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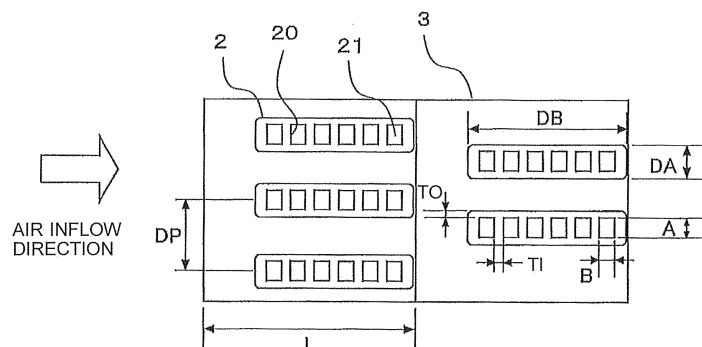
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(54) **REFRIGERATION CYCLE APPARATUS**

(57) In a refrigeration cycle apparatus, HFO-1123 or a refrigerant mixture containing HFO-1123 is used as refrigerant, at least one of a first or second heat exchanger includes a plurality of fins and a plurality of flat tubes, the plurality of flat tubes each include a plurality of pas-

sages through which the refrigerant flows, and, where the plurality of flat tubes each have a thickness DA, a relationship expressed by $0.9 \text{ mm} \leq DA \leq 3.0 \text{ mm}$ is satisfied.

FIG. 4



Description

Technical Field

[0001] The present invention relates to a refrigeration cycle apparatus including heat exchangers.

Background Art

[0002] A conventional refrigeration cycle apparatus has been proposed in which HFO-1123 is used as working fluid (see, for example, Patent Literature 1).

Citation List

Patent Literature

[0003] Patent Literature 1: Japanese Unexamined Patent Application Publication No. 2014-098166

Summary of Invention

Technical Problem

[0004] Refrigerant containing HFO refrigerant as in HFO-1123 of Patent Literature 1 conventionally has a low global warming potential (hereinafter referred to as "GWP"), and therefore, has a short atmospheric lifetime. That is, due to poor stability, the refrigerant is easily decomposed even in the case where the refrigerant is in a refrigerant circuit including heat exchangers. Consequently, a reduction in performance of the refrigeration cycle apparatus and a reduction in reliability of the refrigeration cycle apparatus are caused.

[0005] The present invention has been made in view of the above-described problem, and is intended to provide a refrigeration cycle apparatus having improved heat transfer performance of a heat exchanger in the case of using HFO-1123 or a refrigerant mixture containing HFO-1123 as refrigerant circulating in a refrigerant circuit.

Solution to Problem

[0006] A refrigeration cycle apparatus of an embodiment of the present invention includes a refrigerant circuit including a compressor, a first heat exchanger, an expansion unit, and a second heat exchanger connected in sequence to each other via pipes to circulate refrigerant through the refrigerant circuit. The refrigerant is one of HFO-1123 and a refrigerant mixture containing HFO-1123. At least one of the first heat exchanger and the second heat exchanger includes a plurality of fins and a plurality of flat tubes. The plurality of flat tubes each include a plurality of passages through which the refrigerant flows. Where the plurality of flat tubes each have a thickness DA, a relationship expressed by $0.9 \text{ mm} \leq \text{DA} \leq 3.0 \text{ mm}$ is satisfied.

[0007] According to the embodiment of the present invention, the relationship of $0.9 \text{ mm} \leq \text{DA} \leq 3.0 \text{ mm}$ is satisfied, where the thickness of the flat tube is represented by DA. Thus, heat transfer performance of the heat exchanger can be improved in the case of using HFO-1123 or a refrigerant mixture containing HFO-1123 as the refrigerant circulating in the refrigerant circuit.

vention, the relationship of $0.9 \text{ mm} \leq \text{DA} \leq 3.0 \text{ mm}$ is satisfied, where the thickness of the flat tube is represented by DA. Thus, heat transfer performance of the heat exchanger can be improved in the case of using HFO-1123 or a refrigerant mixture containing HFO-1123 as the refrigerant circulating in the refrigerant circuit.

Brief Description of Drawings

[0008]

[Fig. 1] Fig. 1 is a circuit diagram of an example of a refrigerant circuit of a refrigeration cycle apparatus of Embodiment 1 of the present invention.

[Fig. 2] Fig. 2 is a perspective view of an example of heat exchangers of the refrigeration cycle apparatus of Embodiment 1 of the present invention.

[Fig. 3] Fig. 3 is a cross-sectional view of an example of a flat tube of the refrigeration cycle apparatus of Embodiment 1 of the present invention.

[Fig. 4] Fig. 4 is a cross-sectional view of an example of the heat exchanger of the refrigeration cycle apparatus of Embodiment 1 of the present invention.

[Fig. 5] Fig. 5 is a characteristic graph of a relationship between a short-axis diameter DA of the flat tube and a coefficient of performance (COP) in the refrigeration cycle apparatus of Embodiment 1 of the present invention.

[Fig. 6] Fig. 6 is a cross-sectional view of an example of a flat tube of a refrigeration cycle apparatus of Embodiment 2 of the present invention.

[Fig. 7] Fig. 7 is an enlarged cross-sectional view of an example of the flat tube of the refrigeration cycle apparatus of Embodiment 2 of the present invention.

[Fig. 8] Fig. 8 is a cross-sectional view of an example of the case where sludge is accumulated between protrusions when a plurality of protrusions are provided in the flat tube.

[Fig. 9] Fig. 9 is a cross-sectional view of an example of the flat tube of the refrigeration cycle apparatus of Embodiment 2 of the present invention.

[Fig. 10] Fig. 10 is an enlarged cross-sectional view of the example of the flat tube of the refrigeration cycle apparatus of Embodiment 2 of the present invention.

[Fig. 11] Fig. 11 is a characteristic graph of a relationship between pressure resistance performance and a ratio between a wall thickness TO and a short-axis diameter DA of a flat tube of a refrigeration cycle apparatus of Embodiment 3 of the present invention.

Description of Embodiments

[0009] Embodiments of the present invention are described hereinafter with reference to the drawings.

[0010] Note that size relationships among components in the drawings described below may be different from actual size relationships. Moreover, the same reference

signs in the drawings described below are used to represent the same or equivalent elements. The reference signs are common in the entire text of the specification. Further, the form of the components described across the entire text of the specification is merely an example, and is not limited to such description.

[0011] Note that, in the embodiments described below, an air-conditioning apparatus is described as an example of a refrigeration cycle apparatus of the present invention, but the present invention is not limited to the air-conditioning apparatus. For example, the refrigeration cycle apparatus of the present invention is applicable to other apparatuses having heat exchangers, such as a refrigeration apparatus and a hot water dispenser.

Embodiment 1

[0012] Fig. 1 is a circuit diagram of an example of a refrigerant circuit of a refrigeration cycle apparatus of Embodiment 1 of the present invention.

[0013] As illustrated in Fig. 1, a refrigeration cycle apparatus 10 includes a refrigerant circuit including a compressor 11, a first heat exchanger 12, an expansion unit 13, and a second heat exchanger 14 connected in sequence to each other via pipes to circulate refrigerant through the refrigerant circuit. Moreover, the first heat exchanger 12 is provided with a fan 15a configured to send air, and the second heat exchanger 14 is provided with a fan 15b configured to send air.

