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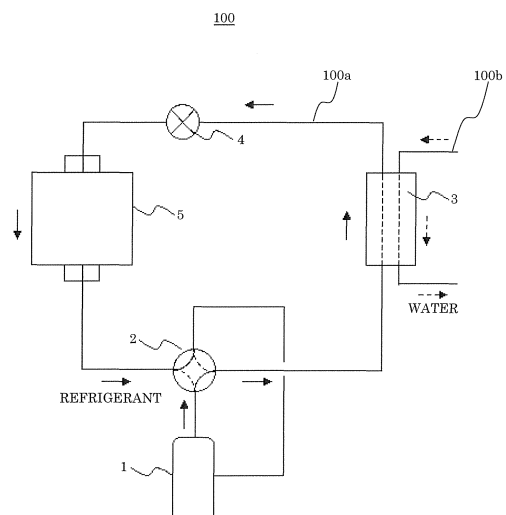
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(54) **HEAT PUMP APPARATUS**

(57) A heat pump apparatus includes a refrigerant circuit to which a compressor (1), a condenser (3 or 5), an expander (4), an evaporator (5 or 3) are connected in a loop by a refrigerant tube. One of the condenser (3 or 5) and the evaporator (5 or 3) is a plate heat exchanger which performs heat exchange between a refrigerant and a liquid. The refrigerant which is filled in the refrigerant circuit contains R32 as a main component, which reduces refrigerant drift current inside the condenser (3 or 5) and pressure losses, and thereby increases energy-saving performances.

FIG. 1



EP 3 220 075 A1

Description

Citation List

TECHNICAL FIELD

Patent Literature

[0001] The present invention relates to a heat pump apparatus.

5 **[0008]** PTL 1: Japanese Unexamined Patent Application Publication No. 2001-248922

BACKGROUND ART

Non-patent Literature

[0002] Conventional heat pump apparatuses include a heat pump apparatus including a finned tube heat exchanger and configured to be smaller by using R32 as a refrigerant so as to reduce the volume ratio between an indoor heat exchanger and an outdoor heat exchanger (for example, see Patent Literature 1).

10 **[0009]**

[0003] FIG. 7 is a schematic configuration diagram of a conventional heat pump apparatus disclosed in Patent Literature 1. As illustrated in FIG. 7, the heat pump apparatus includes a refrigerant circuit to which compressor 101, outdoor heat exchanger 102, expansion valve 103, indoor heat exchanger 104, and four-way valve 105 are connected in a loop.

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NPL 1: "Jouhatsu seinou wo koujoushita naimen mizo tsuki kuda no kaihatsu" ("Development of grooved tube which provides enhanced evaporation performances"), Mamoru Hofuku and two others, Hitachi Cable, Ltd., No. 26 (2007-1), p. 59.

[0004] This heat pump apparatus uses R32 as a refrigerant. In addition, this heat pump apparatus is intended to be applied to an air conditioner. For this reason, each of outdoor heat exchanger 102 and indoor heat exchanger 104 is a finned tube heat exchanger which exchanges a refrigerant and air. Ratio $m (= V_{out} / V_{in})$ between internal volume V_{out} of outdoor heat exchanger 102 and internal volume V_{in} of indoor heat exchanger 104 is set to a value in a range of $0.7 \leq m \leq 1.5$. In this way, compared to the heat pump apparatus configured using a conventional refrigerant such as R410A in particular, the heat pump apparatus is configured to save energy and have a smaller outdoor heat exchanger.

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NPL 2: "Daigae reibai you kouseinou naimen mizo tsuki kuda HF taipu no kaihatsu" ("Development of HF-type High-Performance Multi-Grooved Tubes for Alternative Refrigerants"), Yasutoshi Mori and three others, Furukawa denkou jihou (Furukawa Electric News Reports), No. 106, p. 8, July, Heisei 12 (2000), Furukawa Electric Co., Ltd.

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SUMMARY OF THE INVENTION

TECHNICAL PROBLEM

[0005] In addition, in order to reduce the internal volume of the finned tube heat exchanger, there is a need to reduce the tube diameters of heat transfer tubes in the heat exchanger. In this case, it is an object of securing heat transfer performances while reducing pressure losses of the refrigerant flowing inside the heat transfer tubes.

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[0010] However, the above-described conventional technique uses R32 as the refrigerant assuming that the heat pump apparatus is applied to an air conditioner, and thus is not configured considering the case in which the heat pump apparatus is applied to a device which performs heat exchange between the refrigerant and a liquid.

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[0006] There are techniques which clarify the relationship between pressure losses and the flow rates of a refrigerant inside heat transfer tubes at the time when the refrigerant evaporates (see Non-patent Literature 1, and Non-patent Literature 2).

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[0011] Accordingly, when the conventional configuration is directly applied to the heat pump apparatus which performs heat exchange between the refrigerant and the liquid, there is a problem that energy-saving performances are not sufficiently increased.

[0007] In FIG. 8, (a) is a graph showing the relationship between pressure losses and the flow rates of a refrigerant at the time when the refrigerant evaporates, disclosed in Non-patent Literature 1. In FIG. 8, (b) is a graph showing the relationship between pressure losses and the flow rates of a refrigerant at the time when the refrigerant evaporates, disclosed in Non-patent Literature 2. These graphs show that pressure losses are approximately proportional to the squares of the flow rates.

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[0012] The present invention was made to solve the conventional problem, and has an object to increase the energy-saving performance of the heat pump apparatus which performs heat exchange between a refrigerant and a liquid.

