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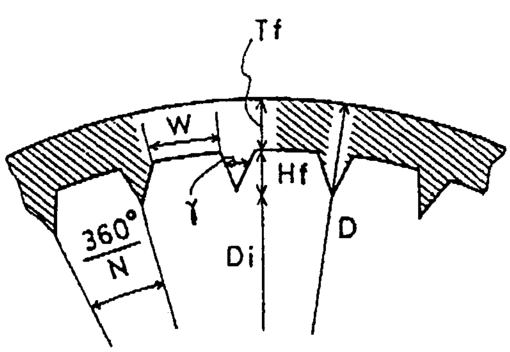
(54) **Heat transfer tube with grooved inner surface.**

(57) The present invention provides a heat transfer tube with grooved inner surface (1) superior in heat transfer performance and more lightweight.

The said heat transfer tube with grooved inner surface (1) has spiral grooves (2) on its inner surface (1) in which the helix angle (alpha) of grooves from the tube axis ranges from 16 to 22°, the ratio (Hf/Di) of the groove depth (Hf) to the inside diameter (Di) of the tube ranges from 0.023 to 0.025, the wall thickness (Tf) between the groove bottom and the tube surface ranges from 0.25 to 0.35 mm, and in-tube fluid causes a phase change; wherein the ratio (W/Hf) of the groove bottom width (W) to the groove depth (Hf) on a cross section perpendicular to the axis is set to 1.22-1.27 and the apex angle (gamma) of a ridge formed between grooves is set to 50-55° at the cross section perpendicular to grooves.

FIG.3



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FIELD OF THE INVENTION

The present invention relates to a heat transfer tube used for the heat exchanger of air conditioners, refrigerators, and boilers; particularly to a heat transfer tube with grooved inner surface suitable for purposes in which a fluid flowing through the tube causes a phase change.

BACKGROUND OF THE INVENTION

As shown in Figs. 1 and 2, the heat transfer tube with grooved inner surface 1 has ridges (fins) 3 made by forming many spiral grooves 2 on the inner surface of a metallic tube such as a copper tube, which is used for purposes in which a fluid flowing through the tube causes a phase change between liquid and gas phases.

The factors determining the performance of the heat transfer tube with grooved inner surface under the above phase change state are considered to be the agitation effect by a fluid due to irregularity of the inner surface, heat transfer acceleration effect by increase of the inner surface area, and liquid-line fluctuation effect in the region of irregularity. For example, evaporation of a liquid is described below. The liquid flowing through a tube at a speed faster than a certain speed is quickly raised through an ultra-fine spiral groove by a combination of capillary action and the force caused by flow velocity to become a ring flow and evaporation is accelerated throughout the tube.

A heat transfer tube with grooved inner surface in actual use at present is disclosed in, for example, US Patent 4658892 and European Patent0148609. Table 1 shows typical shape of the tube. Each dimensional item shown in Table 1 corresponds to that of Figs. 1 and 3.

The heat transfer tube shown in Table 1 has a satisfactory heat transfer performance. However, it is desired to make the heat exchanger more compact, lightweight, and low cost, while improving the performance of the heat transfer tube and decrease its weight.

Table 1

Item			Dimension
Outside diameter of tube	D	mm	9.52
Inside diameter of tube	Di	mm	8.52
Groove depth (Fin height)	Hf	mm	0.20
Bottom wall thickness (Wall thickness from groove bottom to tube surface)	Tf	mm	0.30
Ridge apex angle (Fin apex angle)	γ	Degree	53 Cross section perpendicular to grooves
Lead angle (Helix angle of groove from tube axis)	α	Degree	18
Number of ridges (Number of fins)	N	-	60
Unit weight (Material: Copper)	-	g/m	94

SUMMARY OF THE INVENTION

It is an object of the present invention to provide a heat transfer tube with grooved inner surface having high heat-transfer characteristics.

It is an another object of the present invention to provide a heat transfer tube with grooved inner surface with less weight per unit length.

It is a further object of the present invention to provide a heat transfer tube with grooved inner surface with fins to be easily formed.

The heat transfer tube with grooved inner surface for achieving these objects has been developed as the result of studying the correlation between the shape of grooves, shape of ridges (fins) formed between grooves, and heat transfer performance from various perspectives.