[0014] Note that, in description below, the first heat exchanger 12 and the second heat exchanger 14 are each referred to as a "heat exchanger 1" when the first heat exchanger 12 and the second heat exchanger 14 are not distinguished from each other.

[0015] As refrigerant circulating in the refrigerant circuit, 1,1,2-trifluoroethylene (HFO-1123) having a single double bond in a molecular structure or a refrigerant mixture containing HFO-1123 is used.

[0016] In a conventional refrigeration cycle apparatus, a hydrofluorocarbon (HFC) refrigerant such as HFC-410A is used as working fluid. Unlike hydrochlorofluorocarbon (HCFC) refrigerants such as HCFC-22, HFC-410A has zero ozone depletion potential (ODP), which is an indication of influence on the ozone layer of the stratosphere, and therefore, does not damage the ozone layer. However, HFC refrigerants have the probability of causing global warming, and have a high GWP value, which is an indication of such a probability.

[0017] Of HFC refrigerants having no carbon double bond in a composition, refrigerant having a low GWP includes HFC-32 having a lower GWP than that of HFC-410A. However, in the case where HFC-32 is used as working fluid of the refrigeration cycle apparatus 10, a discharge temperature at an outlet of the compressor 11 is higher than that of the conventional refrigerant, and for this reason, for example, a material resistant to a high temperature needs to be used.

[0018] Moreover, the refrigerant having a low GWP in-

cludes hydrofluoroolefin (HFO) having a carbon double bond in a composition and readily decomposed by an OH radical in the atmosphere. Examples of such HFO refrigerant include HFO-1234yf and HFO-1234ze. However, these refrigerants have a high standard boiling point, and thus, the pressure of these refrigerants is lower than that at a saturation temperature equivalent to that in the case of the conventional refrigerant. That is, a refrigerant density is lower than that of the conventional HFC refrigerant, and the frequency of the compressor 11 needs to be increased to maintain the flow rate of refrigerant circulating in the refrigeration cycle apparatus 10 at a flow rate equivalent to that of the conventional refrigerant. Thus, power consumption increases, leading to a lower energy saving performance.

[0019] As described above, HFO-1123 or a refrigerant mixture containing HFO-1123 is used as the refrigerant circulating in the refrigerant circuit in Embodiment 1. Unlike HFO-1234yf and HFO-1234ze described above, HFO-1123 has a low standard boiling point, and thus, the pressure of the refrigerant is higher than that at the saturation temperature equivalent to that in the case of the conventional HFC refrigerant. That is, a refrigerant density is higher than that of the conventional refrigerant, and the flow rate of refrigerant circulating in the refrigeration cycle apparatus 10 can be maintained at the flow rate equivalent to that of the conventional refrigerant even when the frequency of the compressor 11 is decreased. Thus, among other types of HFO refrigerants, HFO-1123 has excellent energy saving performance. Moreover, HFO-1123 has greater latent heat than that of the conventionally-used HFC-410A, and thus, the flow rate of refrigerant circulating in the refrigeration cycle apparatus 10 can be decreased. That is, the refrigeration cycle apparatus 10 in which HFO-1123 is used as working fluid is capable of reducing a pressure loss in a flat tube 2 (a heat transfer pipe) in the heat exchanger 1 and producing a heat transfer (cooling energy) capacity equivalent to that in the case of using the conventional refrigerant.

[0020] The compressor 11 is configured to compress refrigerant discharged from the second heat exchanger 14 to supply high-temperature high-pressure refrigerant to the first heat exchanger 12. The first heat exchanger 12 acts as a condenser to exchange heat between air supplied from the fan 15a and the refrigerant, thereby condensing and liquefying the refrigerant. The expansion unit 13 includes an expansion valve, a capillary tube, a pressure reduction device, or an expansion device, for example. The expansion unit 13 is configured to expand the refrigerant discharged from the first heat exchanger 12 into low-temperature low-pressure refrigerant to supply the refrigerant to the second heat exchanger 14. The second heat exchanger 14 acts as an evaporator to exchange heat between air supplied from the fan 15b and the refrigerant, thereby evaporating and gasifying the refrigerant.

[0021] Note that a flow switching device such as a four-

way valve may be provided to switch a flow direction of refrigerant in the refrigerant circuit to cause the first heat exchanger 12 to act as the evaporator and the second heat exchanger 14 to act as the condenser.

[0022] Fig. 2 is a perspective view of an example of the heat exchanger of the refrigeration cycle apparatus of Embodiment 1 of the present invention.

[0023] In Fig. 2, the heat exchanger 1 includes a plurality of flat tubes 2, a plurality of fins 3, and a pair of headers 4a and 4b. The plurality of fins 3 are arranged at intervals to allow fluid such as air to flow through the intervals. The plurality of flat tubes 2 are inserted into the plurality of fins 3. A direction along a longitudinal dimension of a flat cross section of each of the plurality of flat tubes 2 (hereinafter referred to as a "long-axis direction") is in a direction of air flowing between adjacent ones of the fins 3, and the plurality of flat tubes 2 are arranged at intervals in a direction along a thickness dimension of the flat cross section (hereinafter referred to as a "short-axis direction").

[0024] The headers 4a and 4b extend in the vertical direction, and are arranged to face each other. The headers 4a and 4b are each connected to a corresponding one of both ends of each flat tube 2. Refrigerant having flowed into the header 4a is branched in the header 4a, and then, flows into each flat tube 2. The refrigerant circulating in a refrigerant passage of each flat tube 2 exchanges heat with air flowing between adjacent ones of the plurality of fins 3 and between adjacent ones of the plurality of flat tubes 2, and then, flows into the header 4b. The refrigerant having flowed into the header 4b is joined in the header 4b and then flows out from the header 4b.

[0025] Note that the headers 4a and 4b are illustrated as being elongated in the vertical direction, and refrigerant flows toward the flat tubes 2 in the horizontal direction in the drawing; however, the direction of the headers 4a and 4b is not limited to such a direction. For example, the headers 4a and 4b may be placed in a landscape orientation, and refrigerant may flow in the vertical direction.

[0026] Note that a branched pipe for branching refrigerant may be provided instead of at least one of the headers 4a and 4b.

[0027] The fins 3 are made of aluminum, for example. Each fin 3 is formed into an arbitrary shape by, for example, cutting an aluminum bar into a predetermined size and then pressing or processing the aluminum bar. Note that the material of the fin 3 is not limited to aluminum, and an arbitrary material such as copper can be used. Moreover, the shape of the fin 3 may be an arbitrary shape such as a plate-shaped plate fin and a corrugated fin formed in a wave shape. Further, the fin 3 may be formed in a shape having improved heat exchange performance by, for example, cutting and raising processing or formation of a recessed-raised portion.

[0028] Fig. 3 is a cross-sectional view of an example of the flat tube of the refrigeration cycle apparatus of Em-

bodiment 1 of the present invention.

[0029] As illustrated in Fig. 3, the flat tube 2 has a flat cross section. The flat tube 2 includes a flat perforated tube, and a plurality of passages 21 divided by partition walls 20 are formed along the longitudinal direction of the flat tube 2. The flat tube 2 is made of aluminum, for example.