SOLUTION TO PROBLEM

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[0013] In order to solve the above-described conventional problem, a heat pump apparatus according to the present invention includes: a refrigerant circuit in which a compressor, a condenser, an expander, and an evaporator are connected in a loop by a refrigerant pipe, wherein one of the condenser and the evaporator is a plate heat exchanger which performs heat exchange between a refrigerant and a liquid, and the refrigerant which is filled in the refrigerant circuit contains R32 as a main component.

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[0014] Under the conditions for enabling the heat pump apparatus using R32 to exhibit the same performances as the heat pump apparatus using conventional R410A, enthalpy difference h in a condensation process performed by the heat pump apparatus using R32 is larger than enthalpy difference h in a condensation process performed by the heat pump apparatus using R410A. Thus, when R32 is used, a refrigerant circulation amount required to obtain the predetermined performances is reduced. Accordingly, a flow velocity of the refrigerant in the heat exchanger decreases. This reduces a refrigerant drift current which occurs in a flow path, from a closer part to a more distant part with respect to a refrigerant inlet of a plate heat exchanger. As a result, the energy-saving performances of the heat pump apparatus increase.

ADVANTAGEOUS EFFECT OF INVENTION

[0015] According to the present invention, it is possible to provide the heat pump apparatus having increased energy-saving performances.

BRIEF DESCRIPTION OF DRAWINGS

[0016]

FIG. 1 is a schematic configuration diagram illustrating a heat pump apparatus according to Embodiment 1 of the present invention.

FIG. 2 is a Mollier (p - h) chart showing the states of R410A and R32.

In FIG. 3, regarding a plate heat exchanger of the heat pump apparatus according to Embodiment 1 of the present invention, (a) is a graph illustrating a flow velocity distribution at a refrigerant inlet header in the case where the refrigerant inlet and a refrigerant outlet are formed in different surfaces of the plate heat exchanger; and (b) is a graph illustrating a flow velocity distribution at a refrigerant inlet header in the case where the refrigerant inlet and a refrigerant outlet are formed in the same surface of the plate heat exchanger.

FIG. 4 is a graph illustrating the relationship between compression ratios of a compressor and volume ratios R of a heat exchanger in the case of the heat pump apparatus using R410A.

FIG. 5 is a graph illustrating the relationship between compression ratios of a compressor and volume ratios R of the heat exchanger in the case of the heat pump apparatus using R32.

FIG. 6 is a Mollier chart illustrating differences between a compression process of a low-pressure shell compressor and a compression process of a high-pressure shell compressor.

FIG. 7 is a schematic configuration diagram of a conventional heat pump apparatus.

In FIG. 8, each of (a) and (b) is a graph illustrating

the relationship between refrigerant flow rates and pressure losses at the time when the refrigerant evaporates.

DESCRIPTION OF EXEMPLARY EMBODIMENT

[0017] A first aspect of the present invention is a heat pump apparatus which includes: a refrigerant circuit in which a compressor, a condenser, an expander, and an evaporator are connected in a loop by a refrigerant pipe, wherein one of the condenser and the evaporator is a plate heat exchanger which performs heat exchange between a refrigerant and a liquid, and the refrigerant which is filled in the refrigerant circuit contains R32 as a main component.

[0018] Under the conditions for enabling the heat pump apparatus using R410A as the refrigerant and the heat pump apparatus using R32 as the refrigerant to exhibit the same performances, enthalpy difference h in the condensation process performed by the heat pump apparatus using R32 is larger than enthalpy difference h in the condensation process performed by the heat pump apparatus using R410A. Thus, a refrigerant circulation amount required to obtain predetermined performances is reduced by the heat pump apparatus using R32. Accordingly, a flow velocity of the refrigerant in the heat exchanger decreases. This reduces a refrigerant drift current which occurs in a flow path from a closer part to a more distant part with respect to a refrigerant inlet of the plate heat exchanger. As a result, heat exchange in the plate heat exchanger is performed efficiently as a whole, which increases the energy-saving performances of the heat pump apparatus.

[0019] In addition, since enthalpy difference h in the condensation process performed by the heat pump apparatus using R32 is larger, dryness x at the inlet of an evaporator decreases when the pressure of the refrigerant of each of the heat pump apparatuses is reduced in the same super cooling state using the same pressure. Thus, when the plate heat exchanger is used as the evaporator, the ratio of a liquid refrigerant increases in the refrigerant flowing inside the plate heat exchanger. Accordingly, the flow rate of the refrigerant flowing inside the plate heat exchanger decreases. This reduces a refrigerant drift current which occurs in a flow path, from a closer part to a more distant part with respect to the refrigerant inlet of the plate heat exchanger. As a result, heat exchange in the plate heat exchanger is performed efficiently as a whole, which increases the energy-saving performances of the heat pump apparatus.

[0020] According particularly to the first aspect of the present invention, in a second aspect of the present invention, the condenser is the plate heat exchanger, the evaporator is a finned tube heat exchanger which performs heat exchange between the refrigerant and gas, and ratio $R (= V_f / V_p)$ of an internal volume (V_f) of the evaporator with respect to an internal volume (V_p) of the condenser satisfies $0.5 \leq R \leq 4.0$.

[0021] In this way, it is possible to increase the energy-saving performances of the heat pump apparatus, and also to provide a smaller evaporator which is a finned tube heat exchanger. In addition, the reliability of the compressor increases. In addition, the evaporator configured to be smaller reduces the internal volume of the heat exchanger, which makes it possible to reduce a refrigerant amount required for the heat pump apparatus.

[0022] According particularly to the first or second aspect of the present invention, in a third aspect of the present invention, the compressor is a high-pressure shell compressor.

[0023] In a low-pressure shell compressor having a low pressure inside the shell, compression is started after a refrigerant is heated by a motor in the shell. On the other hand, in the high-pressure compressor, a refrigerant fed to the compressor is directly fed to a compression room inside the compressor. For this reason, the compression is started before the refrigerant is heated by the motor. Thus, even in the case of using R32 which is easy to increase an outlet temperature, it is possible to reduce an excessive increase in outlet temperature, which increases the reliability of the compressor.

