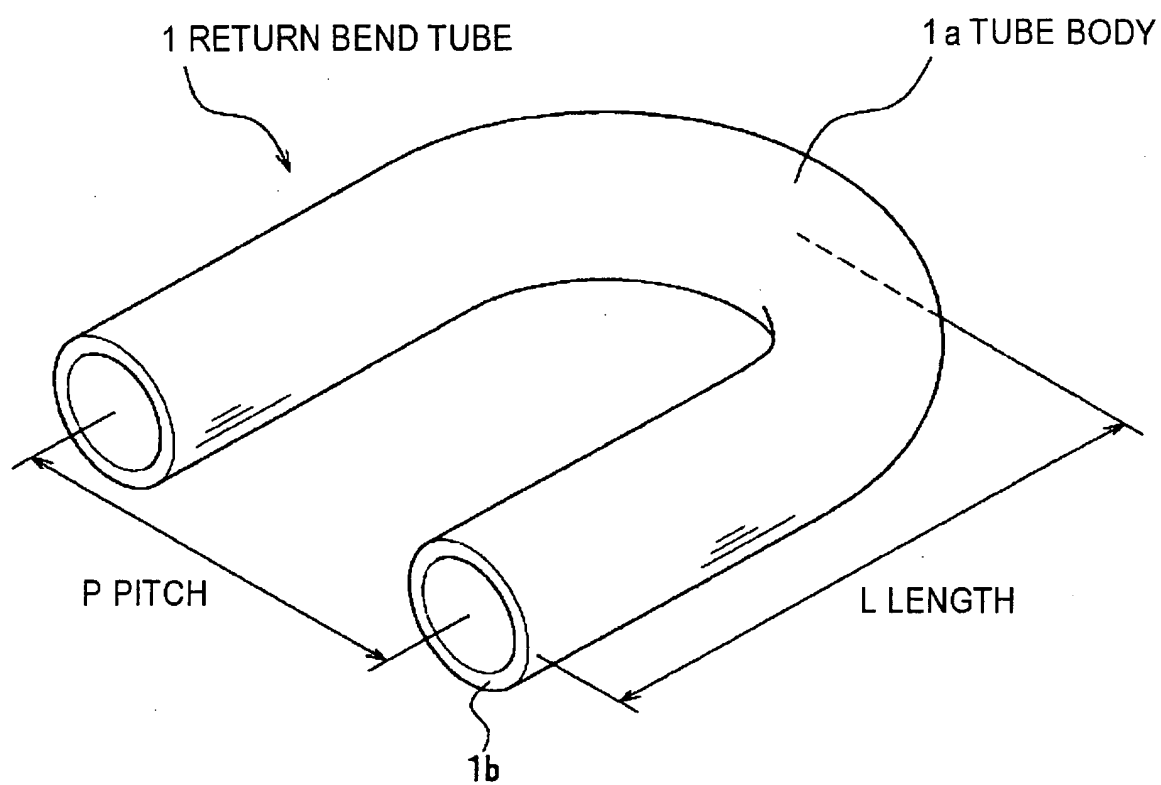


Fig.2

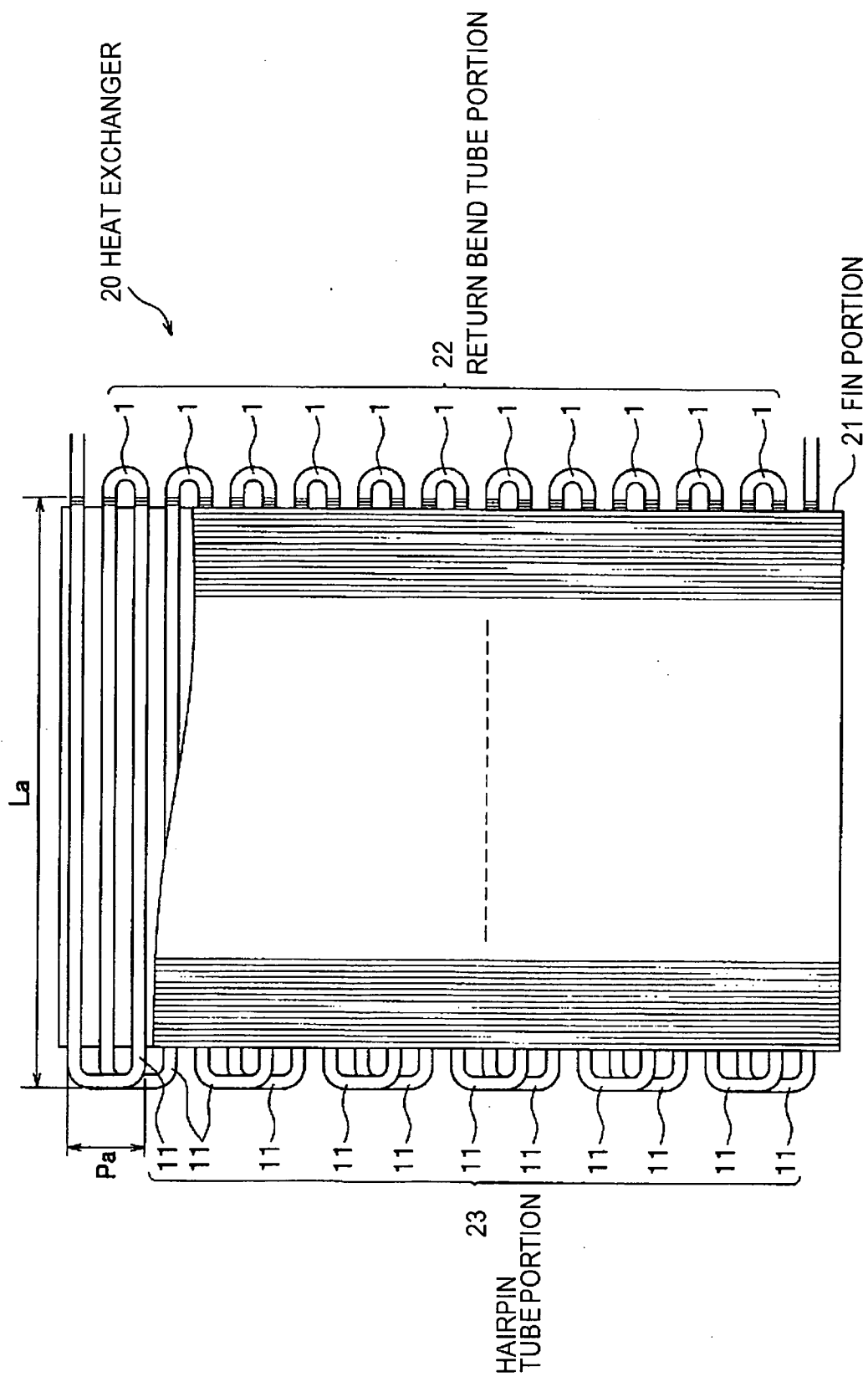