[0030] Note that, for example, aluminum is used as the materials of the fin 3 and the flat tube 2, and thus, corrosion due to contact between different types of metal can be prevented. Moreover, brazing can be used for bonding the fins 3 and the flat tubes 2. Note that, in addition to brazing, an adhesive may be used for bonding the fins 3 and the flat tubes 2. In the case where the fins 3 and the flat tubes 2 are bonded together by brazing using a brazing material such as an aluminum brazing material, adhesion among the fins 3 and the flat tubes 2 is better than that in the case of using the adhesive. Moreover, bonding the fins 3 and the flat tubes 2 by brazing can, as compared to a tube expansion method in which bonding is made by expansion of the tube inserted into the fin 3, provide the heat exchanger 1 having a higher heat transfer performance and excellent heat exchange performance.

[0031] Fig. 4 is a cross-sectional view of an example of the heat exchanger of the refrigeration cycle apparatus of Embodiment 1 of the present invention.

[0032] In the example of Fig. 4, the flat tubes 2 are arranged in two columns in a column direction, which is the direction of air flow. Moreover, the plurality of flat tubes 2 are arranged in a row direction perpendicular to the column direction. Further, the flat tubes 2 of adjacent columns do not overlap with each other in the row direction (for example, a zig-zag pattern). Air flows from a side of parts of the fins 3 at an upstream side of the flat tubes 2.

[0033] Each flat tube 2 has the thickness dimension (hereinafter referred to as a "short-axis diameter DA") in the flat cross section of 2.0 mm and the longitudinal dimension (hereinafter referred to as a "long-axis diameter DB") in the flat cross section of 19.0 mm, for example. In each flat tube 2, the number of holes of the passages 21 is 20 and the thickness T1 of the partition wall 20 of the passage 21 is 0.20 mm, for example. A wall thickness TO between an outer wall of the flat tube 2 and an inner wall of the passage 21 is 0.30 mm. Moreover, a row pitch DP, which is a distance in the row direction between the centers of adjacent ones of the flat tubes 2, is 13.6 mm.

[0034] Each fin 3 has a width L in the direction of air flow of 22 mm. The heat exchanger 1 has a height H_HEX of 600 mm and a fin stack width H_W of 850 mm (see Fig. 2).

[0035] Note that these dimensions are merely an example, and the present invention is not limited to these dimensions as long as a shape of each flat tube 2 satisfies later-described dimension conditions.

[0036] Fig. 5 is a characteristic graph of a relationship between the short-axis diameter DA of the flat tube and a coefficient of performance (COP) in the refrigeration cycle apparatus of Embodiment 1 of the present inven-

tion.

[0037] Fig. 5 shows a result obtained by simulation of the coefficient of performance (COP) for each short-axis diameter DA using the short-axis diameter DA of the flat tube 2 as a parameter when the heat exchanger 1 is used as the evaporator as an example. Moreover, Fig. 5 shows comparison between the case of using HFC-410A as the refrigerant circulating in the refrigerant circuit and the case of using HFO-1123 as the refrigerant circulating in the refrigerant circuit.

[0038] In the result of the simulation shown in Fig. 5, an aspect ratio B/A, which is a ratio between the inner length B of the passages 21 in the long-axis direction of the flat tube 2 and the inner length A of the passages 21 in the short-axis direction of the flat tube 2, is a preset constant value. For example, the aspect ratio B/A is equivalent to that of the above-described dimension example. That is, in the above-described dimension example, the inner length B of the passages 21 is 0.73 mm, and the inner length A of the passages 21 is 1.4 mm. Thus, the aspect ratio B/A is 0.52. Moreover, a ratio between the row pitch DP of the flat tubes 2 and the short-axis diameter DA of the flat tube 2 is a preset constant value. For example, such a ratio is equivalent to that of the above-described dimension example. That is, DP/DA is 6.8, for example. Further, a ratio between the wall thickness TO and the short-axis diameter DA of the flat tube 2 is a preset constant value. For example, such a ratio is equivalent to that of the above-described dimension example. That is, TO/DA is 0.15, for example. In addition, the long-axis diameter DB of the flat tube 2 and the height H_HEX and the fin stack width H_W of the heat exchanger 1 are equivalent to those of the above-described dimension example.

[0039] Moreover, in the result of the simulation shown in Fig. 5, the amount of heat transferred from the fins 3 and the flat tube 2 (a received heat amount) is constant. Further, other parameters of the heat transfer pipe and the heat exchanger 1 than above, such as the pitch of the fins 3, the number of holes, and a path configuration of the heat exchanger 1, and operation conditions in a refrigeration cycle are values when the coefficient of performance is substantially optimal.

[0040] As shown in Fig. 5, air passage resistance between adjacent ones of the flat tubes 2 decreases as the short-axis diameter DA of the flat tube 2 decreases. Since the row pitch DP/the short-axis diameter DA is the preset constant value, the flat tubes 2 can be more densely mounted as the short-axis diameter DA of the flat tube 2 decreases. Further, since the aspect ratio B/A and the wall thickness TO/the short-axis diameter DA are the preset constant values, the number of holes of the passages 21 in the flat tube 2 increases as the short-axis diameter DA of the flat tube 2 decreases. Thus, an area is expanded, in which refrigerant in the flat tube 2 contacts a heat transfer surface, thereby leading to efficient heat exchange. Thus, performance of the heat exchanger 1 is improved, and the coefficient of performance is also im-

proved.

[0041] Note that, when the short-axis diameter DA of the flat tube 2 is extremely small, the cross-sectional area of each passage 21 of the flat tube 2 is narrowed, and thus, a pressure loss in the tube increases. To suppress an increase in the pressure loss in the tube, the number of paths of the heat exchanger 1 needs to be increased. However, in the case of branching refrigerant, the refrigerant needs to be equally distributed by the headers 4a and 4b for sufficient performance. However, when the short-axis diameter DA of the flat tube 2 is extremely small, refrigerant is difficult to be equally distributed. Moreover, the upper limit of the number of paths is determined depending on the size of the heat exchanger 1. For this reason, when the short-axis diameter DA decreases, an increase in the pressure loss in the tube cannot be suppressed by an increase in the number of paths. Thus, when the short-axis diameter DA of the flat tube 2 is extremely small, the performance of the heat exchanger 1 is lowered, and the coefficient of performance is also lowered.

[0042] In addition, when the short-axis diameter DA of the flat tube 2 falls below 0.8 mm, extruding and other manufacturing of the flat tube 2 become difficult.

[0043] On the other hand, when the short-axis diameter DA of the flat tube 2 is extremely large, air passage resistance between adjacent ones of the flat tubes 2 increases. Moreover, the number of holes of the passages 21 in the flat tube 2 decreases. For this reason, the area is narrowed, in which refrigerant in the flat tube 2 contacts the heat transfer surface, thereby decreasing heat exchange efficiency. Thus, sufficient heat exchange performance is difficult to be obtained and the coefficient of performance is also lowered.

[0044] As shown in Fig. 5, in comparison between the types of refrigerant, the pressure loss in the flat tube 2 is smaller in the case of using HFO-1123 as refrigerant flowing through the flat tube 2 than in the case of HFC-410A. Thus, even for the same short-axis diameter DA, the number of paths can be reduced, and the coefficient of performance is improved. In particular, in the case where the short-axis diameter DA is $0.9 \text{ mm} \leq DA \leq 3.0 \text{ mm}$, the coefficient of performance in the case of using HFO-1123 as refrigerant is substantially equal to or higher than a peak value of the coefficient of performance in the case of HFC-410A. Thus, sufficient performance can be exhibited.