[0024] Hereinafter, an embodiment of the present invention is described with reference to the drawings. It is to be noted that this embodiment does not limit the present invention.

EMBODIMENT 1

[0025] FIG. 1 is a schematic configuration diagram illustrating a heat pump apparatus according to this embodiment. FIG. 2 is a Mollier (p-h) chart showing the states of R410A and R32. FIG. 3 includes graphs each showing a flow velocity distribution at a refrigerant inlet header of a plate heat exchanger; (a) is a graph in the case where a refrigerant inlet and a refrigerant outlet are formed in different surfaces of the plate heat exchanger, and (b) is a graph in the case where a refrigerant inlet and a refrigerant outlet are formed in the same surface of the plate heat exchanger.

[0026] In addition, FIG. 4 is a graph illustrating the relationship between compression ratios of a compressor and volume ratios R of a heat exchanger in the case of using R410A for the heat pump apparatus. FIG. 5 is a graph illustrating the relationship between compression ratios of a compressor and volume ratios R of the heat exchanger in the case of using R32 for the heat pump apparatus. FIG. 6 is a Mollier (p-h) chart illustrating differences between a compression process performed by a low-pressure shell compressor and a compression process performed by a high-pressure shell compressor.

[0027] As illustrated in FIG. 1, heat pump apparatus 100 in this embodiment includes refrigerant circuit 100a to which the following parts are connected in order using a refrigerant pipe: compressor 1 which compresses a refrigerant; four-way valve 2 which switches the flowing direction of the refrigerant; first heat exchanger 3; expan-

sion means 4 which reduces the pressure of the high-pressure refrigerant to expand the refrigerant; and second heat exchanger 5.

[0028] Compressor 1 includes a shell which is an outer shell. Compressor 1 includes a compression mechanism and a motor inside the shell. The compression mechanism includes a compression room. A rotation of the motor reduces the volume of the compression room, which compresses the refrigerant. Compressor 1 is a high-pressure shell compressor having a high pressure inside the shell.

[0029] First heat exchanger 3 is a plate heat exchanger in which the refrigerant and a liquid (for example, water) are thermally exchanged. Second heat exchanger is a finned tube heat exchanger in which the refrigerant and gas (for example, air) are thermally exchanged.

[0030] In addition, heat pump apparatus 100 includes liquid circuit 100b to which first heat exchanger 3, a circulation pump (not illustrated), a heating terminal (not illustrated) such as a floor heating panel etc. are connected using a liquid pipe.

[0031] Expansion means 4 can be an expansion valve capable of adjusting the cross-sectional area of a flow path in which a refrigerant flows, or a capillary tube whose cross-sectional area is constant.

[0032] Heat pump apparatus 100 is capable of causing four-way valve 2 to switch the flowing direction of the refrigerant between when first heat exchanger 3 and second heat exchanger 5 are used as a condenser and an evaporator, respectively, and when first heat exchanger 3 and second heat exchanger 5 are used as an evaporator and a condenser, respectively. In the condenser, the high-pressure refrigerant compressed by compressor 1 emits heat. In the evaporator, the refrigerant having two phases of gas and a liquid whose pressure is reduced and expanded by expansion means 4 evaporates.

[0033] A refrigerant which is used as the refrigerant to be filled in the refrigerant circuit contains R32 (difluoromethane) as a main component. It is preferable that the refrigerant contain R32 at a percentage of 70% or more. In addition, it is desirable that the refrigerant substantially consist of R32. It is to be noted that few impurities may be contained even when the refrigerant substantially consists only of R32.

[0034] The finned tube heat exchanger performs heat exchange between the refrigerant flowing inside the heat exchanger and the gas (air) flowing outside the heat exchanger. The finned tube heat exchanger includes a plurality of heat transfer tubes and a plurality of plate-shaped fins. The plurality of heat transfer tubes penetrate through the plurality of fins. The finned tube heat exchanger is configured to enable the refrigerant flowing inside the heat transfer tubes and air flowing between the fins to be thermally exchanged. A fan (not illustrated) for sending air between the fins is provided near the finned tube heat exchanger.

[0035] The plate heat exchanger performs heat exchange between the refrigerant flowing in the refrigerant

path inside the heat exchanger and the liquid (water) flowing in the liquid path inside the heat exchanger. The plate heat exchanger includes a stacked structure of plates with recesses and protrusions. A refrigerant path or a liquid path is formed between adjacent plates. Since the plurality of plates are stacked, the plate heat exchanger includes refrigerant paths and liquid paths which are alternately formed. The plate heat exchanger is configured to allow the refrigerant flowing in the refrigerant paths and a liquid (water) flowing in the liquid paths are thermally exchanged.

[0036] The plate heat exchanger includes a refrigerant inlet header at one of ends in the direction perpendicular to the layer stacking direction, and a refrigerant outlet header at the other end. The refrigerant inlet header has, at one end, a refrigerant inlet which is connected to a refrigerant pipe located outside. The refrigerant inlet header penetrates through the plurality of plates in the layer stacking direction from the refrigerant inlet, to be connected to the refrigerant flow path. In addition, the refrigerant outlet header has, at one end, a refrigerant outlet which is connected to a refrigerant pipe located outside. The refrigerant outlet header penetrates through the plurality of plates in the layer stacking direction from the refrigerant outlet, to be connected to the refrigerant flow path.

[0037] The refrigerant inlet and the refrigerant outlet may be provided in different surfaces of the plate heat exchanger (see the outline configuration in (a) of FIG. 3). Alternatively, the refrigerant inlet and the refrigerant outlet may be provided in the same surface (see the outline configuration in (b) of FIG. 3).

