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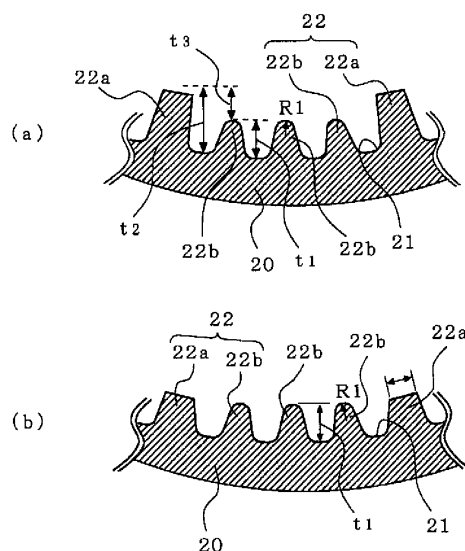
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(54) **HEAT TRANSFER TUBE FOR HEAT EXCHANGER, HEAT EXCHANGER, REFRIGERATION CYCLE DEVICE, AND AIR CONDITIONING DEVICE**

(57) A heat transfer pipe for a heat exchanger including high threads 22a and low threads 22b that are lower than the high threads 22a. The high threads 22a and the low threads 22b are provided at respective prescribed heights in a helical manner in a pipe axial direction on an inner surface of a pipe 20, the high threads 22a are formed with 11 to 19 threads, the low threads 22b are formed with 3 to 6 threads between each pair of the high threads 22a, the high threads 22a before expansion of the pipe each have a trapezoidal shape in cross-section so that a crest before the expansion of the pipe is flat, and a ratio $W1/D$ of a tip width $W1$ of the crest portion after the expansion of the pipe to an outer diameter D of the heat transfer pipe 20 is 0.011 to 0.040. Further, the high threads 22a before the expansion of the pipe are higher by 0.04 mm or more than the low threads 22b.

FIG. 2



Description

Technical Field

5 **[0001]** The present invention is related to a heat transfer pipe for a heat exchanger having a groove on the inner surface of the pipe, a heat exchanger, a refrigeration cycle apparatus, and an air-conditioning apparatus.

Background Art

10 **[0002]** Conventionally, in a heat exchanger used in a refrigeration apparatus, an air-conditioning apparatus, a heat pump, and the like, typically, a plurality of fins aligned at prescribed intervals are provided with through holes, and a heat transfer pipe that has grooves formed in its inner surface is disposed in the through holes. The heat transfer pipe becomes a portion of a refrigerant circuit in a refrigeration cycle apparatus and a refrigerant (fluid) is made to flow in the pipe.

15 **[0003]** The grooves on the inner surface of the pipe are fabricated such that the pipe axial direction and the direction in which the grooves extend form a fixed angle. Herein, concavities and convexities are created on the inner surface of the pipe by the formation of the grooves. The space in the concavity is called a groove portion and the convexity formed by the sidewalls of adjacent grooves is called a thread portion.

20 **[0004]** A refrigerant that flows through such a heat transfer pipe changes phase (condenses or evaporates) by heat exchange with the air outside of the heat transfer pipe or the like. In order to carry out the phase change effectively, the heat transfer performance of the heat transfer pipe is improved by increasing the surface area in the pipe, by a fluid stirring effect by the groove portions, by a liquid film retaining effect between the groove portions by the capillary action of the groove portions, and the like (for example, refer to Patent Literature 1).

Citation List

25 Patent Literature

[0005]

30 Patent Literature 1: Japanese Unexamined Patent Application, First Publication No. 60-142195 (page 2 and Fig. 1)

Summary of Invention

Technical Problem

35 **[0006]** The heat transfer pipe as described in the above Patent Literature 1 is generally made of metal of copper or copper alloy. In the manufacture of the heat exchanger, a mechanical pipe expanding technique is used in which pipe expanding balls are pushed into the pipe to expand the heat transfer pipe from the inside, and thus the fins and the heat transfer pipe are adhered and fixed. However, when expanding the pipe, there have been problems in that the thread portions falls over due to the pipe expanding balls, leading to reduction in adherence between the fins and the heat transfer pipe. Further, the pressure loss in the pipe increases causing a drop in the heat transfer performance.

40 **[0007]** The present invention has been made to solve the above problems, and an object of the present invention is to provide a heat transfer pipe for a heat exchanger in which the adherence between the heat transfer pipe and the fins is improved and a prescribed heat transfer performance can be obtained without increasing the pressure loss in the pipe, as well as providing a heat exchanger using this heat transfer pipe, a refrigeration cycle apparatus using the heat exchanger, and an air-conditioning apparatus using the refrigeration cycle apparatus.

Solution to Problem

50 **[0008]** A heat transfer pipe for a heat exchanger according to the present invention includes high threads and low threads that are lower than the high threads, in which the high threads and the low threads are provided at respective prescribed heights in a helical manner in a pipe axial direction on a pipe inner surface, the high threads are formed with 11 to 19 threads, the low threads are formed of 3 to 6 threads between each pair of the high threads, the high threads before expansion of the pipe each have a trapezoidal shape in cross-section such that a crest thereof is flat, and a ratio of a tip width of the crest portion after the expansion of the pipe to an outer diameter of the heat transfer pipe is 0.011 to 0.040.

55 **[0009]** The heat exchanger according to the present invention includes a plurality of fins for exchanging heat; and any of the above heat transfer pipes that penetrates the fins, in which the heat transfer pipe is pressurized from an inner

surface side, is expanded, and is fixed to the fins.

[0010] The refrigeration cycle apparatus according to the present invention includes a refrigerant circuit including a compressor that compresses a refrigerant, a condenser that condenses the refrigerant by heat exchange, an expansion means for decompressing the condensed refrigerant, and an evaporator that evaporates the decompressed refrigerant by heat exchange connected by piping to circulate the refrigerant, in which any of the above heat exchanger is provided to both or either one of the condenser and the evaporator.

[0011] The air-conditioning apparatus of the present invention in which the refrigeration cycle apparatus carries out cooling/heating of a conditioned space

Advantageous Effects of Invention

[0012] According to the heat transfer pipe for a heat exchanger of the present invention, when expanding the heat transfer pipe by the mechanical pipe expanding technique, although pipe expanding balls contact the high threads and the crest portions are crushed flat, the crest portions do not fall over, and compared to a conventional heat transfer pipe, the heat transfer performance in the pipe can be increased without increasing the pressure loss. Further, the outer surface of the heat transfer pipe is fabricated into a polygonal shape, springback of the heat transfer pipe is suppressed, and the adherence between the heat transfer pipe and the fins can be improved.

In addition, a highly efficient heat exchanger, refrigeration cycle apparatus, and air-conditioning apparatus can be provided using this heat transfer pipe.

Brief Description of Drawings

[0013]

[Fig. 1] Fig. 1 is a perspective view of a heat exchanger according to Embodiment 1 of the present invention, and a cross-sectional side view when the heat exchanger is cut parallel to a fin surface.

[Fig. 2] Fig. 2 is an enlarged cross-sectional view of portion A of the heat transfer pipe in Fig. 1 before pipe expansion and after pipe expansion.

[Fig. 3] Fig. 3 is an explanatory diagram showing the expanded state of the heat transfer pipe of Fig. 1 during pipe expansion by a mechanical pipe expanding technique.

[Fig. 4] Fig. 4 is a diagram showing the relationship between the number of high threads and the heat exchange rate in the heat transfer pipe according to Embodiment 1.

[Fig. 5] Fig. 5 is a diagram showing the relationship between the ratio of the tip width of the high threads after pipe expansion to the outer diameter of the heat transfer pipe and the heat exchange rate in the heat transfer pipe according to Embodiment 1.

[Fig. 6] Fig. 6 is a cross-sectional view showing a pipe inner surface shape after pipe expansion of a heat transfer pipe according to Embodiment 2 of the present invention.