Fig. 3

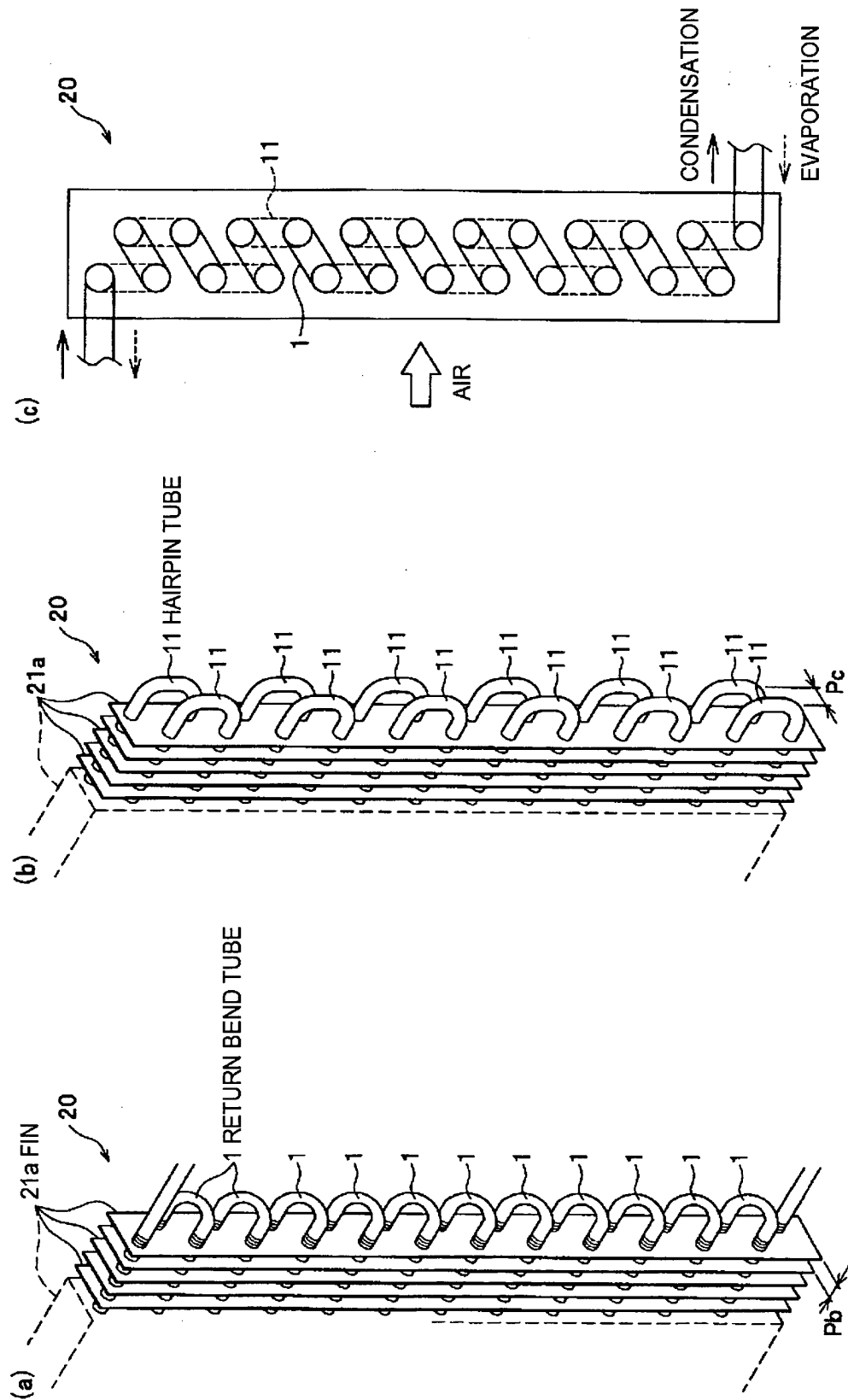