[0038] In addition, the plate heat exchanger includes a liquid inlet header and a liquid outlet header.

[0039] In heat pump apparatus 100 according to this embodiment, the refrigerant flow path of first heat exchanger 3 which is a plate heat exchanger is connected to refrigerant circuit 100a, and the liquid flow path of first exchanger 3 is connected to liquid circuit 100b. Heat pump apparatus 100 includes first heat exchanger 3 as a condenser, and is applicable to a hot water generating apparatus which heats a liquid.

[0040] More specifically, heat pump apparatus 100 is applicable to a hot-water heating apparatus which heats a room by circulating the heated liquid (hot water) in a heating terminal such as a floor heating panel or the like. It is to be noted that heat pump apparatus 100 according to this embodiment is capable of cooling a liquid using first heat exchanger 3 by switching the liquid flowing direction using four-way valve 2, and thus is also applicable to a cold and hot water generating apparatus.

[0041] Operation states in a freezing cycle in the case where heat pump apparatus 100 is operated under normal operation conditions are described using a Mollier (p-h) chart. As illustrated in FIG. 2, enthalpy difference h_2 in the case where R32 is used as a refrigerant is approximately 1.5 times larger than enthalpy difference h_1 in the case where R410A is used as a refrigerant. Thus,

R32 requires a smaller refrigerant circulation amount than R410A in order to enable the heat pump apparatus using R32 to exhibit the same heating performances as the heat pump apparatus using R410A. More specifically, the refrigerant circulation amount for R32 may be approximately 66% with respect to the refrigerant circulation amount for R410A.

[0042] In addition, as illustrated in FIG. 2, compared to the case of using R410A, the use of R32 as the refrigerant decreases dryness x at the inlet of the evaporator when the pressure of the refrigerant is reduced in the same super cooling state using the same condensation pressure. For example, under the operation conditions as illustrated in FIG. 2, dryness x of the refrigerant at the inlet of the evaporator of the heat pump apparatus using R32 is reduced to 0.19 while dryness x of the refrigerant at the inlet of the evaporator of the heat pump apparatus using R410A is 0.24.

[0043] Here, the flow velocity distributions of the refrigerant at the refrigerant inlet headers of the plate heat exchangers are described. The flow velocity distributions vary between the case in which a refrigerant inlet and a refrigerant outlet are provided in different surfaces of the plate heat exchanger ((a) in FIG. 3), and the case in which a refrigerant inlet and a refrigerant outlet are provided in the same surface of the plate heat exchanger ((b) in FIG. 3). More specifically, as illustrated in (a) of FIG. 3, when the refrigerant inlet and the refrigerant outlet are provided in the different surfaces, the flow is drifted so that the flow velocity at the side of the surface with the refrigerant inlet is slow and the flow velocity at the side of the surface (the depth side) with the refrigerant outlet is fast. As illustrated in (b) of FIG. 3, when the refrigerant inlet and the refrigerant outlet are provided in the same surface, the flow velocity at the side of the surface with the refrigerant inlet and the refrigerant outlet is fast and the flow velocity at the opposite side (the depth side) is slow.

[0044] Compared to the case in which R410A is used, the use of R32 decreases the refrigerant circulation amount, which reduces the refrigerant flow velocity. Thus, as illustrated in (a) and (b) of FIG. 3, irrespective of whether the refrigerant inlet and the refrigerant outlet are provided in the different surfaces or in the same surface, the variation in the refrigerant flow velocity decreases, which improves the refrigerant branch current.

[0045] In addition, dryness x of the refrigerant at the inlet of the evaporator decreases as described above, the ratio of the liquid refrigerant in the refrigerant flowing in the evaporator increases, which reduces the flow velocity of the refrigerant. In this way, especially when the plate heat exchanger is used as the evaporator, the drift current of the refrigerant inside the plate heat exchanger is improved, which improves the refrigerant branch current. Accordingly, since heat exchange is performed efficiently in each of the refrigerant flow paths in the plate heat exchanger, the heat exchange efficiency increases. As a result, the energy-saving performances of heat pump apparatus 100 increase.

[0046] Next, a description is given of appropriate volume ratio $R (= V_f / V_r)$ between internal volume V_f of the finned tube heat exchanger and internal volume V_p of the plate heat exchanger.

[0047] Internal volume V_f of the finned tube heat exchanger does not include the internal volume of the refrigerant pipe which is connected to the inlet and outlet of the heat exchanger so as to be connected to expansion means 4 and four-way valve 2, but includes the internal volume of the heat transfer pipe of the heat exchanger and the internal volume of the refrigerant pipe which connects the heat transfer tubes to each other. It is to be noted that the refrigerant pipe which connects the heat transfer tubes to each other includes branch tubes, header tubes, etc. for distributing the refrigerant to the plurality of transfer tubes, or for collecting the refrigerant from the plurality of heat transfer tubes.

[0048] Internal volume V_r of the plate heat exchanger does not include the internal volume of the refrigerant pipe which is connected to the inlet and outlet of the heat exchanger so as to be connected to expansion means 4 and four-way valve 2, but includes the internal volume of the refrigerant flow path, the internal volume of the refrigerant inlet header, and the internal volume of the refrigerant outlet header.

[0049] In general, in order to secure the reliability, compressor 1 needs to perform compression at the compression ratio in the range from 1.5 to 10. Thus, when heat pump apparatus 100 is applied to a hot water generating apparatus, there is a need to set the internal volume of the heat exchanger so that the compression ratio of compressor 1 falls between 1.5 to 10 in the actual use range of the hot water generating apparatus. It is to be noted that, the range employed as the actual use range in this embodiment is a range which yields a rated power ranging from 7 to 16 kW, an ambient air temperature ranging from -15 to 12 degrees Celsius, and a liquid (water) temperature ranging from 35 to 55 degrees Celsius.