[0045] In the case where the short-axis diameter DA of the flat tube 2 is 0.9 mm, when the aspect ratio B/A is 0.52 and the wall thickness TO/the short-axis diameter DA is 0.15 as in the above-described dimension example, the inner length B of the passage 21 in the long-axis direction of the flat tube 2 is 0.33 mm, for example. In this case, the size of sludge caused due to chemical reaction of a decomposition product of HFO-1123 is about 0.15 mm. Thus, in the case where the inner length B of the passage 21 is 0.33 mm, even when sludge is caused due to decomposition of HFO-1123 flowing through the

passage 21, the sludge can flow without being accumulated in the passage 21. Consequently, sufficient reliability can be ensured.

[0046] Thus, the short-axis diameter DA of the flat tube 2 is set to satisfy a relationship of $0.9 \text{ mm} \leq \text{DA} \leq 3.0 \text{ mm}$. Consequently, heat transfer performance of the heat exchanger 1 can be improved in the case of using HFO-1123 or the refrigerant mixture containing HFO-1123 as refrigerant circulating in the refrigerant circuit, and a reduction in reliability can be suppressed even when sludge is caused due to refrigerant decomposition.

[0047] Note that, when a smaller value is selected as the aspect ratio B/A, the inner length B of the passage 21 in the long-axis direction of the flat tube 2 is not sufficient, and the pressure loss in the tube increases. For this reason, in the case of a small short-axis diameter DA as described above, such a diameter is typically increased depending on the aspect ratio to such an extent that pressure resistance performance is maintained.

[0048] As shown in Fig. 5, when the short-axis diameter DA of the flat tube 2 is $1.0 \text{ mm} \leq \text{DA} \leq 1.8 \text{ mm}$, the coefficient of performance in the case of using HFO-1123 as refrigerant flowing through the flat tube 2 is within 2% of a maximum value. Thus, the short-axis diameter DA of the flat tube 2 is set to satisfy a relationship of $1.0 \text{ mm} \leq \text{DA} \leq 1.8 \text{ mm}$ so that excellent performance can be particularly exhibited.

[0049] As described above, in Embodiment 1, in the case of using HFO-1123 or the refrigerant mixture containing HFO-1123 as refrigerant circulating in the refrigerant circuit, the heat transfer performance of the heat exchanger 1 can be improved. Moreover, the refrigeration cycle apparatus 10 can be provided, with which a reduction in the reliability can be suppressed even when sludge is caused due to refrigerant decomposition.

Embodiment 2

[0050] Fig. 6 is a cross-sectional view of an example of a flat tube of a refrigeration cycle apparatus of Embodiment 2 of the present invention.

[0051] Fig. 7 is an enlarged cross-sectional view of the example of the flat tube of the refrigeration cycle apparatus of Embodiment 2 of the present invention.

[0052] As illustrated in Figs. 6 and 7, a flat tube 2 of Embodiment 2 is configured so that one or more protrusions 22 are formed in a passage direction on each inner wall surface of each passage 21. Note that, in the example of Fig. 7, the case where two-phase gas-liquid refrigerant flows is illustrated, "25" indicates a two-phase flow liquid film, and "26" indicates a gas-liquid interface.

[0053] With the protrusions 22 formed on the inner wall surface of each passage 21, a contact area between the inner wall surface of the passage 21 and refrigerant can be increased. Moreover, disturbance of the flow of refrigerant in the passage 21 can be promoted. Further, as illustrated in Fig. 7, liquid refrigerant is concentrated on a bottom portion 24 of each protrusion 22 due to surface

tension of refrigerant. Consequently, drainability in a tube axial direction can be improved, and the thickness of the two-phase flow liquid film 25 around the bottom portion 24 of the protrusion 22 can be reduced. Thus, heat can be more efficiently exchanged than that of the case where the protrusions 22 are provided.

[0054] Note that, in the example illustrated in Figs. 6 and 7, the protrusion 22 has a triangular cross-sectional shape, but the present invention is not limited to such a shape. For example, even when the protrusion 22 has any cross-sectional shapes such as a semicircular shape, an oval shape, and a rectangular shape, an advantageous effect similar to the above-described purpose can be obtained.

[0055] As illustrated in Fig. 8, when the plurality of protrusions 22 are formed on the inner wall surface of the passage 21 of the flat tube 2, sludge 31 caused in decomposition of HFO-1123 tends to be accumulated between adjacent ones of the protrusions 22, for example. In particular, the sludge 31 tends to be accumulated on a lower one of the inner wall surfaces 23 of the passage 21 of the flat tube 2 in the direction of gravitational force. Such sludge 31 accumulated in the passage 21 leads to an increase in a pressure loss in the tube or clogging. Thus, a pressure in a refrigerant circuit increases more than necessary, and for this reason, reliability may be lowered.

[0056] The size of sludge caused due to chemical reaction of a decomposition product of HFO-1123 is smaller than 0.15 mm. For this reason, a distance D between adjacent ones of the protrusions 22 are set to satisfy $0.15 \text{ mm} \leq D$ so that refrigerant can flow without accumulation of the sludge 31 in the passage 21. Moreover, the pressure loss in the tube can be reduced by increasing the distance D between adjacent ones of the protrusions 22. However, when the distance D between adjacent ones of the protrusions 22 becomes too large, the incremental modulus of a heat transfer area of the inner wall surface of the passage 21 decreases. Further, disturbance of the flow of refrigerant in the passage 21 cannot be sufficiently promoted. For these reasons, the distance D between adjacent ones of the protrusions 22 is preferably equal to or shorter than 0.50 mm.

[0057] Thus, in the case where the plurality of protrusions 22 are formed on a lower one of the inner wall surfaces 23 of the passage 21 in the direction of gravitational force, the distance D between adjacent ones of the protrusions 22 is set to satisfy a relationship of $0.15 \text{ mm} \leq D \leq 0.50 \text{ mm}$. With this configuration, even when the plurality of protrusions 22 are formed on the inner wall surface of the passage 21, refrigerant can flow without accumulation of the sludge 31.

[0058] Fig. 9 is a cross-sectional view of an example of the flat tube of the refrigeration cycle apparatus of Embodiment 2 of the present invention.

[0059] Fig. 10 is an enlarged cross-sectional view of the example of the flat tube of the refrigeration cycle apparatus of Embodiment 2 of the present invention.

[0060] In Figs. 9 and 10, a single protrusion 22 is provided on a lower one of inner wall surfaces 23 of a passage 21 in the direction of gravitational force, and a single protrusion 22 is provided on an upper one of the inner wall surfaces of the passage 21 in the direction of gravitational force. Note that the number of protrusions 22 is not limited to such an example, and a plurality of protrusions 22 may be provided on a single inner wall surface. Note that, in the case of providing the plurality of protrusions 22, a distance D between these protrusions 22 is set to satisfy the above-described relationship.

[0061] In the case of forming one or more protrusions 22 on the inner wall surface of the passage 21, sludge 31 caused in decomposition of HFO-1123 tends to be accumulated between the protrusion 22 and a side wall of the passage 21. In particular, the sludge 31 tends to be accumulated on a lower one of the inner wall surfaces 23 of the passage 21 of a flat tube 2 in the direction of gravitational force. For this reason, a distance C from a bottom portion 24 of the protrusion 22 to the side wall of the passage 21 is set to satisfy $0.15 \text{ mm} \leq C$ so that refrigerant can flow without accumulation of the sludge 31 in the passage 21. Moreover, when the distance C becomes too large, the incremental modulus of a heat transfer area of the inner wall surface of the passage 21 decreases. Further, disturbance of the flow of refrigerant in the passage 21 cannot be sufficiently promoted. For these reasons, the distance C is preferably equal to or shorter than 0.50 mm.