[Fig. 7] Fig. 7 is a diagram showing the relationship between the difference between the groove portion and the thread portion after pipe expansion and the heat exchange rate in the heat transfer pipe according to Embodiment 2.

[Fig. 8] Fig. 8 is a cross-sectional front view of a pipe inner surface of a heat transfer pipe according to Embodiment 3 of the present invention.

[Fig. 9] Fig. 9 is a diagram showing the relationship between a lead angle of a helical groove and the heat exchange rate in the heat transfer pipe according to Embodiment 3.

[Fig. 10] Fig. 10 is a cross-sectional view showing a pipe inner surface shape of a heat transfer pipe according to Embodiment 4 of the present invention.

[Fig. 11] Fig. 11 is a diagram showing the relationship between the apex angle of the high threads and the heat exchange rate in the heat transfer pipe according to Embodiment 4.

[Fig. 12] Fig. 12 is a system circuit diagram of an air-conditioning apparatus according to Embodiment 5 of the present invention.

Description of Embodiments

Embodiment 1

[0014] A heat exchanger in Fig. 1 is a fin-tube heat exchanger widely used as an evaporator and condenser in a refrigeration apparatus, an air-conditioning apparatus, and the like.

The heat exchanger 1 is constituted by a plurality of fins 10 for a heat exchanger and a heat transfer pipe 20. Through holes 11 are provided in each of the plurality of fins 10 that are aligned at prescribed intervals, and the heat transfer

pipe 20 penetrates through the through holes 11. The heat transfer pipe 20 becomes a portion of a refrigerant circuit in a refrigeration cycle apparatus. By transferring heat between the refrigerant flowing within the heat transfer pipe 20 and the air flowing outside via the fins 10, the heat transfer area expands and heat exchange between the refrigerant and the air can be efficiently performed.

[0015] As shown in Fig. 2, the pipe inner surface of the heat transfer pipe 20 is provided with groove portions 21 and thread portions 22 made by groove formation, and as also shown in Fig. 2, the thread portions 22 is further constituted by two types of thread portion, high threads 22a and low threads 22b. A plurality of low threads 22b is formed between the high threads 22a. The high threads 22a each have a trapezoidal shape in cross-section so that their crests before pipe expansion (Fig. 2(b)) are flat, and a ratio ($W1/D$) of a tip width $W1$ of the crest portions of the high threads 22a after pipe expansion (Fig. 2(a)) to an outer diameter D of the heat transfer pipe 20 is 0.011 to 0.040. A height $t1$ of the low threads 22b before pipe expansion is lower by $t3$, that is, 0.04 mm or greater, than a height $t2$ of the high threads 22a. However, if the difference between the high threads 22a and the low threads 22b is too great (even if the low threads 22b are too low), the heat performance may be reduced due to a reduction in the surface area in the pipe or the like, and thus in Embodiment 1, the difference is close to 0.04 mm.

[0016] In Fig. 3, in the heat exchanger 1, a pipe is first bent at a middle portion in the longitudinal direction into a hairpin shape with a prescribed bending pitch. A plurality of hairpin pipes that will become the heat transfer pipe 20 are created. Next, the hairpin pipes are inserted through the through holes 11 of the fins 10, and then the hairpin pipes are expanded by a mechanical pipe expanding technique to form the heat transfer pipe 20. The heat transfer pipe 20 is then closely adhered and fixed to the fins 10. The mechanical pipe expanding technique is a method in which a rod 31, which has on its front tip a pipe expanding ball 30 whose diameter is slightly larger than that of the inner diameter of the heat transfer pipe 20, is passed through the inside of the heat transfer pipe 20 to closely adhere the heat transfer pipe 20 to the fins 10 by expanding the outer diameter of the heat transfer pipe 20.

[0017] When pipe expanding by the mechanical pipe expanding technique, since the pipe expanding ball 30 makes contact with the high threads 22a, the crest portions of the high threads 22a are crushed flat to lower the height of the threads. On the other hand, since the crest portions of the low threads 22b are lower than the crushed height portion of 0.04 mm, their shapes do not change (refer to Fig. 2). Not as in the known art, since pressure from the insertion of the pipe expanding ball 30 is not applied to all of the thread portion in the pipe when expanding the pipe, but rather only to the high threads 22a, the outer surface of the heat transfer pipe 20 is fabricated into a polygonal shape, and springback of the heat transfer pipe 20 can be suppressed. Thereby, the adherence between the heat transfer pipe 20 and the fins 10 can be improved, and the efficiency of heat exchange can be increased.

[0018] Fig. 4 shows a relationship between the number of high threads 22a and the heat exchange rate. On the inner surface of the heat transfer pipe 20, 11 to 19 high threads 22a are continuously formed in a helical manner in the axial direction, and 3 to 6 low threads 22b are formed between a high thread 22a and a high thread 22a.

[0019] The reason that the number of high threads 22a of the heat transfer pipe 20 is set in the range of 11 to 19, as in the above heat exchanger 1, is that during pipe expansion, while the pipe expanding ball 30 contacts the high threads 22a so that their crest portions are crushed flat by about 0.04 mm to lower the height of the threads, if the number of high threads 22a of the heat transfer pipe 20 is lower than 11, the crest portions of the low threads 22b will also be crushed flat, thereby decreasing the heat transfer performance in the pipe. Further, if the number of high threads is greater than 19, the number of low threads 22b will be reduced, thereby decreasing the heat transfer performance in the pipe.

[0020] In the heat transfer pipe 20 after pipe expansion, the ratio ($W1/D$) of the tip width $W1$ of the crest portions of the high threads 22a to the outer diameter D of the heat transfer pipe 20 is 0.011 to 0.040 (refer to Fig. 2).

[0021] Fig. 5 shows a relationship between the ratio ($W1/D$) of the tip width $W1$ of the high threads 22a after pipe expansion to the outer diameter D of the heat transfer pipe 20 and the heat exchange rate. If the ratio ($W1/D$) of the tip width $W1$ after pipe expansion to the outer diameter D of the heat transfer pipe 20 is set to be 0.011 or less, when pipe expanding using the pipe expanding ball 30, the upper crest portions becomes crushed and the insertion pressure becomes weak. Therefore, the pipe expansion of the heat transfer pipe 20 becomes insufficient, the adherence between the heat transfer pipe 20 and the fins 10 is hindered, and reduction in the heat exchange rate becomes prominent. If the ratio ($W1/D$) of the tip width $W1$ to the outer diameter D of the heat transfer pipe 20 is set to be 0.040 or greater, the cross-section area of the groove portion 21 decreases, and thus the liquid film of the refrigerant becomes thick which leads to a remarkable reduction in the heat transfer coefficient.

[0022] On the other hand, if the curvature radius $R1$ of the tip portions (crests) of the low threads 22b is set to 0.03 mm to 0.035 mm, the bottom width of the thread becomes narrow and the overall thread is formed narrowly. Thereby, the heat transfer area increases and the heat transfer coefficient in the pipe is improved (refer to Fig. 2).

[0023] The high threads 22a each have a trapezoidal shape in cross-section so that their crests before pipe expansion are flat, the pressure of the crests becomes smaller, and the amount that the crests are crushed is reduced. However, if the curvature radii of the flat surface and both side surfaces of the crests are 0.01 mm or less, the manufacturing cost of the heat transfer pipe 20 may increase. Thus, it is preferable for the curvature radii of the flat surface and both side

surfaces of the crests to be 0.01 to 0.03 mm.