[0050] Each of FIGs. 4 and 5 is a graph in which the horizontal axis denotes volume ratios R , the vertical axis denotes compression ratios, and the maximum values and the minimum values of compression ratios required in the actual use range are plotted.

[0051] When R410A is used, as illustrated in FIG. 4, the maximum values and minimum values of the compression ratios tend to decrease with increase in the volume ratio R . This is due to the reasons below. When the internal volume of the finned tube heat exchanger for use as an evaporator relatively increases, the heat exchange amount of the evaporator increases, which increases the pressure at the low-voltage side of heat pump apparatus 100. This increase results in decrease in the compression ratio required to obtain predetermined heating performances of the plate heat exchanger for use as a condenser.

[0052] In this way, according to increase or decrease in volume ratio $R (V_f / V_o)$, the maximum values and minimum values of the compression ratios also tend to increase or decrease. Thus, when R410A is used as the

refrigerant, the plate heat exchanger is used as the condenser, and the finned tube heat exchanger is used as the evaporator, the volume ratio R needs to satisfy $0.5 \leq R \leq 5$ as illustrated in FIG. 4 so that the compression ratio ranges from 1.5 to 10 in order to secure the reliability of the condenser.

[0053] As illustrated in FIG. 5, also when R32 is used, the maximum values and minimum values of the compression ratios tend to decrease with increase in the volume ratio R . Similarly to the case of R410A, also when R32 is used, compressor 1 needs to be operated so that the compression ratio ranges from 1.5 to 10, in terms of the maintenance in the reliability of heat pump apparatus 100, especially compressor 1. For this reason, as illustrated in FIG. 5, the volume ratio R needs to satisfy $0.5 \leq R \leq 4$ as illustrated in FIG. 5.

[0054] This indicates that, compared to the case where R410A is used for heat pump apparatus 100 with the same condenser (plate heat exchanger), it is possible to configure an evaporator (finned tube heat exchanger) to be smaller when R32 is used instead.

[0055] In addition, the case when R410A is used for heat pump apparatus 100 including the same condenser (plate heat exchanger) and the evaporator (finned tube heat exchanger) with heat transfer tubes having the same length L and the case when R32 is used for the same are compared. This comparison shows that the cross-sectional area S of the heat transfer tubes of the evaporator can be made smaller when R32 is used. It is to be noted that the cross-sectional area S of the heat transfer tubes is obtained by dividing internal volume V_f of the finned tube heat exchanger by length L of the heat transfer tubes of the finned tube heat exchanger (V_f / L).

[0056] More specifically, the cross-sectional area S of the heat transfer tubes of the finned tube heat exchanger in the case where R32 is used ranges from 0.8 to 1.0 times with respect to the cross-sectional area S in the case where R410A is used. Here, 0.8 can be obtained by dividing 4 which is the maximum volume ratio in the case where R32 is used by 5 which is the maximum volume ratio when R410A is used. In addition, 1.0 is obtained by dividing 0.5 which is the minimum volume ratio in the case where R32 is used by 0.5 which is the minimum volume ratio in the case where R410A is used.

[0057] The flow velocity of the refrigerant inside the heat transfer tubes of the finned tube heat exchanger in the case where R32 is used as the refrigerant ranges from 1.0 to 1.25 times with respect to the flow velocity of the refrigerant in the case where R410A is used as the refrigerant. Here, 1.0 is an inverse number of 1.0 which is a maximum value in the cross-sectional area S . In addition, 1.25 is an inverse number of 0.8 which is a minimum value in the cross-sectional area S .

[0058] Furthermore, the enthalpy difference h increases by approximately 1.5 times when R32 is used with respect to the enthalpy difference h obtained when R410A is used, and thus it is possible to secure the same performances even when the refrigerant flow velocity is

decreased to approximately 66%. As illustrated in (a) and (b) of FIG. 8 illustrating a conventional art, it is known that the pressure loss dP_c at the time when a refrigerant evaporates is proportional to a square of the flow velocity of the refrigerant. As described above, the refrigerant flow velocity of R32 ranges from 1.0 to 1.25 times with respect to the refrigerant flow velocity of R410A, and thus the pressure loss of R32 ranges from 0.44 to 0.68 times with respect to the pressure loss of R410A. Here, 0.44 is obtained when a value is squared. The value is obtained according to 0.66×1.0 wherein 0.66 indicates the ratio of the refrigerant flow velocity in the case where R32 is used and 1.0 indicates a minimum ratio of the flow velocity in the case where R32 is used. In addition, 0.68 is obtained when a value is squared. The value is obtained according to 0.66×1.25 wherein 0.66 indicates the ratio of the refrigerant flow velocity in the case where R32 is used and 1.25 indicates a minimum ratio of the flow velocity in the case where R32 is used.

[0059] As described above, the flow velocity of the refrigerant flowing in the condenser for enabling heat pump apparatus 100 using R32 to exhibit certain heating performances is smaller than the flow velocity of the refrigerant flowing in the condenser for enabling heat pump apparatus 100 using R410A to exhibit the same heating performances. Accordingly, it is possible to reduce the rotation rate of compressor 1, which decreases the motor power of compressor 1. Furthermore, compared to the case where R410A is used, the pressure losses in the case where R32 is used decrease, and thus the pressure of the refrigerant at the outlet of the evaporator, that is, the pressure of the refrigerant to be fed to compressor 1 increases. Accordingly, the specific volume of the refrigerant fed to compressor 1 decreases, resulting in increase in the amount of refrigerant to be fed to compressor 1. As a result, even when the motor power of compressor 1 is reduced, it is possible to secure circulation of a sufficient amount of refrigerant. In this way, it is possible to increase the condensation performances and evaporating performances.