The heat transfer tube of the present invention premises that spiral grooves are formed on the inner surface, the helix angle of grooves from the tube axis (lead angle "alpha") ranges from 16 to 22°, the ratio (Hi/Di) of the groove depth (Hf) to the inside diameter of the tube (Di) ranges from 0.023 to 0.025, and the wall thickness between the groove bottom and the tube surface (bottom wall thickness Tf) ranges from 0.25 to 0.35 mm. Among these prerequisites, selection of the (Hf/Di) ratio is based mainly on the performance of the heat transfer tube, and the groove lead angle is determined by also considering the ease of manufacturing the heat transfer tube. The heat transfer tube of the present invention obtains maximum heat transfer performance by combining these prerequisites with requirements for the shape of grooves and ridges to be described later. It is preferable for the outside diameter of the tube (D) to range between 9.5 and 10 mm.

The first requirement for shape in the present invention is to keep the ratio (W/Hf) of the groove bottom width (W) to the groove depth or fin height (Hf) on a cross section perpendicular to the axis at a range of 1.22 to 1.27. For a ratio (W/Hf) of less than 1.22, if the condensate of a fluid in the tube remains in the grooves, the fins will easily become covered and condensation capacity will decrease. For a ratio (W/Hf) of more than 1.27, however, the inner surface area of the tube decreases and the heat transfer performance drops. When the number of ridges (number of fins) is decreased, for example, the ratio (W/Hf) exceeds the specified range of the present invention.

The second requirement for shape in the present invention is to set the apex angle (gamma) of the ridge (fin) to 50-55° on a cross section perpendicular to grooves. In general, as the apex angle (gamma) decreases, the heat transfer performance is improved both for evaporation and condensation. However, it is necessary to keep the apex angle at a range between 50 to 55° in consideration of the machinability of fins when manufacturing heat transfer tubes.

The present invention makes it possible to obtain a lightweight heat transfer tube with grooved inner surface with a satisfactory machinability for manufacturing tubes by keeping the ratio (W/Hf) of the groove bottom width (W) to the fin height (Hf) within the above mentioned range to specify the groove shape, combining the specified groove shape with the fin shape whose apex angle is limited to accelerate heat transfer, and properly keeping the flow line of their regular region to improve performance.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is a drawing of longitudinal section of a heat transfer tube with grooved inner surface;
Fig. 2 is a cross sectional drawing of the heat transfer tube with grooved inner tube;
Fig. 3 is a partially enlarged cross sectional drawing of the heat transfer tube with grooved inner tube showing dimensional items;
Fig. 4 is a graph showing the relationship between the refrigerant flow rate and the vaporization heat transfer rate on the heat transfer tube with grooved inner surface of the present invention, a heat transfer tube with grooved inner tube of the prior art, and a bare tube free from inner-surface grooves;
Fig. 5 is a graph showing the relationship between the refrigerant flow rate and the condensation heat transfer rate on the heat transfer tube with grooved inner surface of the present invention, a heat transfer tube with grooved inner tube of the prior art, and a bare tube free from inner-surface grooves; and
Fig. 6 is a graph showing the relationship between the ratio of groove width to groove depth and the condensation heat transfer rate on the heat transfer tube with grooved inner surface of the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

The following are comparisons between examples of the present invention and comparative examples. However, the present invention is not restricted to these embodiments.

Example 1

Evaporation and condensation tests are performed by using Freon-R22 as a refrigerant and pouring the refrigerant into the heat transfer tube of the present invention with spiral grooves on its inner surface and having the dimensions in Table 3 by changing the refrigerant flow rate in accordance with the conditions shown in Table 2.

Fig. 4 shows the relationship between the refrigerant flow rate and the vaporization heat transfer rate on the heat transfer tube obtained through the above tests and Fig. 5 shows the relationship between the refrigerant flow rate and the condensation heat transfer rate on it.

The existing-shape heat transfer tube with spiral grooves on its inner surface shown in Table 1 and a smooth copper tube with an outside diameter of 9.52 mm and inside diameter of 8.52 mm free from spiral grooves on its inner surface are fabricated, and evaporation and condensation tests are applied to the tubes in accordance with the conditions shown in Table 1. Figs. 4 and 5 show the test results.