[0024] As described above, according to the heat exchanger 1 of Embodiment 1, thread portions 22 constituted by high threads 22a and low threads 22b are formed in a helical manner in a pipe axial direction on the pipe inner surface of the heat transfer pipe 20. The high threads 22a that are formed with 11 to 19 threads at a prescribed height each have a trapezoidal shape in cross-section so that their crests before pipe expansion are flat, and the ratio ($W1/D$) of the tip width $W1$ of the crest portions after pipe expansion to the outer diameter D of the heat transfer pipe 20 is 0.011 to 0.040. The low threads 22b have a height that is lower than the high threads 22a and are formed with 3 to 6 threads between a high thread 22a and a high thread 22a, and the curvature radius $R1$ of the crests of the low threads 22b is 0.03 mm to 0.045 mm. Therefore, the heat transfer performance in the heat transfer pipe 20 can be improved. Further, since the pipe expanding ball 30 contacts only the high threads 22a to expand the pipe, the outer surface of the heat transfer pipe 20 is fabricated into a polygonal shape, and springback of the heat transfer pipe 20 is suppressed. Thereby, the adherence between the heat transfer pipe 20 and the fins 10 can be improved, and the heat exchange rate (ratio of the amount of heat before and after passage through the heat transfer pipe) can be increased so as to conserve energy. Further, the amount of refrigerant within the refrigerant circuit can be reduced while maintaining high efficiency and miniaturization can also be achieved.

Embodiment 2

[0025] Fig. 6 shows a shape of a pipe inner surface of a heat transfer pipe 20 according to Embodiment 2 of the present invention. Constitution of a heat exchanger 1 is the same as that in Embodiment 1. The same reference numerals as those used in Embodiment 1 will be assigned to those parts that play an identical or corresponding role (the same applies to the subsequent Embodiments below). In Embodiment 2, a difference H between a groove portion 21 and a thread portion 22 after pipe expansion will be explained.

[0026] Fig. 7 shows a relationship between a difference between the groove portion 21 and the thread portion 22 after pipe expansion (high threads 22a after pipe expansion) and the heat exchange rate. In the heat transfer pipe 20, as the difference H between the groove portion 21 and the thread portion 22 after pipe expansion increases, the surface area in the pipe increases, and thus the heat transfer coefficient also increases. However, if the difference H between the groove portion 21 and the thread portion 22 becomes larger than 0.26 mm, the amount of increase in the pressure loss becomes greater than the amount of increase in the heat transfer coefficient, and thus the heat exchange rate decreases. On the other hand, if the difference H between the groove portion 21 and the thread portion 22 is less than 0.1 mm, the heat transfer coefficient does not improve. Therefore, in the heat transfer pipe 20, the high threads 22a and low threads 22b are formed so that the difference H between the groove portion 21 and the thread portion 22 after pipe expansion becomes 0.1 mm to 0.26 mm.

According to the heat exchanger 1 of Embodiment 2 described above, since the high threads 22a and the low threads 22b are formed so that the difference H between the groove portion 21 and the thread portion 22 after pipe expansion becomes 0.1 mm to 0.26 mm, the heat transfer performance in the heat transfer pipe 20 can be improved.

Embodiment 3

[0027] Fig. 8 shows a shape of a pipe inner surface of a heat transfer pipe 20 according to Embodiment 3 of the present invention. An angle (lead angle or helix angle) γ formed by a straight line that is parallel to the pipe axial direction and a direction in which the groove portion (helical groove) 21 (thread portion 22) extend on the pipe inner surface of the heat transfer pipe 20 is 10 to 50 degrees.

[0028] Fig. 9 shows a relationship between the lead angle γ of the groove portion (helical groove) 21 in the heat transfer pipe 20 and the heat exchange rate. Basically, the reason that the lead angle γ of the groove portion (helical groove) 21 in the heat transfer pipe 20 is set in the range of 10 to 50 degrees is that if the lower limit of the lead angle γ of the groove portion (helical groove) 21 is set to 10 degrees or less, there is a remarkable reduction in the heat exchange rate, and if the upper limit of the lead angle γ of the groove portion (helical groove) 21 is set to 50 degrees or more, the pressure loss in the pipe increases. Thereby, a flow which flows over the groove portion (helical groove) 21 does not easily occur and the heat exchange rate can be improved without an increase in the pressure loss in the pipe, and thus a highly efficient air-conditioner can be obtained.

[0029] According to the heat exchanger 1 of Embodiment 3 described above, threads are formed so that the lead angle γ of the groove portion (helical groove) 21 in the heat transfer pipe 20 is 10 to 50 degrees. Therefore, the heat transfer performance in the heat transfer pipe 20 can be improved.

Embodiment 4

[0030] Fig. 10 shows a shape of a pipe inner surface of a heat transfer pipe 20 according to Embodiment 4 of the

present invention. In a heat exchanger 1, an apex angle α of high threads 22a of the heat transfer pipe 20 is 15 to 30 degrees, and an apex angle β of low threads 22b is 5 to 15 degrees.

[0031] Basically, as the apex angle of each thread portion decreases, the heat transfer area of the entire heat transfer pipe 20 increases, and thus the heat transfer coefficient increases. However, as shown in Fig. 11, if the apex angle α of the high threads 22a becomes smaller than 15 degrees, since workability when manufacturing the heat exchanger 1 decreases remarkably, the heat exchange rate ultimately decreases. On the other hand, if the apex angle α becomes larger than 30 degrees, since the cross-section area of the groove portion 21 decreases leading to the liquid film of refrigerant to overflow from the groove portion 21 and causing the liquid film to cover up to the crest portions, the heat transfer coefficient decreases.

[0032] On the other hand, by setting the apex angle β of the low threads 22b to 5 to 15 degrees, the bottom width of the threads is also formed narrowly, and the overall thread is formed narrowly. Thereby, the heat transfer area is increased and the heat transfer coefficient in the pipe is increased.

[0033] According to the heat transfer pipe 20 of Embodiment 4 described above, the high threads 22a and the low threads 22b are formed so that the apex angle α of the high threads 22a is 15 to 30 degrees and the apex angle β of the low threads 22b is 5 to 15 degrees. Therefore, the heat transfer performance in the heat transfer pipe 20 can be improved.

Embodiment 5

[0034] Fig. 12 shows an air-conditioning apparatus that is a refrigeration cycle apparatus according to Embodiment 5 of the present invention. The air-conditioning apparatus includes a heat source side unit (outdoor unit) 100 and a load side unit (indoor unit) 200, and these units are connected by refrigerant piping to constitute a refrigerant circuit for circulating a refrigerant. Among the refrigerant piping, a piping in which a refrigerant of gas flows is called a gas piping 300, and a piping in which a refrigerant of liquid (liquid refrigerant, there are cases of a gas-liquid two-phase refrigerant, also) flows is called a liquid piping 400. Herein, the refrigerant used includes, for example, an HC single refrigerant, a mixed refrigerant including an HC refrigerant, R32, R410A, R407C, tetrafluoropropene (for example, 2,3,3,3-tetrafluoropropene), a non-azeotropic refrigerant mixture consisting of HFC refrigerant having a boiling point that is lower than the tetrafluoropropene, carbon dioxide, or the like.

[0035] In Embodiment 5, the heat source side unit 100 includes the following devices (means) such as a compressor 101, an oil separator 102, a four-way valve 103, a heat source side heat exchanger 104, a heat source side fan 105, an accumulator 106, a heat source side expansion device (expansion valve) 107, refrigerant-to-refrigerant heat exchanger 108, a bypass expansion device 109, and a heat source side control device 110.

[0036] The compressor 101 has an electric motor, sucks the refrigerant and compresses it to a high-temperature high-pressure gas state, and then conveys it to the refrigerant piping. Regarding the operation control of the compressor 101, the compressor 101 is equipped with, for example, a master-side inverter circuit, a slave-side inverter circuit, and the like, and the capacity (the amount of refrigerant sent out per unit time) of the compressor 101 can be finely changed by arbitrarily changing the operation frequency.