[0060] In this way, compared to the case where R410A is used, in the case where R32 is used, the plate heat exchanger is used as the condenser, and the finned tube heat exchanger is used as the evaporator, it is possible to configure the evaporator to be smaller, reduce the required amount of refrigerant due to decrease in the internal volume of the evaporator, and increase the energy-saving performances of the heat pump apparatus.

[0061] It is preferable that a high-pressure shell compressor be used as compressor 1. In a low-pressure shell compressor having low pressure inside the shell, a refrigerant fed to the compressor once flows to the inside of the shell, and the refrigerant heated by the motor having a high temperature is fed to the compressor. Thus, as illustrated in FIG. 6, compression is started under the condition that the refrigerant has a temperature increased with respect to the temperature at the time when the refrigerant is fed to the compressor in the compres-

sion process. For this reason, there is a fear that the refrigerant after the compression is completed has a too high temperature (outlet temperature).

[0062] On the other hand, in a high-pressure shell compressor having a high pressure inside the shell, a refrigerant fed to the compressor is directly fed to a compression room. Thus, compression is started before the temperature of the refrigerant increases excessively. For this reason, it is possible to reduce excessive increase in the outlet temperature. In this way, even in the case of using R32 which is easier to increase an outlet temperature than R410A, it is possible to reduce an excessive increase in outlet temperature, which increases the reliability of the compressor.

[0063] Although this embodiment has been described assuming that the heat pump apparatus includes one condenser and one evaporator, it is to be noted that the heat pump apparatus may include a plurality of condensers and a plurality of evaporators. In this case, it is desirable that all of the heat exchangers used as the condensers be plate heat exchangers, and all of the heat exchangers used as the evaporators be finned tube heat exchangers. In addition, in this case, internal volume V_f of the finned tube heat exchanger is a sum of internal volumes of all of the finned tube heat exchangers. Internal volume V_r of the plate heat exchanger is a sum of internal volumes of all of the plate heat exchangers.

INDUSTRIAL APPLICABILITY

[0064] According to the present invention, it is possible to provide the heat pump apparatus which provides excellent energy saving properties. Thus, the present invention is applicable to hot water generating apparatuses, hot-water heating apparatuses, etc.

REFERENCE MARKS IN THE DRAWINGS

[0065]

- 1 compressor
- 2 four-way valve
- 3 first heat exchanger
- 4 expander
- 5 second heat exchanger

Claims

1. A heat pump apparatus, comprising a refrigerant circuit in which a compressor, a condenser, an expander, and an evaporator are connected in a loop by a refrigerant pipe, wherein one of the condenser and the evaporator is a plate heat exchanger which performs heat exchange between a refrigerant and a liquid, and the refrigerant which is filled in the refrigerant circuit contains R32 as a main component.

2. The heat pump apparatus according to claim 1,
wherein the condenser is the plate heat exchanger,
the evaporator is a finned tube heat exchanger which
performs heat exchange between the refrigerant and
gas, and 5
ratio R of an internal volume of the evaporator with
respect to an internal volume of the condenser sat-
isfies $0.5 \leq R \leq 4.0$.
3. The heat pump apparatus according to claim 1 or 10
claim 2,
wherein the compressor is a high-pressure shell
compressor.

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Amended claims under Art. 19.1 PCT

1. A heat pump apparatus, comprising
a refrigerant circuit in which a compressor, a con-
denser, an expander, and an evaporator are con- 20
nected in a loop by a refrigerant pipe,
wherein one of the condenser and the evaporator is
a plate heat exchanger which performs heat ex-
change between a refrigerant and a liquid,
the compressor is a high-pressure shell compressor, 25
and
the refrigerant which is filled in the refrigerant circuit
contains R32 as a main component.
2. The heat pump apparatus according to claim 1, 30
wherein the evaporator is a finned tube heat ex-
changer which performs heat exchange between the
refrigerant and gas,
the compressor is a high-pressure shell compressor,
and 35
ratio R of an internal volume of the evaporator with
respect to an internal volume of the condenser sat-
isfies $0.5 \leq R \leq 4.0$.