[0062] Thus, in the case where one or more protrusions 22 are formed on a lower one of the inner wall surfaces 23 of the passage 21 in the direction of gravitational force, the distance C from the bottom portion 24 of the protrusion 22 to the side wall of the passage 21 is set to satisfy a relationship of $0.15 \text{ mm} \leq D \leq 0.50 \text{ mm}$. Thus, even in the case where one or more protrusions 22 are formed on the inner wall surface of the passage 21, refrigerant can flow without accumulation of the sludge 31.

Embodiment 3

[0063] Fig. 11 is a characteristic graph of a relationship between pressure resistance performance and a ratio between a wall thickness TO and a short-axis diameter DA of a flat tube in a refrigeration cycle apparatus of Embodiment 3 of the present invention.

[0064] In the case of using HFO-1123 or a refrigerant mixture containing HFO-1123 as refrigerant flowing through a refrigerant circuit, HFO-1123 has a lower standard boiling point than that of conventionally-used HFC-410A. Thus, a pressure is higher than that at a saturation temperature equivalent to the case of HFC-410A. For this reason, sufficient pressure resistance performance of a flat tube 2 needs to be ensured.

[0065] In an example of Fig. 11, the wall thickness TO and the short-axis diameter DA of the flat tube 2 are used as parameters under calculation conditions in simulation of a coefficient of performance as shown in Fig. 5 of Em-

bodiment 1 described above. In Fig. 11, the horizontal axis represents the ratio TO/DA between the wall thickness TO and the short-axis diameter DA of the flat tube 2, and the vertical axis represents a ratio Pr/Pn between the pressure resistance performance Pr of the flat tube 2 and required pressure resistance Pn of the flat tube 2.

[0066] As shown in Fig. 11, TO/DA and Pr/Pn are in a proportional relationship. That is, regardless of an absolute value of the short-axis diameter DA of the flat tube 2, a decrease in TO/DA results in a decrease in the value of Pr/Pn. As a ratio of the wall thickness TO to the short-axis diameter DA decreases with a decrease in TO/DA, a cross-sectional area of a passage 21 and a heat transfer area of the passage 21 increase, and heat exchange performance is also enhanced. Moreover, as the ratio of the wall thickness TO to the short-axis diameter DA decreases with a decrease in TO/DA, the pressure resistance performance Pr is lowered, and Pr/Pn decreases.

[0067] The pressure resistance performance Pr needs to exceed the required pressure resistance Pn. That is, when Pr/Pn is equal to or smaller than one, sufficient pressure resistance performance cannot be ensured. For this reason, $Pr/Pn > 1$ needs to be satisfied. In the example shown in Fig. 11, when TO/DA falls below 0.10, Pr/Pn is equal to or smaller than one. Thus, a relationship of $0.10 \leq TO/DA$ needs to be satisfied. On the other hand, when TO/DA extremely increases, sufficient pressure resistance performance is ensured. However, the cross-sectional area and the heat transfer area of the passage 21 decreases, and the heat exchange performance is also lowered. In particular, in the case where TO/DA exceeds 0.20, almost half of the flat tube 2 in a short-axis direction is occupied with the wall thickness, and for this reason, sufficient performance cannot be exhibited. As described above, TO/DA is preferably $0.10 \leq TO/DA \leq 0.20$.

[0068] In the case where an absolute value of the wall thickness TO is small, a hole may be bored in the tube due to, for example, corrosion of the flat tube 2. For this reason, in addition to the above-described range of TO/DA, a relationship of $TO \geq 0.10 \text{ mm}$ is preferably satisfied.

[0069] As described above, a relationship of $0.10 \leq TO/DA \leq 0.20$ and a relationship of $TO \geq 0.10 \text{ mm}$ are satisfied. Consequently, in the case of using HFO-1123 or a refrigerant mixture containing HFO-1123, sufficient pressure resistance performance can be ensured, and the heat exchange performance can be improved.

[0070] Note that, in Embodiments 1 to 3 described above, the case where the heat exchanger 1 is used as the evaporator in an outdoor unit of the air-conditioning apparatus has been described as an example. However, a similar advantageous effect can be also obtained in the case where the heat exchanger 1 is used as another heat exchanger 1 used as a condenser in an indoor unit of an air-conditioning apparatus.

[0071] In addition, the present invention is not limited to the air-conditioning apparatus, and a similar advanta-

geous effect can be also obtained in the case where the present invention is applied to the refrigeration cycle apparatus 10 in which the above-described heat exchanger 1 is used as at least one of the first heat exchanger 12 and the second heat exchanger 14 in the refrigerant circuit formed by connecting at least the compressor 11, the first heat exchanger 12, the expansion unit 13, and the second heat exchanger 14 in sequence to each other via pipes. Reference Signs List

[0072] 1 heat exchanger, 2 flat tube, 3 fin, 4a header, 4b header, 10 refrigeration cycle apparatus, 11 compressor, 12 first heat exchanger, 13 expansion unit, 14 second heat exchanger, 15a fan, 15b fan, 20 partition wall, 21 passage, 22 protrusion, 23 inner wall surface, 24 bottom portion, 25 two-phase flow liquid film, 31 sludge

Claims

1. A refrigeration cycle apparatus comprising a refrigerant circuit including a compressor, a first heat exchanger, an expansion unit, and a second heat exchanger connected in sequence to each other via pipes to circulate refrigerant through the refrigerant circuit, the refrigerant comprising one of HFO-1123 and a refrigerant mixture containing HFO-1123, at least one of the first heat exchanger and the second heat exchanger including a plurality of fins and a plurality of flat tubes, the plurality of flat tubes each including a plurality of passages through which the refrigerant flows, where the plurality of flat tubes each have a thickness DA, a relationship expressed by $0.9 \text{ mm} \leq DA \leq 3.0 \text{ mm}$ being satisfied.
2. The refrigeration cycle apparatus of claim 1, wherein the plurality of flat tubes are configured so that each of the plurality of flat tubes satisfies the relationship of $0.9 \text{ mm} \leq DA \leq 3.0 \text{ mm}$, and a ratio between a row pitch corresponding to a distance connecting centers of adjacent ones of the plurality of flat tubes and the thickness DA is a preset value.
3. The refrigeration cycle apparatus of claim 1 or 2, wherein the plurality of flat tubes are configured so that each of the plurality of flat tubes satisfies the relationship of $0.9 \text{ mm} \leq DA \leq 3.0 \text{ mm}$, and a ratio between a length of each of the plurality of passages in one direction and a length of each of the plurality of passages in an other direction and a ratio between a wall thickness of each of the plurality of flat tubes and the thickness DA of each of the plurality of flat tubes are preset values.
4. The refrigeration cycle apparatus of any one of

claims 1 to 3, wherein each of the plurality of flat tubes satisfies a relationship of $1.0 \text{ mm} \leq DA \leq 1.8 \text{ mm}$.