Fig.4

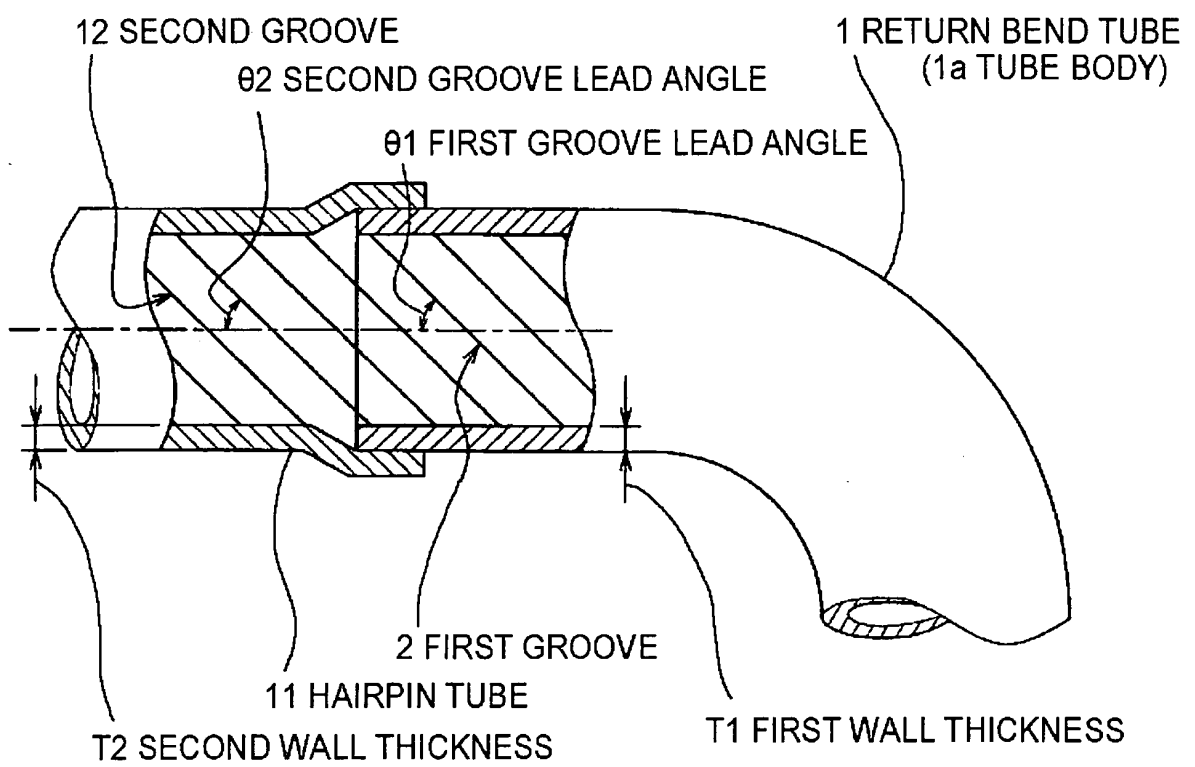


Fig.5