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FIG. 1

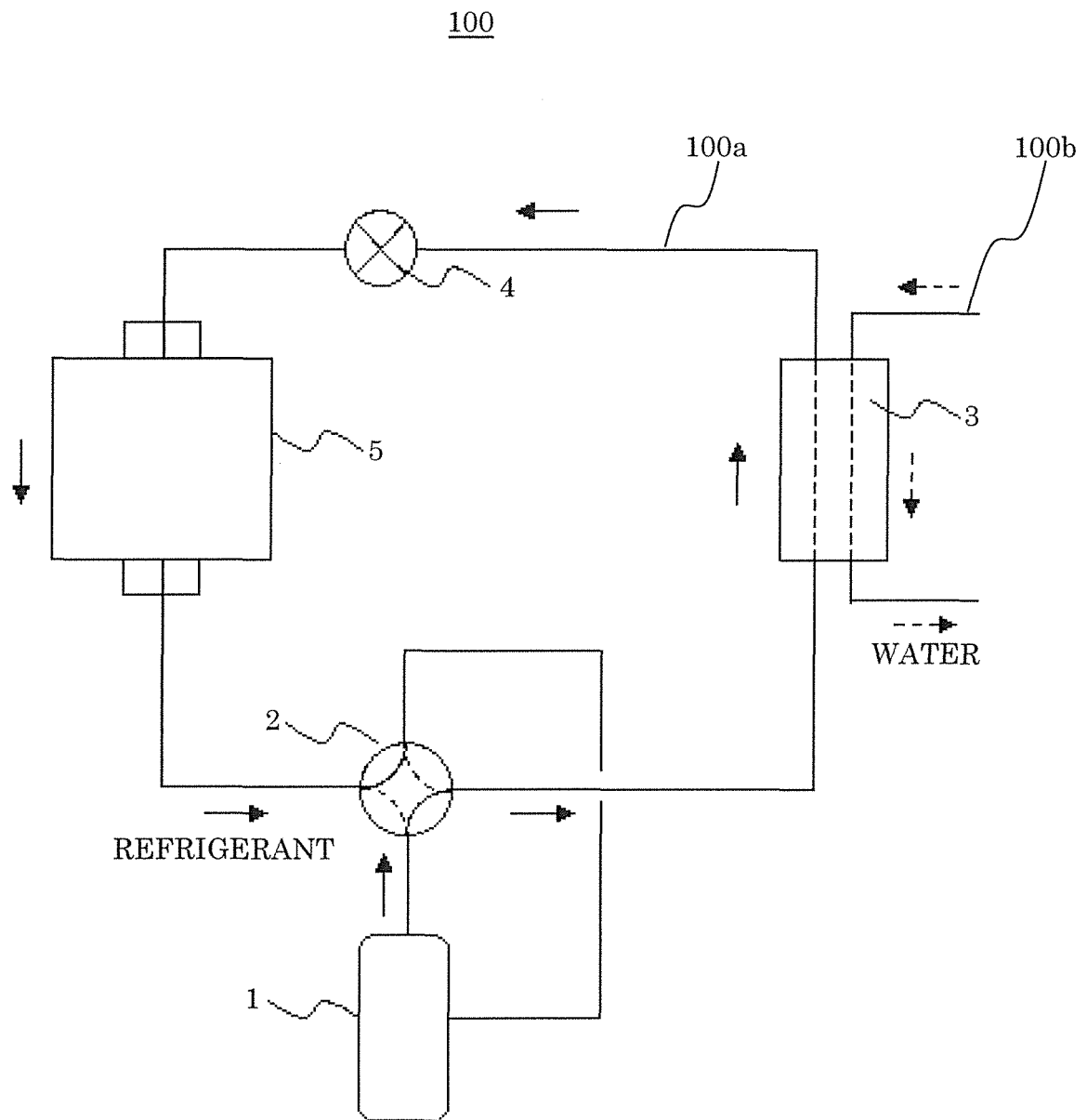


FIG. 2