5. The refrigeration cycle apparatus of any one of claims 1 to 4, wherein each of the plurality of flat tubes has one or more protrusions formed in a passage direction on an inner wall surface of each of the plurality of passages.
6. The refrigeration cycle apparatus of any one of claims 1 to 4, wherein the plurality of flat tubes are configured so that each of the plurality of flat tubes has at least one protrusion formed in a passage direction on a lower one of inner wall surfaces of each of the plurality of passages in a direction of gravitational force, and, where C represents a distance from a bottom portion of the at least one protrusion to a side wall of each of the plurality of passages, a relationship of $0.15 \text{ mm} \leq C \leq 0.50 \text{ mm}$ is satisfied.
7. The refrigeration cycle apparatus of any one of claims 1 to 4, wherein each of the plurality of flat tubes has a plurality of protrusions formed in a passage direction on an inner wall surface of each of the plurality of passages, and each of the plurality of passages satisfies a relationship of $0.15 \text{ mm} \leq D \leq 0.50 \text{ mm}$, where D represents a distance between adjacent ones of the plurality of protrusions.
8. The refrigeration cycle apparatus of any one of claims 1 to 7, wherein each of the plurality of flat tubes satisfies a relationship of $0.10 \leq TO/DA \leq 0.20$ and a relationship of $TO \geq 0.10 \text{ mm}$, where TO represents a wall thickness between an outer wall of each of the plurality of flat tubes and an inner wall of each of the plurality of passages.