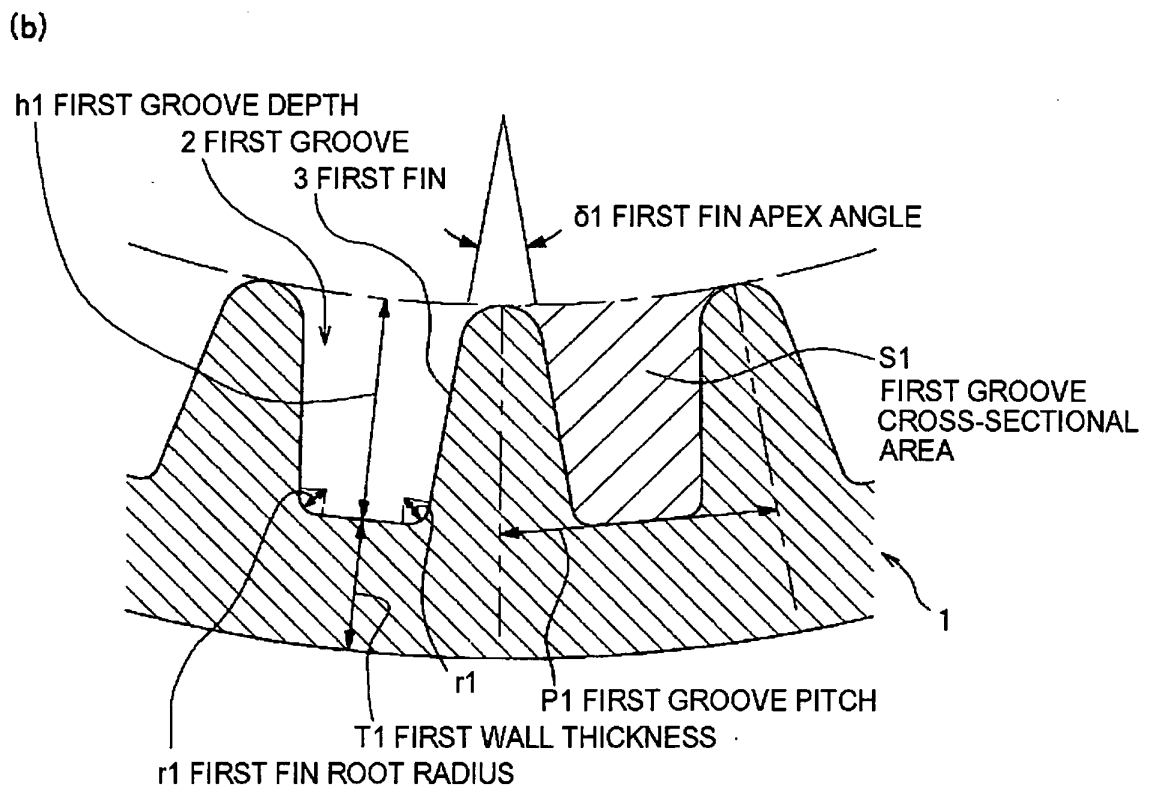
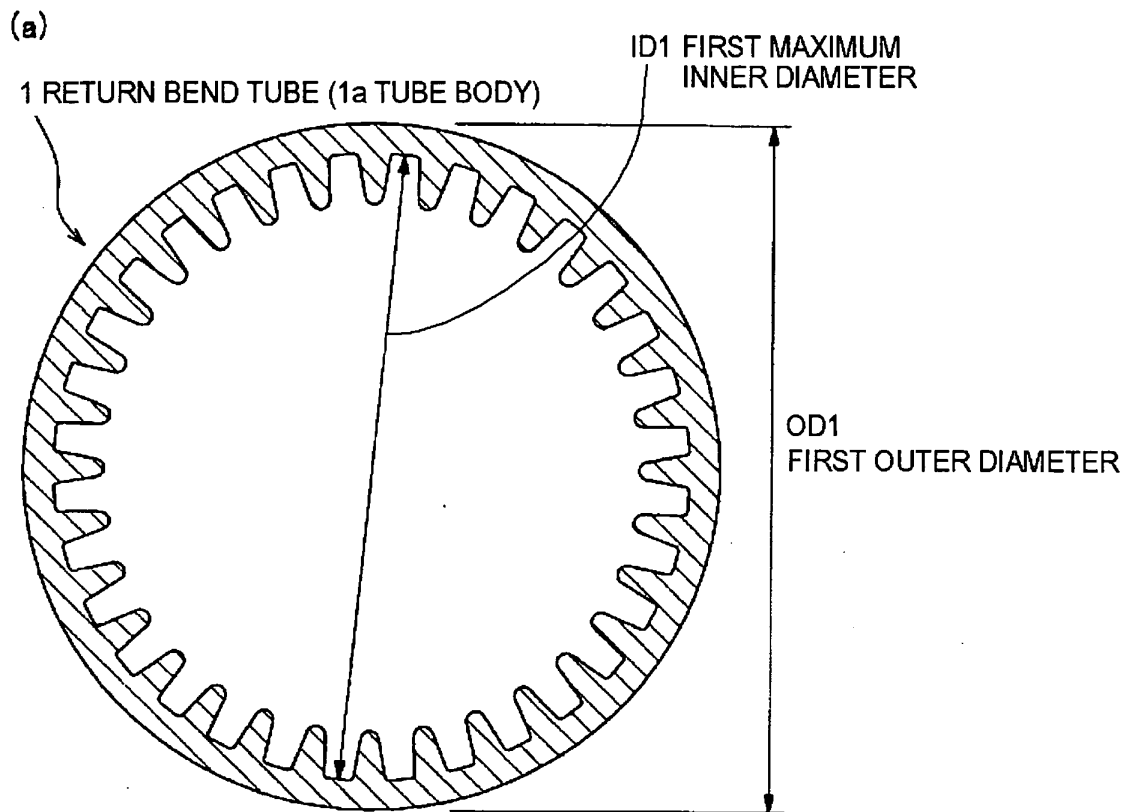