[0037] The oil separator 102 separates lubricant oil that has mixed with the refrigerant and has been discharged from the compressor 101. The separated lubricant oil is returned to the compressor 101. The four-way valve 103 switches the flow of refrigerant according to whether the air-conditioning apparatus is in cooling operation or heating operation, based on instructions from the heat source side control device 110. The heat source side heat exchanger 104 is constituted by the heat exchanger 1 explained in Embodiments 1 to 4, and exchanges heat with the refrigerant and air (outside air). For example, during heating operation, the heat source side heat exchanger 104 functions as an evaporator to exchange heat between a low pressure refrigerant that has flowed in via the heat source side expansion device 107 and air, and evaporates and gasifies the refrigerant. Further, during cooling operation, the heat source side heat exchanger 104 functions as a condenser to exchange heat between a refrigerant that has been compressed in the compressor 101 and that has flowed in from the four-way valve 103 side and air, and condenses and liquefies the refrigerant. The heat source side fan 105 is provided to the heat source side heat exchanger 104 in order to efficiently carry out heat exchange between the refrigerant and air. The heat source side fan 105 can also have an inverter circuit (not illustrated) so that the rotational speed of the fan can be finely changed by arbitrarily changing the operation frequency of the fan motor.

[0038] The refrigerant-to-refrigerant heat exchanger 108 exchanges heat between a refrigerant flowing through the main passage of the refrigerant circuit and the refrigerant that has branched from the main passage and that had its flow regulated by the bypass expansion device (expansion valve) 109. When it is necessary to supercool the refrigerant, particularly during cooling operation, the refrigerant-to-refrigerant heat exchanger 108 supercools the refrigerant and supplies it to the load side unit 200. The refrigerant-to-refrigerant heat exchanger 108 is also constituted by the heat exchanger 1 explained in Embodiments 1 to 4.

[0039] Liquid that flows through the bypass expansion device 109 is returned to the accumulator (liquid separator) 106 via a bypass piping. The accumulator 106 is, for example, a means for retaining excess liquid refrigerant. The heat

source side control device 110 is constituted by, for example, a microcomputer or the like, and can communicate via wires or wirelessly with a load side control device 204. For example, the heat source side control device 110 performs operational control of the entire air-conditioning apparatus by controlling each means in the air-conditioning apparatus, such as operation frequency control of the compressor 101 by inverter circuit control, based on data related to detection of various detection means (sensors) within the air-conditioning apparatus.

[0040] Meanwhile, the load side unit 200 is constituted by a load side heat exchanger 201, an expansion device (expansion valve) 202, a load side fan 203, and the load side control device 204. The load side heat exchanger 201 is also constituted by the heat exchanger 1 explained in Embodiments 1 to 4, and exchanges heat between the refrigerant and air in a space subject to air-conditioning. For example, during heating operation, the load side heat exchanger 201 functions as a condenser, exchanges heat between the refrigerant that has flowed in from the gas piping 300 and air, condenses and liquefies the refrigerant (or turns the refrigerant into a gas-liquid two-phase refrigerant), and discharges it to the liquid piping 400 side. On the other hand, during cooling operation, the load side heat exchanger 201 functions as an evaporator, exchanges heat between the refrigerant that has been turned into a low pressure state by the load side expansion device 202 and air, causes the refrigerant to take away heat from the air and evaporate and gasify itself, and discharges it to the gas piping 300 side. Further, the load side fan 203 for regulating the flow of air that performs heat exchange is provided to the load side unit 200. The operation speed of the load side fan 203 is determined by, for example, a user setting. The load side expansion device 202 regulates the pressure of the refrigerant within the load side heat exchanger 201 by changing the opening degree.

[0041] The load side control device 204 is also constituted by a microcomputer or the like, and can, for example, communicate via wires or wirelessly with the heat source side control device 110. The load side control device 204 controls each device (means) of the load side unit 200 so that, for example, the indoors reaches a certain temperature, based on instructions from the heat source side control device 110 or instructions from a resident or the like. The load side control device 204 also sends signal containing data related to detection of the detection means disposed in the load side unit 200.

[0042] Next, the operation of the air-conditioning apparatus will be explained. First, the basic refrigerant circulation in the refrigerant circuit during cooling operation will be explained. A high-temperature high-pressure gas refrigerant discharged from the compressor 101 by the drive operation of the compressor 101 is condensed while passing through the heat source side heat exchanger 104 via the four-way valve 103, turns into a liquid refrigerant, and flows out of the heat source side unit 100. The refrigerant that has flowed into the load side unit 200 through the liquid piping 400, which is a low-temperature low-pressure liquid refrigerant whose pressure has been controlled by the regulation of the opening degree of the load side expansion device 202, flows through the load side heat exchanger 201, is evaporated, and flows out of the load side heat exchanger 201. The refrigerant then flows into the heat source side unit 100 through the gas piping 300, is sucked into the compressor 101 via the four-way valve 103 and the accumulator 106, and is compressed again and discharged, thereby completing a circulation.

[0043] The basic refrigerant circulation in the refrigerant circuit during heating operation will now be explained. A high temperature, high-pressure gas refrigerant discharged from the compressor 101 by the drive operation of the compressor 101 flows into the load side unit 200 from the four-way valve 103 through the gas piping 300. In the load side unit 200, the pressure of the refrigerant is adjusted by regulation of the opening degree of the load side expansion device 202. The refrigerant is condensed while passing through the load side heat exchanger 201 and flows out of the load side unit 200 being turned into a liquid state or gas-liquid two-phase state at intermediate pressure. The pressure of the refrigerant that has flowed into the heat source side unit 100 through the liquid piping 400 is adjusted by regulation of the opening degree of the heat source side expansion device 107. The refrigerant is made to evaporate while passing through the heat source side heat exchanger 104, turns into a gas refrigerant, is sucked into the compressor 101 via the four-way valve 103 and the accumulator 106, is compressed and discharged as described above, thereby completing a circulation.

[0044] According to the air-conditioning apparatus of Embodiment 5 explained above, the heat source side heat exchanger 104 of the heat source side unit 100, the refrigerant-to-refrigerant heat exchanger 108, and the load side heat exchanger 201 of the load side unit 200 are constituted by the heat exchanger 1 of Embodiments 1 to 4, which has a high heat exchange rate, as an evaporator or a condenser. Therefore, the COP (coefficient of performance: energy efficiency ratio, performance factor) and the like can be improved, and energy can be conserved.

[0045] In Embodiment 5 described above, the application of the heat exchanger according to the present invention in an air-conditioning apparatus was explained. However, the present invention is not limited to such an apparatus, and the invention can be applied to other refrigeration cycle apparatuses that include a heat exchanger that functions as an evaporator and a condenser and that constitutes a refrigerant circuit such as a refrigeration apparatus and heat pump system.

Examples

[0046] Below, examples of the present invention will be explained with comparison to comparative examples that

deviate from the scope of the present invention. As shown in Table 1, heat exchangers 1 were fabricated, wherein the outer diameter is 7 mm, the bottom thickness of the groove 21 is 0.25 mm, the lead angle is 30 degrees, and the number of high threads 22a is 11 or 19 (Examples 1 and 2). Further, as comparative examples, heat exchangers were fabricated, wherein the outer diameter is 7 mm, the bottom thickness of the groove is 0.25 mm, and the number of high threads is 6 or 30 (Comparative Examples 1 and 2).

[0047]

Table 1

	Outer Diameter (mm)	Bottom Thickness (mm)	Lead Angle (degrees)	Thread Number (High Threads) (-)	Heat Exchange Rate (%)
Comparative Example 1	7	0.25	30	6	99
Example 1	7	0.25	30	11	101.3
Example 2	7	0.25	30	19	101
Comparative Example 2	7	0.25	30	30	99.5

[0048] As is clear from Table 1, the heat exchange rate of the heat exchangers 1 in Example 1 and Example 2 was 101.3% and 101%, respectively, and the heat exchange rate of the heat exchangers in Comparative Example 1 and Comparative Example 2 was 99% and 99.5%, respectively. The heat exchangers 1 in Example 1 and Example 2 have a higher heat exchange rate compared to the heat exchangers in both of Comparative Examples 1 and 2, and the heat transfer performance in the pipe showed improvement.