Table 2

Item	Evaporation test	Condensation test
Temperature in front of tube-entrance expansion valve (° C)	40	-
Evaporation temperature (° C)	7.5 (Exit)	-
Condensation temperature (° C)	-	50 (Entrance)
Degree of superheating (° C)	5	35
Degree of subcooling (° C)	-	5
Pressure in front of tube-entrance expansion valve (kgf/cm^2G)	1.842	-
Tube-entrance pressure (kgf/cm^2G)	-	1.842
Tube-exit pressure (kgf/cm^2G)	0.53	-
Refrigerant flow rate (kg/h)	20 ~ 60	20 ~ 60

Table 3

Item			Dimension
Outside diameter of tube	D	mm	9.52
Inside diameter of tube	Di	mm	8.52
Groove depth (Fin height)	Hf	mm	0.20
Bottom wall thickness (Wall thickness from groove bottom to tube surface)	Tf	mm	0.30
Ridge apex angle (Fin appex angle)	γ	Degree (°)	53 (Cross section perpendicular to grooves)
Lead angle (Helix angle of groove from tube axis)	α	Degree (°)	18
Number of ridges (Number of fins)	N	-	55
Unit weight (Material: Copper)	-	g/m	92.6

From Figs. 4 and 5, it is found that the heat transfer tube with grooved inner surface of the present invention performs better, especially in view of the heat transfer rate under condensation, than the conventional-shape heat transfer tube with grooved inner surface.

Example 2

A condensation test is applied to the heat transfer tube of the present invention with spiral grooves on its inner surface and having the dimensions shown in Table 3 by changing the fin width (groove width) of the tube to change the ratio (W/Hf) of the groove width (W) to Hf on a cross section perpendicular to the axis under the same conditions as in embodiment 1. In this case, the refrigerant flow rate is set to 50 kg/h. Fig. 6 shows the test results.

From Fig. 6, it is found that the heat transfer rate reaches its peak at a point where the ratio of groove bottom width W to groove depth (fin height) Hf is approximately 1.25.

As described above, the present invention makes it possible to provide a heat transfer tube with grooved inner surface superior in heat transfer performance and more lightweight than the conventional tube while decreasing the cost of heat exchangers.

Claims

1. A heat transfer tube with grooved inner surface with spiral grooves on its inner surface in which the helix angle of grooves from the tube axis ranges from 16 to 22°, the ratio (Hf/Di) of the groove depth (Hf) to the inside
   diameter (Di) of the tube ranges from 0.023 to 0.025, the wall thickness (Tf) between the groove bottom and the tube surface ranges from 0.25 to 0.35 mm, and in-tube fluid causes a phase change; wherein the ratio (W/Hf) of the groove bottom width (W) to the groove depth (Hf) on a cross section perpendicular to the axis is set to 1.22-1.27 and the apex angle of a ridge formed between grooves is set to 50-55° on a cross section perpendicular to grooves.

2. A heat transfer tube with grooved inner surface according to claim 1, wherein the outside diameter (D) of the tube ranges from 9.5 to 10 mm.