Fig. 6

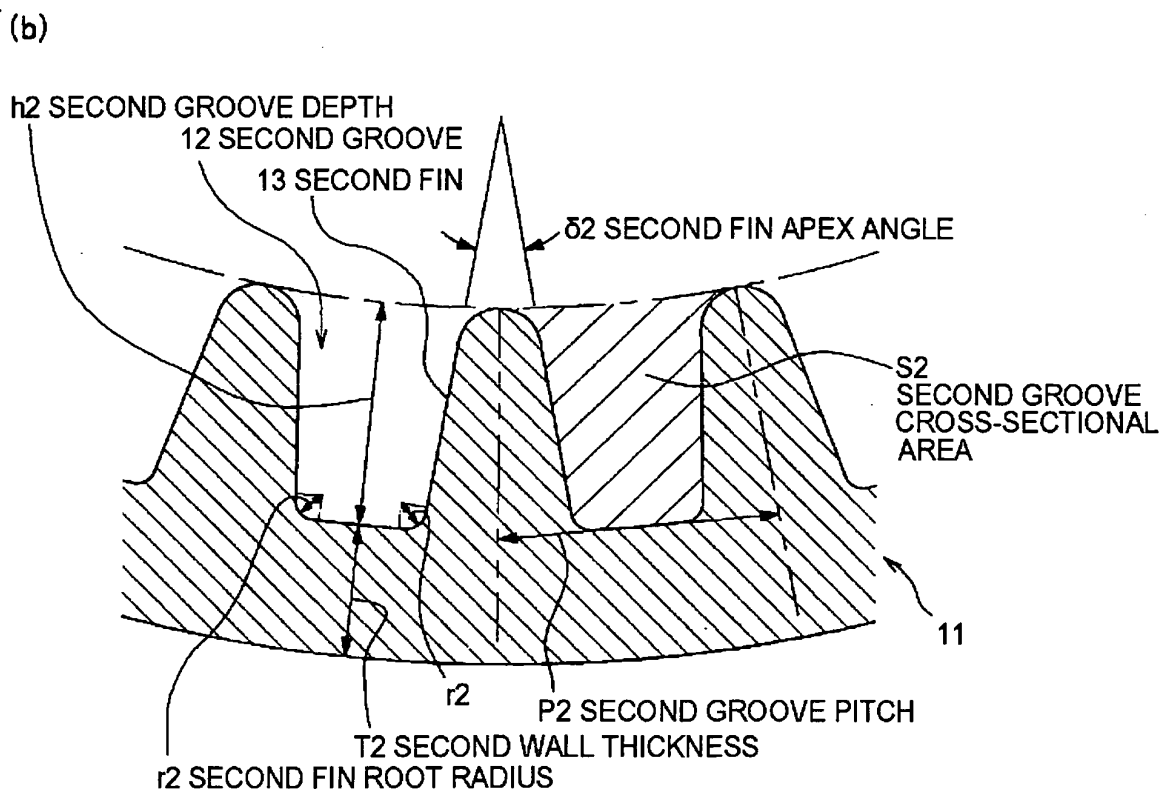