[0049] Next, as shown in Table 2, heat exchangers 1 were fabricated, wherein the outer diameter is 7 mm, the bottom thickness of the grooves 21 is 0.25 mm, the lead angle is 30 degrees, and the ratio ($W1/D$) of the tip width $W1$ of the high threads 22a to the outer diameter D of the heat transfer pipe 20 is 0.011, 0.020, or 0.040 (Examples 3, 4, and 5). Further, as comparative examples, heat exchangers were fabricated, wherein the outer diameter is 7 mm, the bottom thickness of the grooves is 0.25 mm, the lead angle is 30 degrees, and the ratio ($W1/D$) of the tip width of the high threads to the outer diameter of the heat transfer pipe is 0.005 or 0.050 (Comparative Examples 3 and 4).

[0050]

Table 2

	Outer Diameter (mm)	Bottom Thickness (mm)	Lead Angle (degrees)	Tip Width of High Threads to Outer Diameter of Heat Transfer Pipe (-)	Heat Exchange Rate (%)
Comparative Example 3	7	0.25	30	0.005	99.2
Example 3	7	0.25	30	0.011	101.2
Example 4	7	0.25	30	0.02	101.8
Example 5	7	0.25	30	0.04	101
Comparative Example 4	7	0.25	30	0.05	98

[0051] As is clear from Table 2, the heat exchange rate of the heat exchangers 1 in Example 3, Example 4, and Example 5 was 101.2%, 101.8%, and 101%, respectively, and the heat exchange rate of the heat exchangers in Comparative Example 3 and Comparative Example 4 was 99.2% and 98%, respectively. The heat exchangers 1 in Example 3, Example 4, and Example 5 have a higher heat exchange rate compared to the heat exchangers in both of Comparative Examples 3 and 4, and the heat transfer performance within the pipe showed improvement.

[0052] Next, as shown in Table 3, heat exchangers 1 were fabricated, wherein the outer diameter is 7 mm, the bottom thickness of the grooves 21 is 0.25 mm, the lead angle is 30 degrees, and the groove depth after pipe expansion is 0.1

mm or 0.26 mm (Examples 6 and 7). Further, as comparative examples, heat exchangers were fabricated, wherein the outer diameter is 7 mm, the bottom thickness of the grooves is 0.25 mm, the lead angle is 30 degrees, and the groove depth after pipe expansion is 0.05 mm or 0.3 mm (Comparative Examples 5 and 6).

[0053]

Table 3

	Outer Diameter (mm)	Bottom Thickness (mm)	Lead Angle	Groove Depth After Pipe Expansion (mm)	Heat Exchange Rate (%)
Comparative Example 5	7	0.25	30	0.05	99
Example 6	7	0.25	30	0.1	101.5
Example 7	7	0.25	30	0.26	101.2
Comparative Example 6	7	0.25	30	0.3	99.4

[0054] As is clear from Table 3, the heat exchange rate of the heat exchangers 1 in Example 6 and Example 7 was 101.5% and 101.2%, respectively, and the heat exchange rate of the heat exchangers in Comparative Example 5 and Comparative Example 6 was 99% and 99.4%, respectively. The heat exchangers 1 in Example 6 and Example 7 have a higher heat exchange rate compared to the heat exchangers in both of Comparative Examples 5 and 6, and the heat transfer performance in the pipe showed improvement.

[0055] Next, as shown in Table 4, heat exchangers 1 were fabricated, wherein the outer diameter is 7 mm, the bottom thickness of the grooves 21 is 0.25 mm, the apex angle is 30 degrees, and the lead angle γ is 10 degrees, 30 degrees, or 50 degrees (Examples 8, 9, and 10). Further, as comparative examples, heat exchangers were fabricated, wherein the outer diameter is 7 mm, the bottom thickness of the grooves is 0.25 mm, the apex angle is 30 degrees, and the lead angle is 5 degrees or 60 degrees (Comparative Examples 7 and 8).

[0056]

Table 4

	Outer Diameter (mm)	Bottom Thickness (mm)	Apex Angle (degrees)	Lead Angle (degrees)	Heat Exchange Rate (%)
Comparative Example 7	7	0.25	30	5	99.2
Example 8	7	0.25	30	10	100.9
Example 9	7	0.25	30	30	101.5
Example 10	7	0.25	30	50	100.8
Comparative Example 8	7	0.25	30	60	99.5

[0057] As is clear from Table 4, the heat exchange rate of the heat exchangers 1 in Example 8, Example 9, and Example 10 was 100.9%, 101.5%, and 101.8%, respectively, and the heat exchange rate of the heat exchangers in Comparative Example 7 and Comparative Example 8 was 99.2% and 99.5%, respectively. The heat exchangers 1 in Example 8, Example 9, and Example 10 have a higher heat exchange rate compared to the heat exchangers in both of Comparative Examples 7 and 8, and the heat transfer performance within the pipe showed improvement.

[0058] Next, as shown in Table 5, heat exchangers 1 were fabricated, wherein the outer diameter is 7 mm, the bottom thickness of the grooves 21 is 0.25 mm, the lead angle is 30 degrees, and the apex angle α is 15 degrees or 30 degrees (Examples 11 and 12). Further, as comparative examples, heat exchangers were fabricated, wherein the outer diameter is 7 mm, the bottom thickness is 0.25 mm, the lead angle is 30 degrees, and the apex angle is 10 degrees or 40 degrees (Comparative Examples 9 and 10).

[0059]

Table 5

	Outer Diameter (mm)	Bottom Thickness (mm)	Lead Angle (degrees)	Apex Angle (degrees)	Heat Exchange Rate (%)
Comparative Example 9	7	0.25	30	10	99
Example 11	7	0.25	30	15	101
Example 12	7	0.25	30	30	101.3
Comparative Example 10	7	0.25	30	40	99.3

[0060] As is clear from Table 5, the heat exchange rate of the heat exchangers 1 in Example 11 and Example 12 was 101% and 101.3%, respectively, and the heat exchange rate of the heat exchangers in Comparative Example 9 and Comparative Example 10 was 99% and 99.3%, respectively. The heat exchangers 1 in Example 11 and Example 12 have a higher heat exchange rate compared to the heat exchangers in both of Comparative Examples 9 and 10, and the heat transfer performance within the pipe showed improvement.

Reference Signs List

[0061] 1 ... heat exchanger; 10 ... fin; 11 ... through hole; 20...heat transfer pipe; 21 ... groove portion; 22...thread portion; 22a...high thread (thread portion); 22b ... low thread (thread portion); 30...pipe expanding ball; 31 ... rod; 100...heat source side unit; 101 ... compressor; 102 ... oil separator; 103 ... four-way valve; 104...heat source side heat exchanger; 105...heat source side fan; 106... accumulator; 107...heat source side expansion device; 108... refrigerant-to-refrigerant heat exchanger; 109... bypass expansion device; 110...heat source side control device; 200...load side unit; 201...load side heat exchanger; 202...load side expansion device; 203...load side fan; 204 ... load side control device; 300...gas piping; 400...liquid piping; α ...apex angle of high thread; β ... apex angle of low thread; γ ... direction in which the thread portion extend relative to pipe axial direction (lead angle); D...outer diameter of heat transfer pipe; H...height of high thread after pipe expansion; R1...curvature radius of crest of low thread; W1...tip width of crest portion of high thread after pipe expansion.

Claims

1. A heat transfer pipe for a heat exchanger, comprising:

high threads; and
low threads that are lower than the high threads, wherein
the high threads and the low threads are provided at respective prescribed heights in a helical manner in a pipe axial direction on a pipe inner surface,
the high threads are formed with 11 to 19 threads, the low threads are formed of 3 to 6 threads between each pair of the high threads,
the high threads before expansion of the pipe each have a trapezoidal shape in a cross-section such that a crest thereof is flat, and a ratio of a tip width of the crest portion after the expansion of the pipe to an outer diameter of the heat transfer pipe is 0.011 to 0.040.

2. The heat transfer pipe for a heat exchanger of claim 1, wherein the high threads before the expansion of the pipe are higher by 0.04 mm or more than the low threads.

3. The heat transfer pipe for a heat exchanger of claim 1 or 2, wherein in each of the high threads before the expansion of the pipe, each curvature radius of the flat surface of the crest portion and both side surfaces is 0.01 mm to 0.03 mm.