3. A heat transfer tube with grooved inner surface according to claim 1, wherein the tube is made of copper.

FIG.1

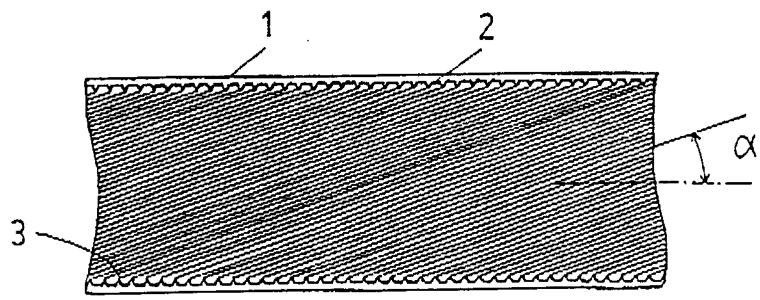


FIG.2

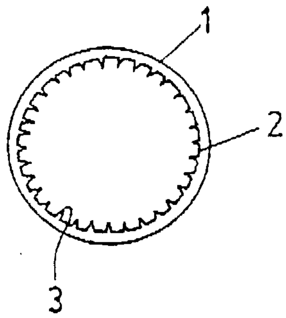


FIG.3

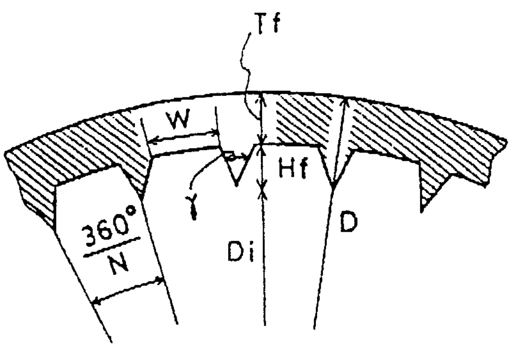
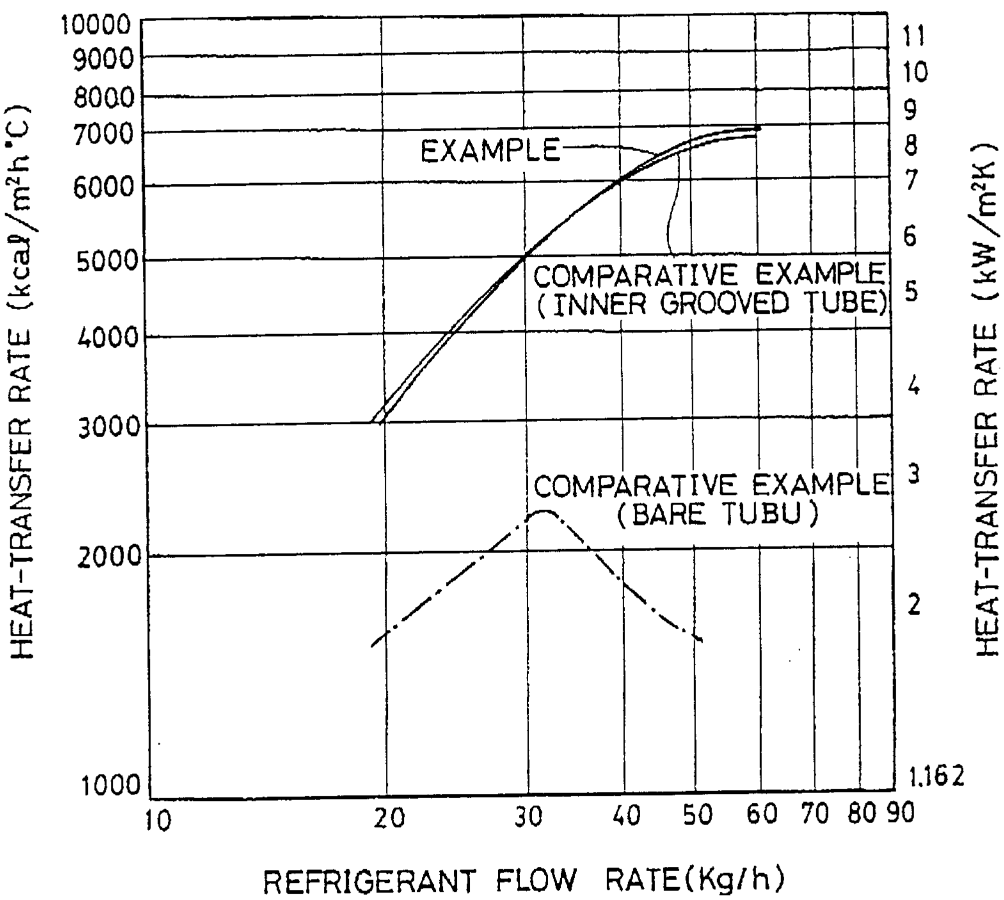