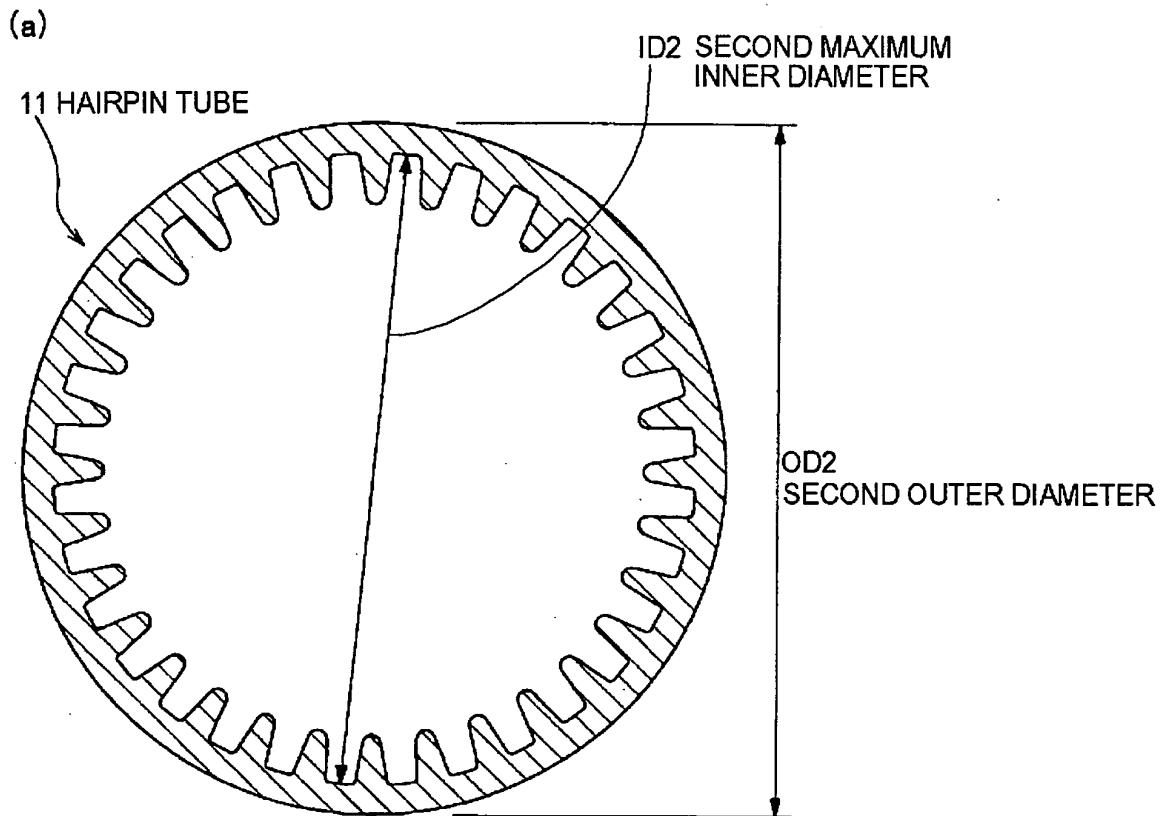