FIG. 1

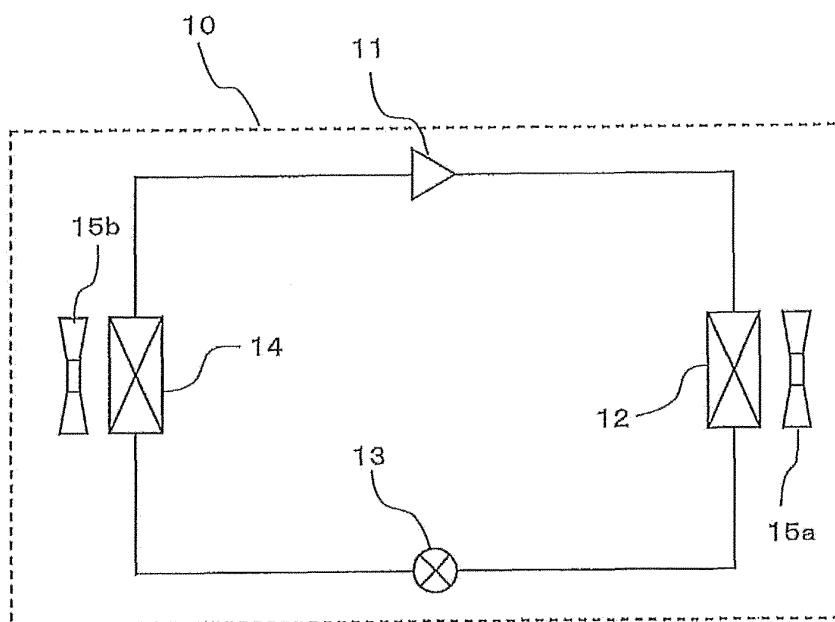


FIG. 2

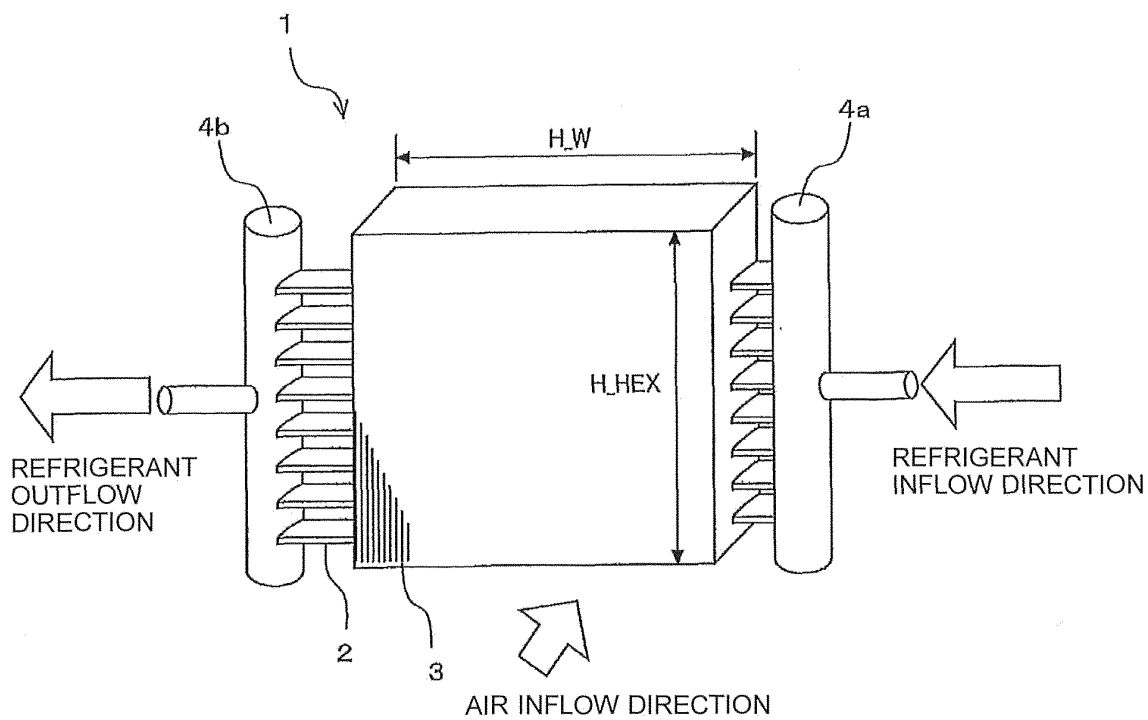


FIG. 3

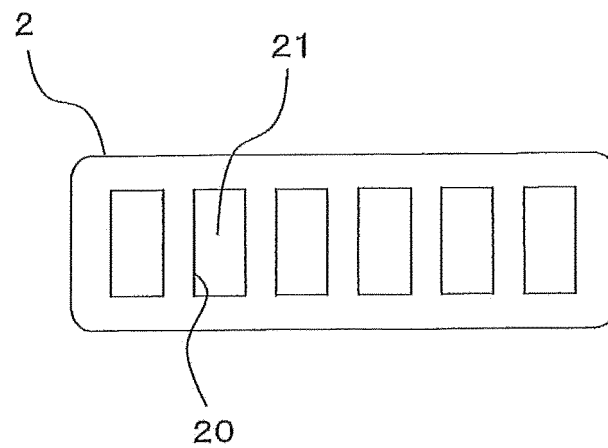


FIG. 4

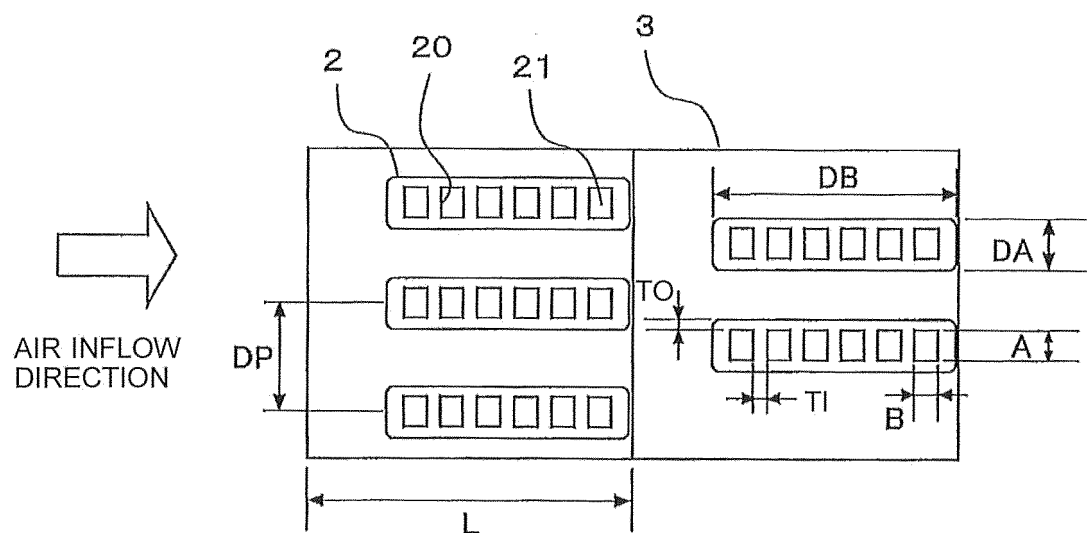


FIG. 5

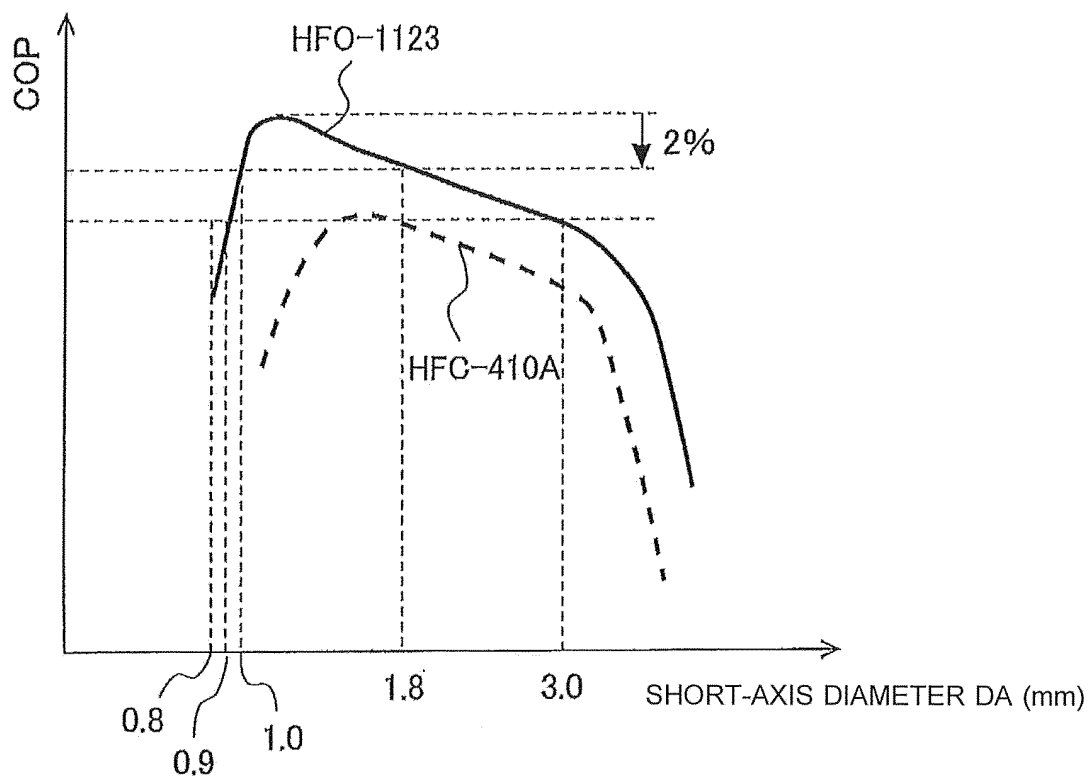


FIG. 6

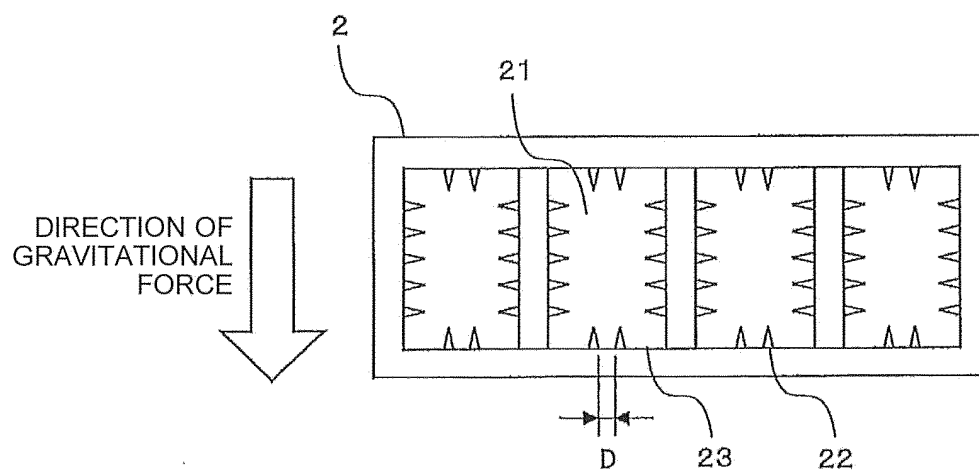


FIG. 7

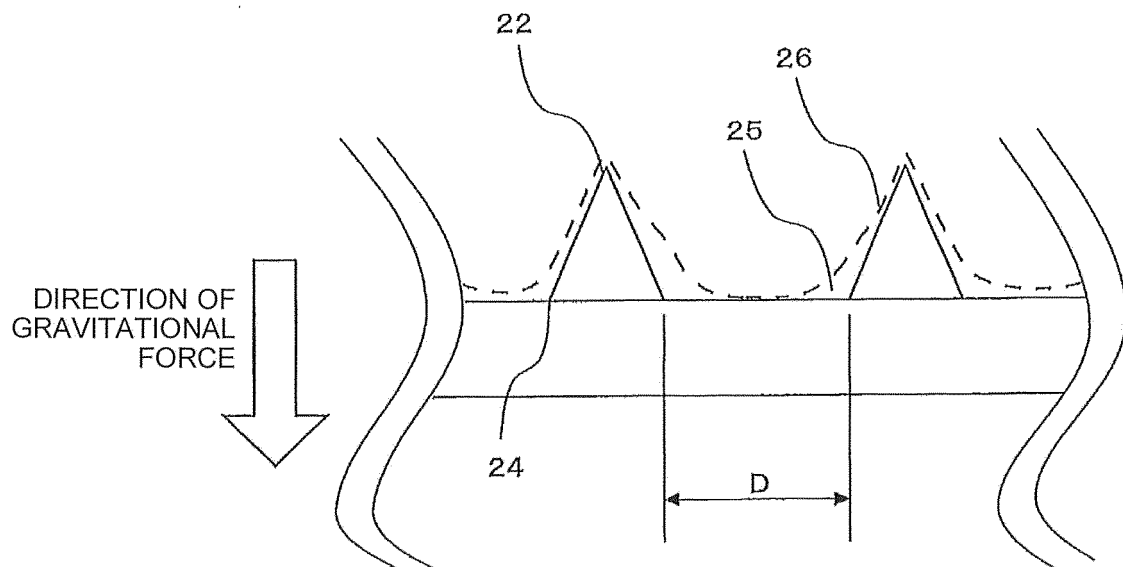


FIG. 8

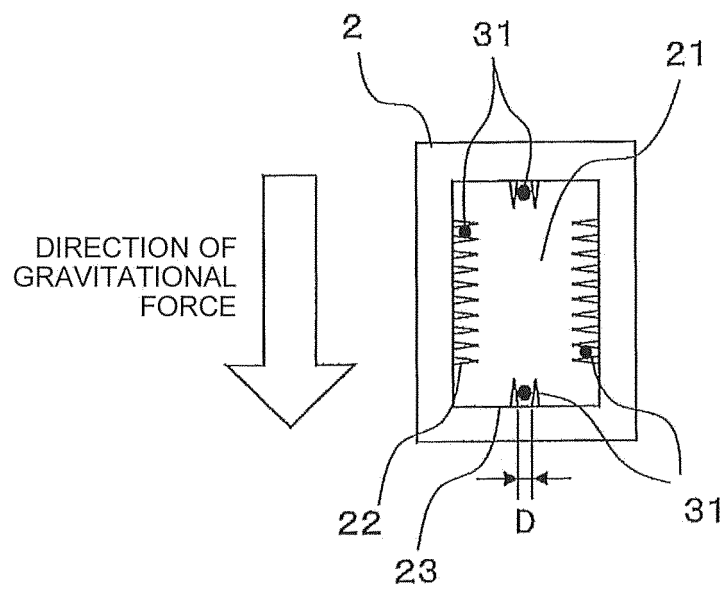


FIG. 9

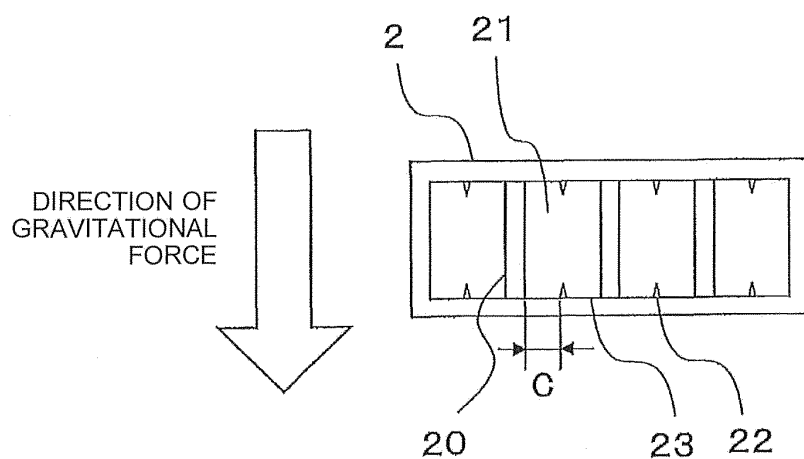


FIG. 10

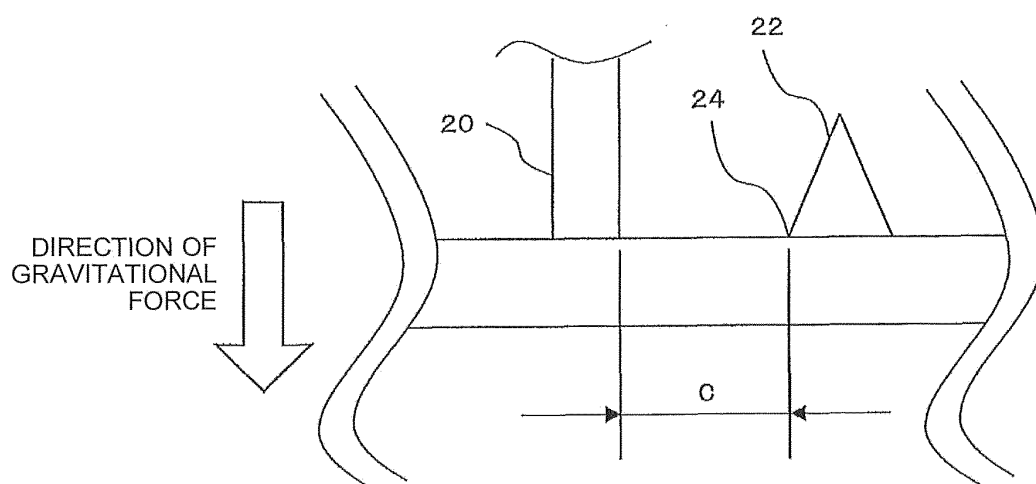
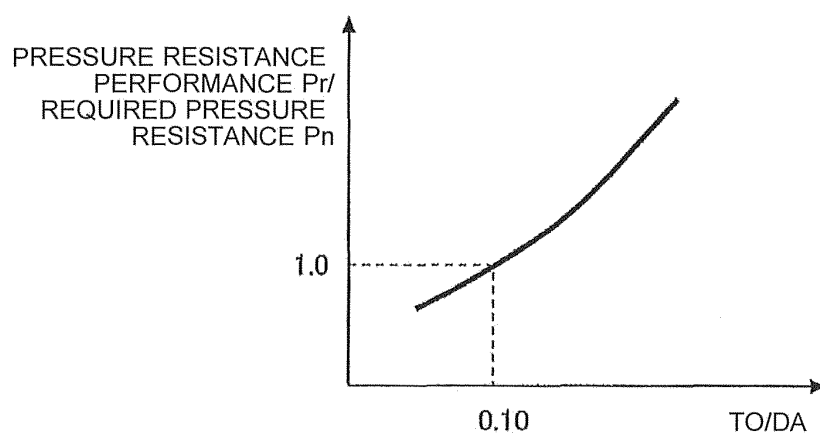


FIG. 11



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2014/084466

A. CLASSIFICATION OF SUBJECT MATTER

F28F1/02(2006.01)i, F25B1/00(2006.01)i, F28D1/053(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)

F28F1/02, F25B1/00, F28D1/053

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 1996-2015
 Kokai Jitsuyo Shinan Koho 1971-2015 Toroku Jitsuyo Shinan Koho 1994-2015

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	WO 2014/203353 A1 (Mitsubishi Electric Corp.), 24 December 2014 (24.12.2014), paragraphs [0013], [0022], [0055]; fig. 1, 2, 10 (Family: none)	1-8
Y	JP 2001-165532 A (Denso Corp.), 22 June 2001 (22.06.2001), paragraphs [0016], [0026] to [0028]; fig. 1 to 13 & US 2001/0004935 A1 & US 2005/0155747 A1 & DE 10060104 A1	1-8
Y	JP 2000-356488 A (Showa Aluminum Corp.), 26 December 2000 (26.12.2000), paragraphs [0017] to [0018]; fig. 1, 3 (Family: none)	3-8

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 ☐ See patent family annex.

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Date of the actual completion of the international search
11 March 2015 (11.03.15)Date of mailing of the international search report
24 March 2015 (24.03.15)Name and mailing address of the ISA/
Japan Patent Office
3-4-3, Kasumigasaka, Chiyoda-ku,
Tokyo 100-8915, Japan

Authorized officer

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INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2014/084466

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
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Y	CN 102269536 A (Danfoss-Sanhua (Hangzhou) Microchannel Heat Exchanger Co., Ltd.), 07 December 2011 (07.12.2011), paragraphs [0027] to [0028]; fig. 3 (Family: none)	3-8
Y	JP 2009-63228 A (Showa Denko Kabushiki Kaisha), 26 March 2009 (26.03.2009), paragraphs [0011], [0018], [0091]; fig. 1 to 10, 14 & US 2009/0065183 A1 & DE 102008045710 A1	4-8

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REFERENCES CITED IN THE DESCRIPTION

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- JP 2014098166 A [0003]