4. The heat transfer pipe for a heat exchanger of any of claims 1 to 3, wherein an apex angle of the high threads before the expansion of the pipe is 15 to 30 degrees, and an apex angle of the low threads is 5 to 15 degrees.

5. The heat transfer pipe for a heat exchanger of any of claims 1 to 4, wherein a curvature radius of crests of the low threads is 0.03 mm to 0.045 mm.

6. The heat transfer pipe for a heat exchanger of any of claims 1 to 5, wherein a direction in which the threads extend relative to a pipe axial direction makes 10 to 50 degrees.

7. A heat exchanger, comprising:

a plurality of fins for exchanging heat; and
the heat transfer pipe of any of claims 1 to 6 that penetrates the fins, wherein
the heat transfer pipe is pressurized from an inner surface to be expanded, and is fixed to the fins.

8. The heat exchanger of claim 7, wherein a height of the high threads after the expansion of the pipe is 0.10 mm to 0.26 mm.

9. A refrigeration cycle apparatus, comprising:

a refrigerant circuit including a compressor that compresses a refrigerant, a condenser that condenses the refrigerant by heat exchange, an expansion means for decompressing the condensed refrigerant, and an evaporator that evaporates the decompressed refrigerant by heat exchange to circulate the refrigerant, the refrigerant circuit being connected by piping, the refrigeration cycle apparatus, wherein
the heat exchanger of claim 7 or 8 is provided to both or either one of the condenser and the evaporator.

10. The refrigeration cycle apparatus of claim 9, wherein as the refrigerant, either one of an HC single refrigerant, a mixed refrigerant including HC, R32, R410A, R407C, tetrafluoropropene, a non-azeotropic refrigerant mixture consisting of HFC refrigerant having a boiling point that is lower than the tetrafluoropropene, or carbon dioxide is used.

11. An air-conditioning apparatus, comprising the refrigeration cycle apparatus of claim 9 or 10 that carries out cooling or heating of a conditioned space.