Fig. 7

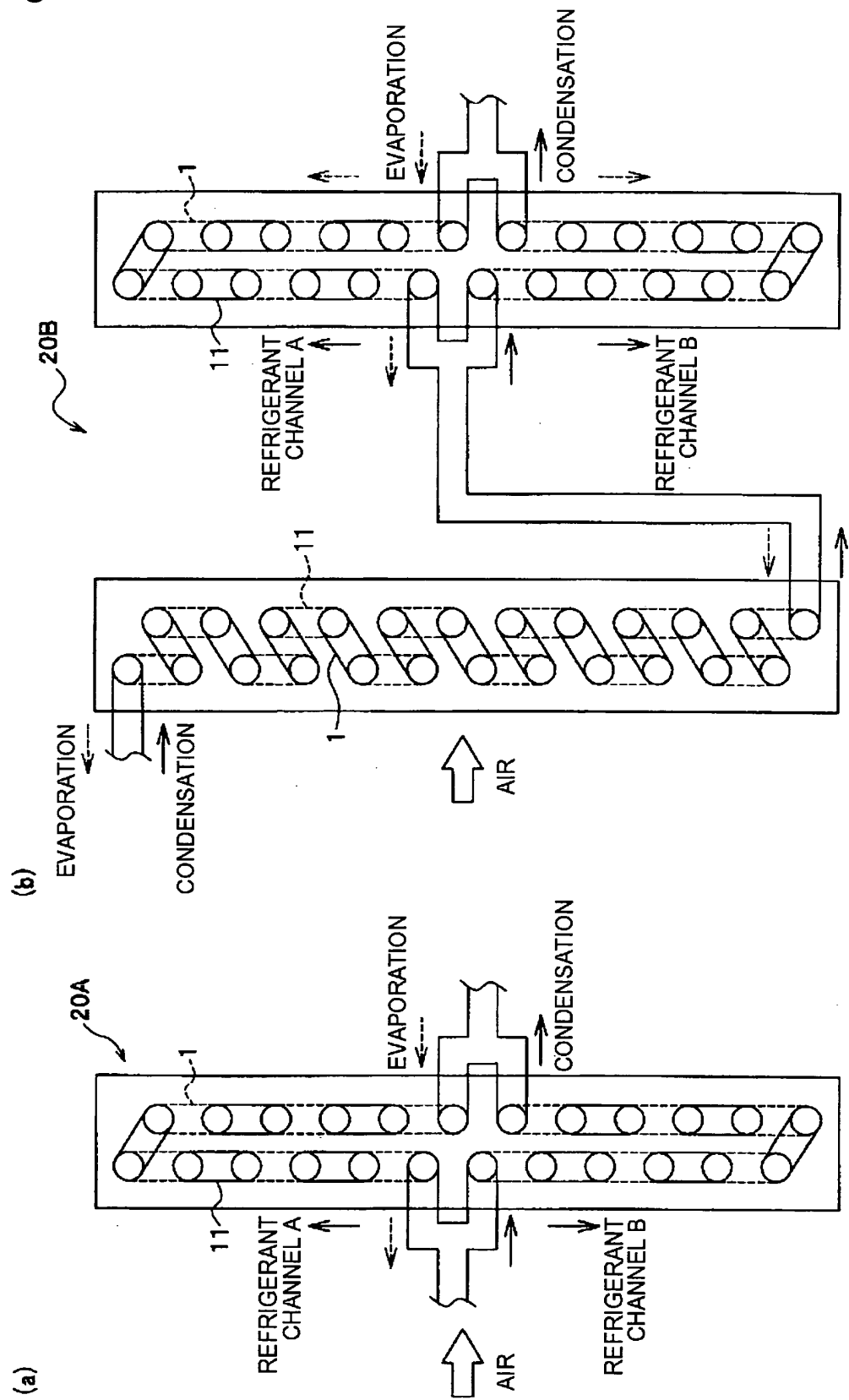
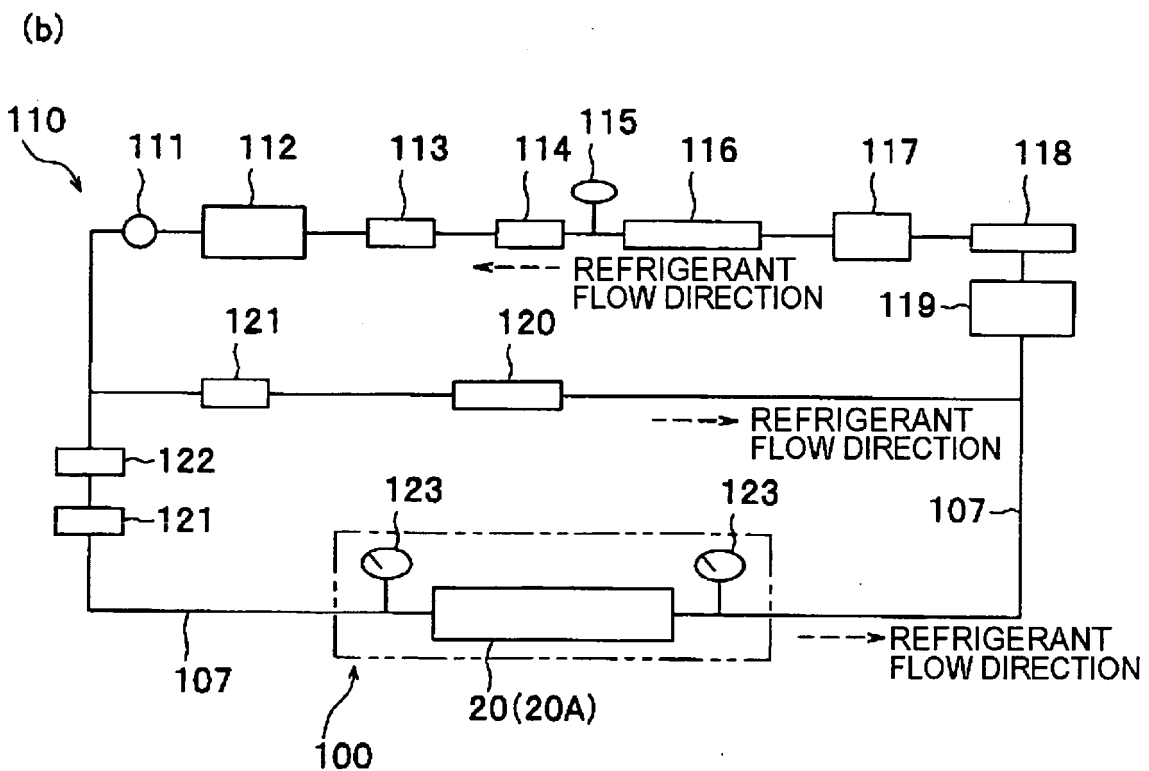
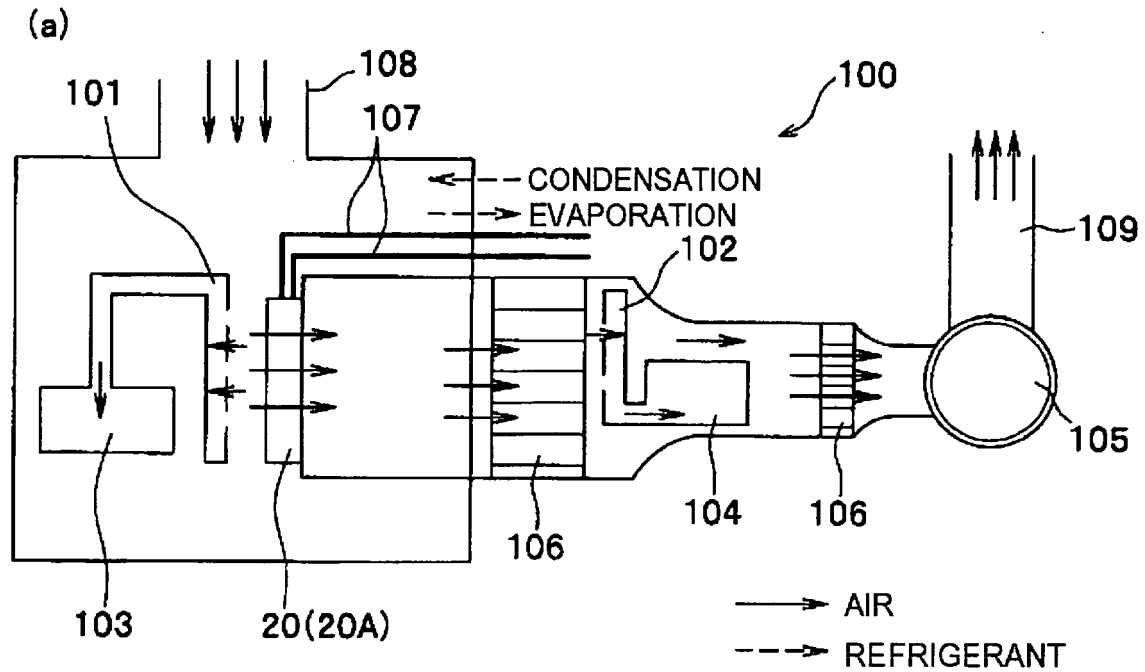


Fig. 8

INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2007/063807

A. CLASSIFICATION OF SUBJECT MATTER

F28D1/047(2006.01)i, F25B1/00(2006.01)i, F25B39/02(2006.01)i, F28F1/40
(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)

F28D1/047, F25B1/00, F25B39/02, F28F1/40

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 1996-2007
Kokai Jitsuyo Shinan Koho 1971-2007 Toroku Jitsuyo Shinan Koho 1994-2007

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	JP 2006-98033 A (Kobelco & Materials Copper	1-9
Y	Tube, Ltd.), 13 April, 2006 (13.04.06), Full text; Figs. 1 to 8 (Family: none)	10
Y	JP 10-292992 A (Toshiba Corp.), 04 November, 1998 (04.11.98), Par. No. [0025]; Fig. 2 (Family: none)	10

☐ Further documents are listed in the continuation of Box C.

☐ See patent family annex.

* Special categories of cited documents:

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"&" document member of the same patent family

Date of the actual completion of the international search
04 October, 2007 (04.10.07)

Date of mailing of the international search report
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REFERENCES CITED IN THE DESCRIPTION

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