FIG.4


10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
HEAT-TRANSFER RATE (kcal/m²h°C)
EXAMPLE
COMPARATIVE EXAMPLE
(INNER GROOVED TUBE)
COMPARATIVE EXAMPLE
(BARE TUBU)
11
10
9
8
7
6
5
4
3
2
1.162
HEAT-TRANSFER RATE (kW/m²K)
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40
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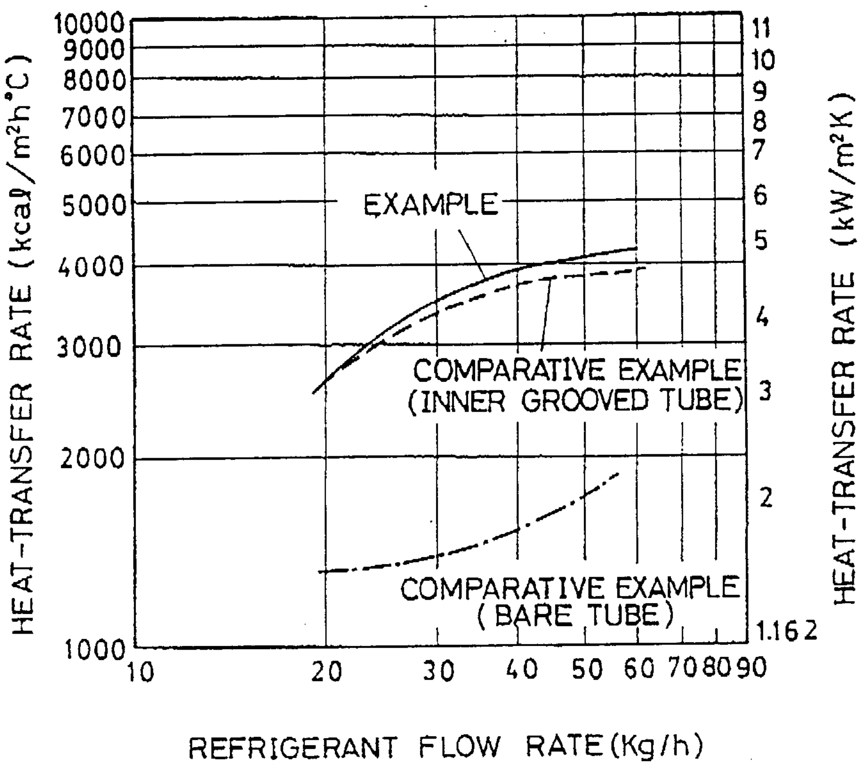
60
70
80
90
REFRIGERANT FLOW RATE(Kg/h)

FIG.5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HEAT-TRANSFER RATE (kcal/m²h°C)
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HEAT-TRANSFER RATE (kW/m²K)
EXAMPLE
COMPARATIVE EXAMPLE (INNER GROOVED TUBE)
COMPARATIVE EXAMPLE (BARE TUBE)
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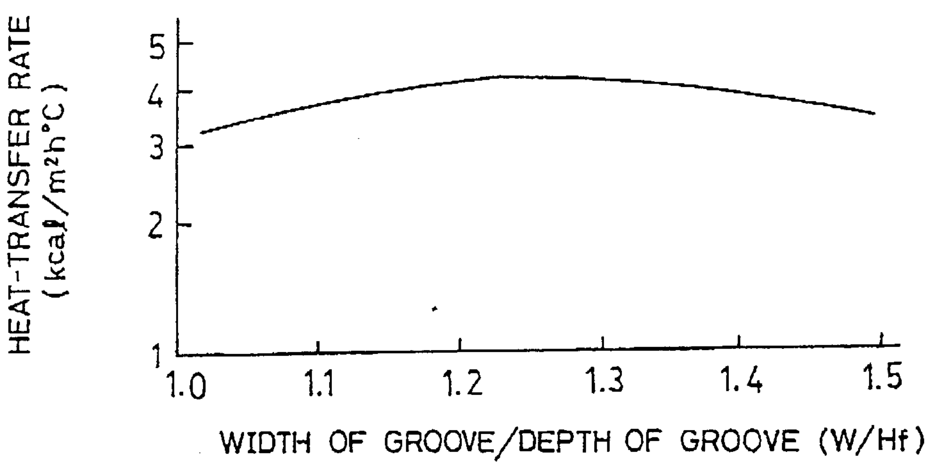
REFRIGERANT FLOW RATE (Kg/h)

FIG.6


HEAT-TRANSFER RATE (kcal/m²h°C)
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3
2
1
1.0
1.1
1.2
1.3
1.4
1.5
WIDTH OF GROOVE/DEPTH OF GROOVE (W/Hf)

European Patent Office

EUROPEAN SEARCH REPORT

Application Number

EP 92 10 9811

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl.5)
D,A	US-A-4 658 892 (SHINOHARA) * the whole document *	1	F28F1/40 F28F13/18
D,A	EP-A-0 148 609 (HITACHI CABLE,LTD) * the whole document *	1	
A	DE-A-3 414 230 (BEHM) * the whole document *	1	
A	DE-A-2 552 679 (HITACHI,LTD) * the whole document *	1	
			TECHNICAL FIELDS SEARCHED (Int. Cl.5)
			F28F

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	08 SEPTEMBER 1992	SMETS E.D.C.

CATEGORY OF CITED DOCUMENTS

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