FIG. 1

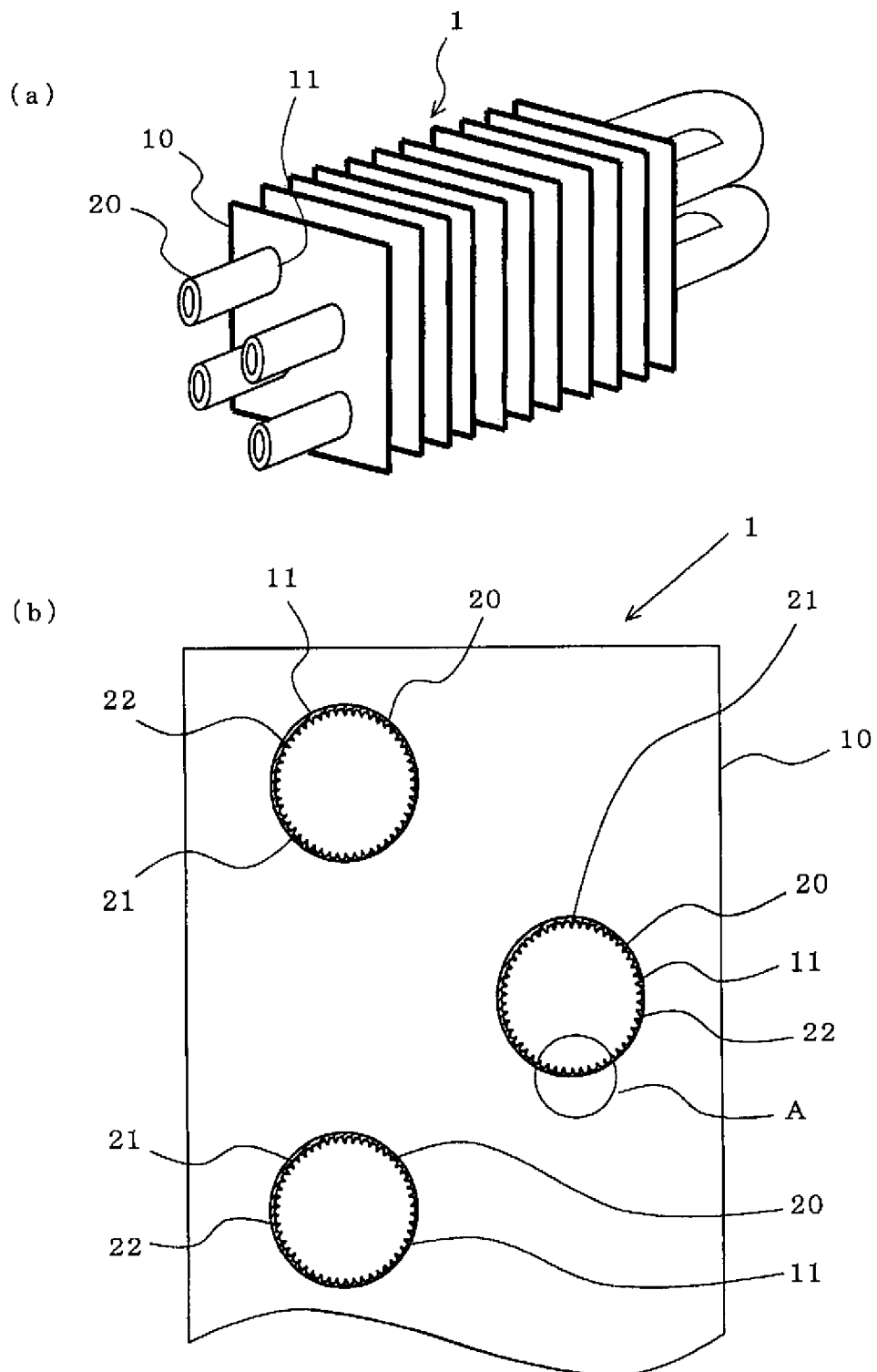


FIG. 2

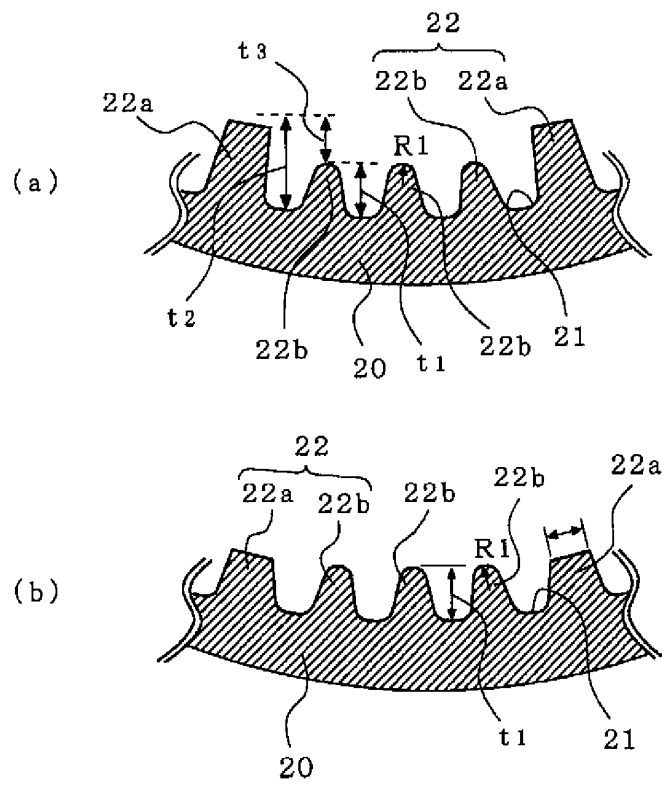


FIG. 3

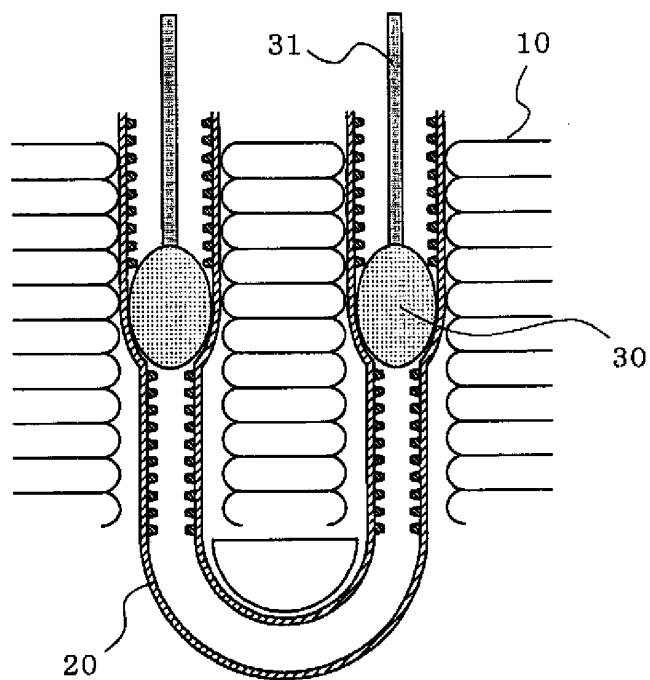


FIG. 4

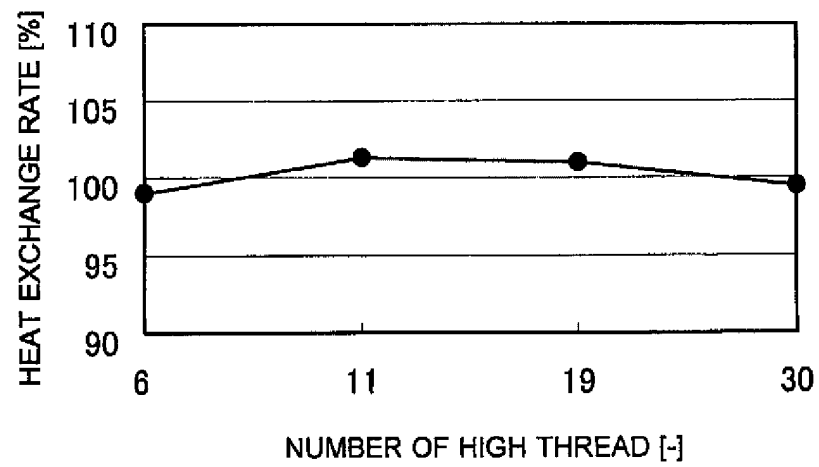


FIG. 5

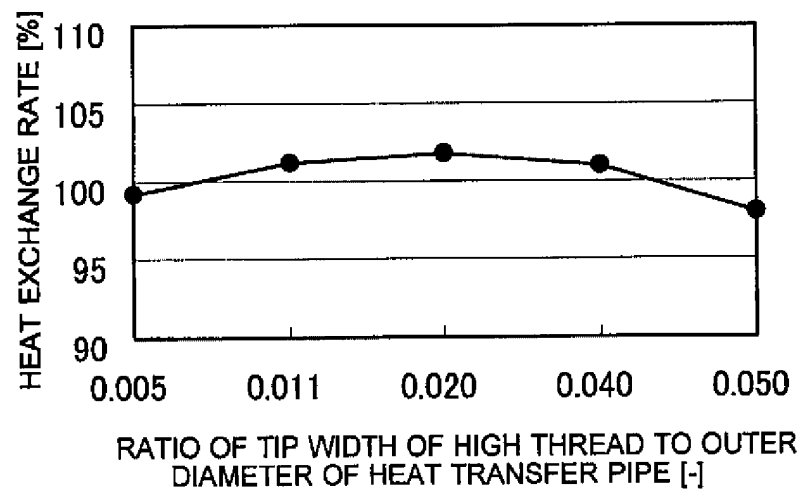


FIG. 6

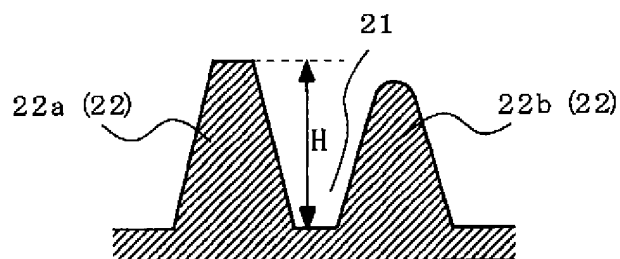


FIG. 7

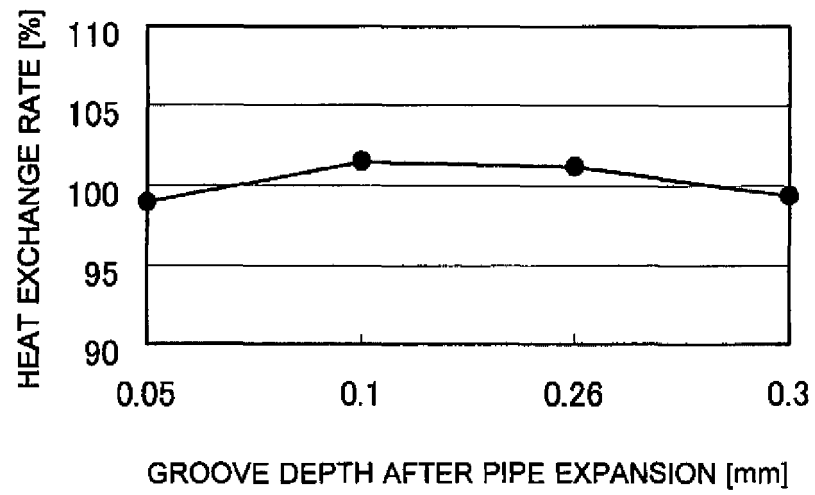


FIG. 8

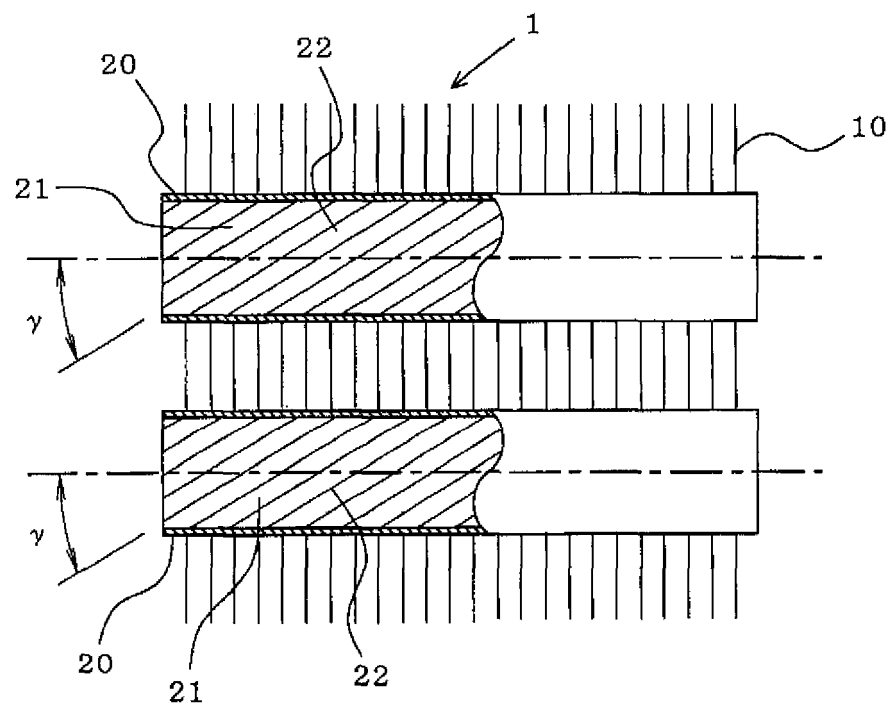


FIG. 9

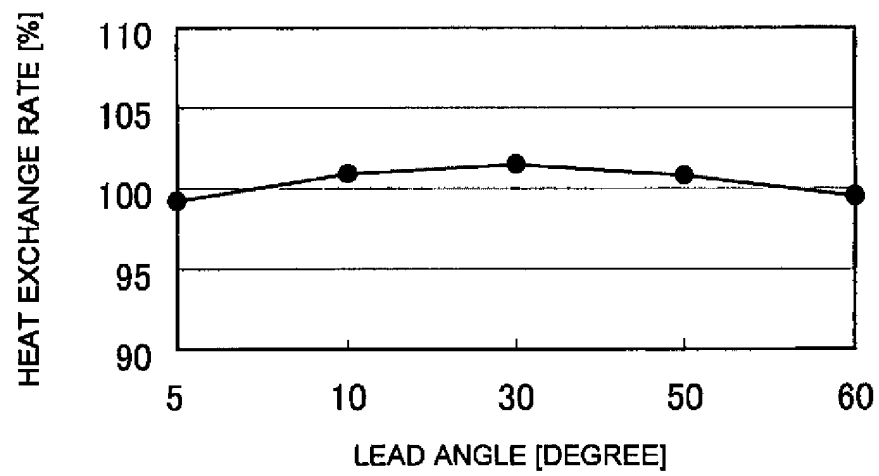


FIG. 10

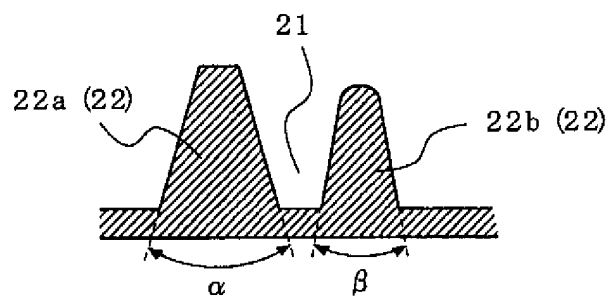


FIG. 11

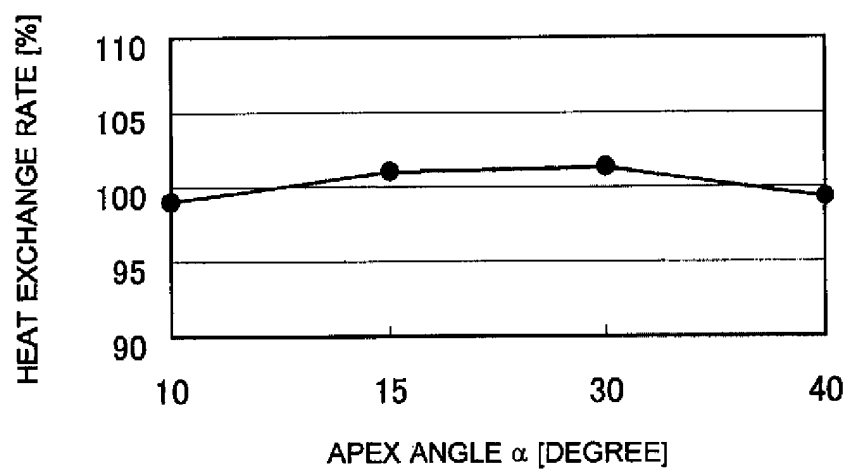
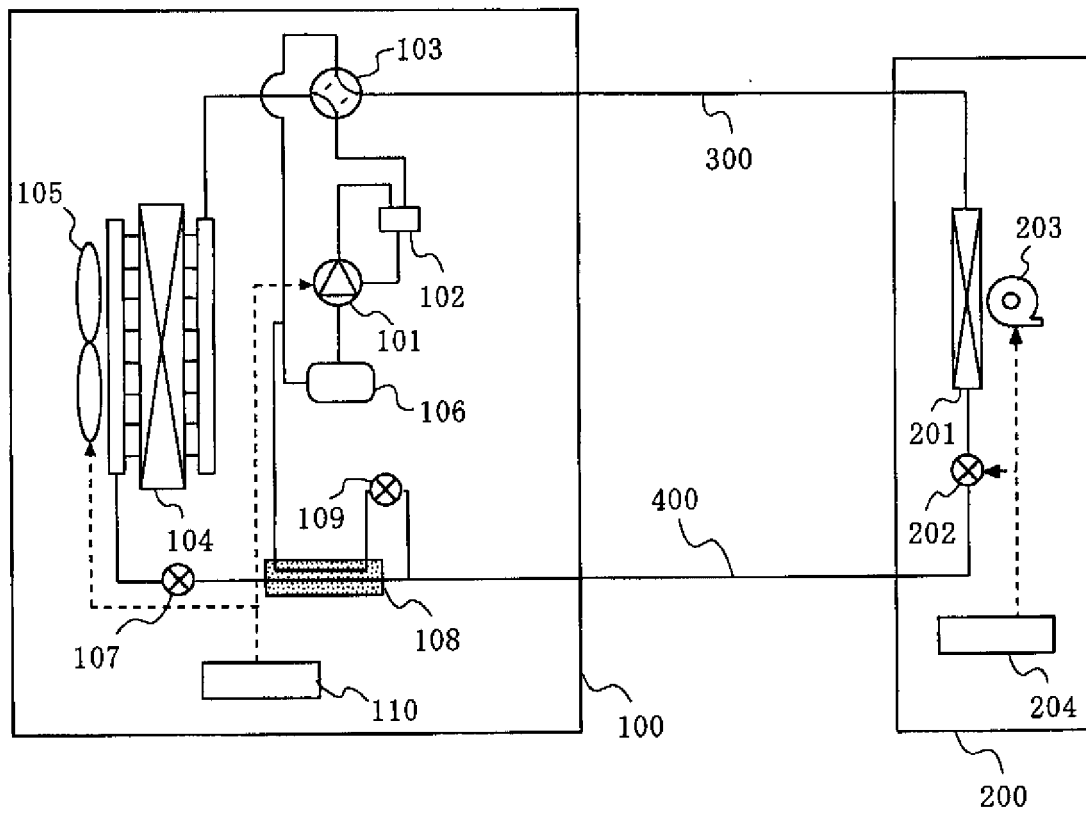


FIG. 12



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2011/000048

A. CLASSIFICATION OF SUBJECT MATTER

F28F1/40 (2006.01) i, F28F1/32 (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)

F28F1/40, F28F1/32

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	1996-2011
Kokai Jitsuyo Shinan Koho	1971-2011	Toroku Jitsuyo Shinan Koho	1994-2011

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	JP 2001-133182 A (Sumitomo Light Metal Industries, Ltd.), 18 May 2001 (18.05.2001), entire text; all drawings (particularly, fig. 1 to 3) (Family: none)	1-11
Y	JP 60-28573 B1 (Daikin Industries, Ltd.), 05 July 1985 (05.07.1985), entire text; all drawings (Family: none)	1-11
Y	WO 2009/131072 A1 (Mitsubishi Electric Corp.), 29 October 2009 (29.10.2009), entire text in the front part; all drawings (particularly, paragraph [0020]; fig. 15) (Family: none)	1-11

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

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Date of the actual completion of the international search
26 January, 2011 (26.01.11)Date of mailing of the international search report
08 February, 2011 (08.02.11)Name and mailing address of the ISA/
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INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2011/000048

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	JP 2009-186130 A (The Furukawa Electric Co., Ltd.), 20 August 2009 (20.08.2009), entire text; all drawings (particularly, paragraph [0001]; fig. 1 to 4) (Family: none)	5
A	JP 2005-526945 A (Trefimetaux), 08 September 2005 (08.09.2005), entire text; all drawings (particularly, fig. 1 to 3) & US 2003/0173071 A1 & EP 1851498 A & WO 2003/076861 A1 & FR 2837270 A & CA 2474558 A & BR 308372 A & HR 20040819 A & NO 20044299 A & PL 370690 A & IL 162942 D & CN 1636128 A & RU 2289076 C & ZA 200405864 A & IL 162942 A	1-11

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- JP 60142195 A [0005]