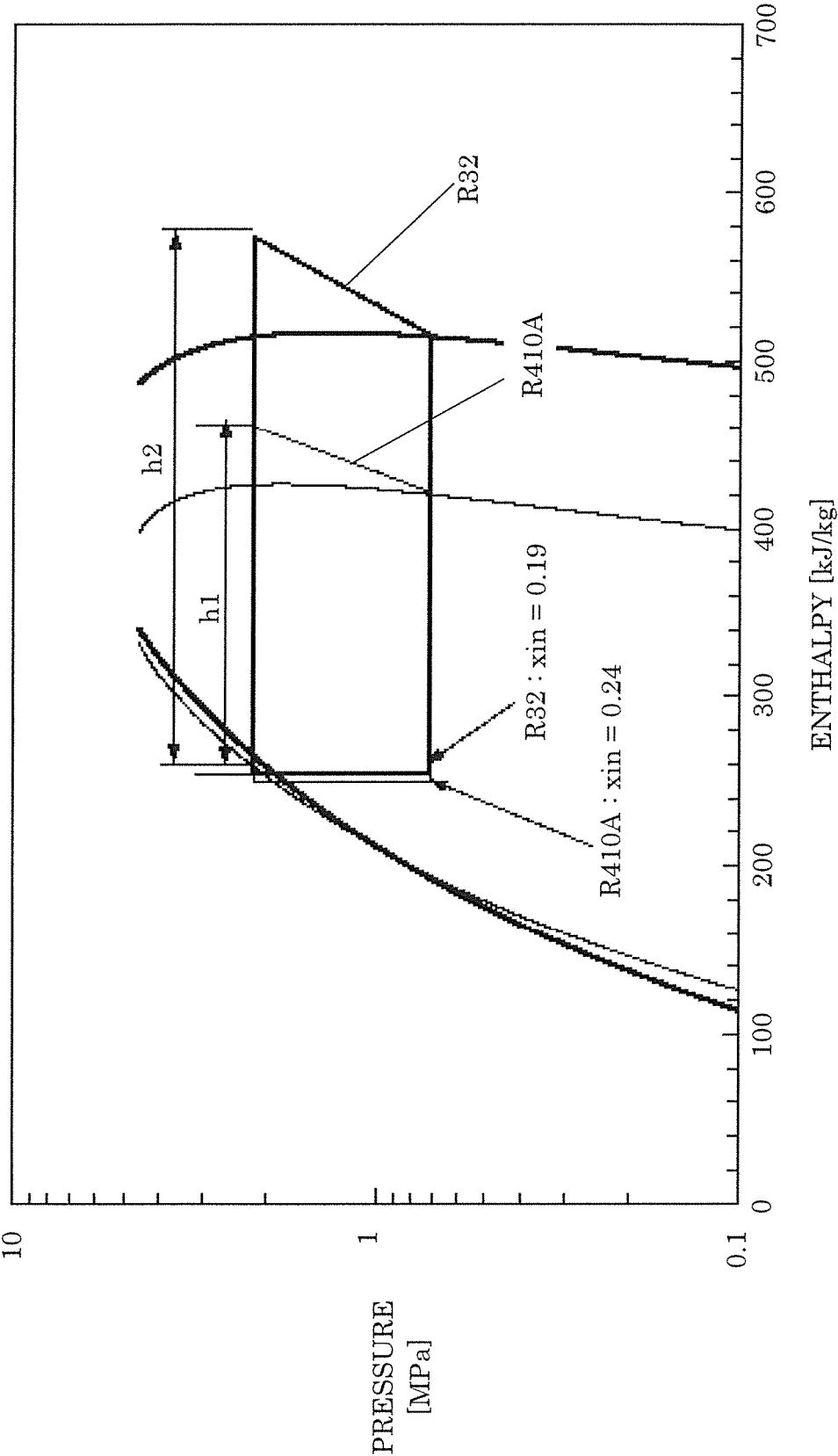


FIG. 3

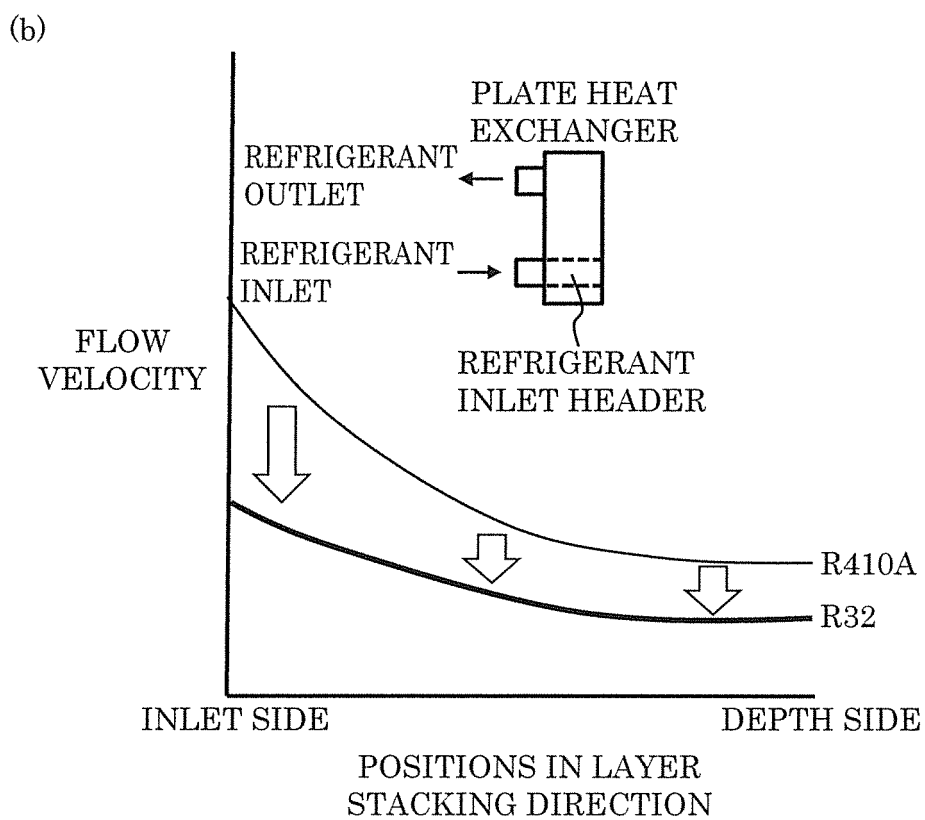
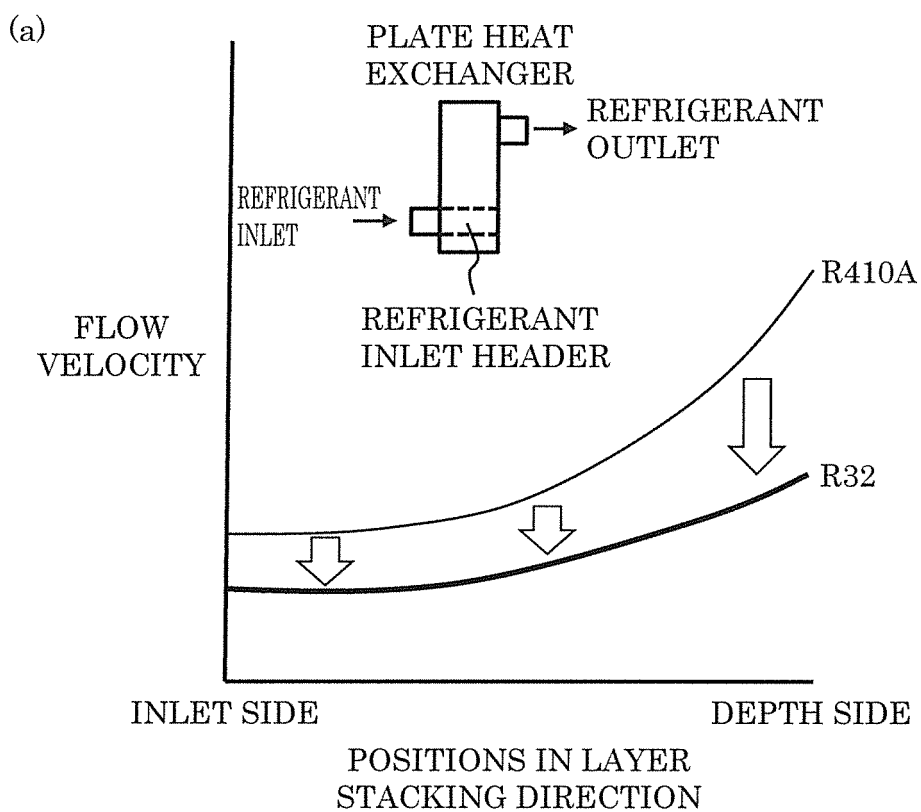


FIG. 4

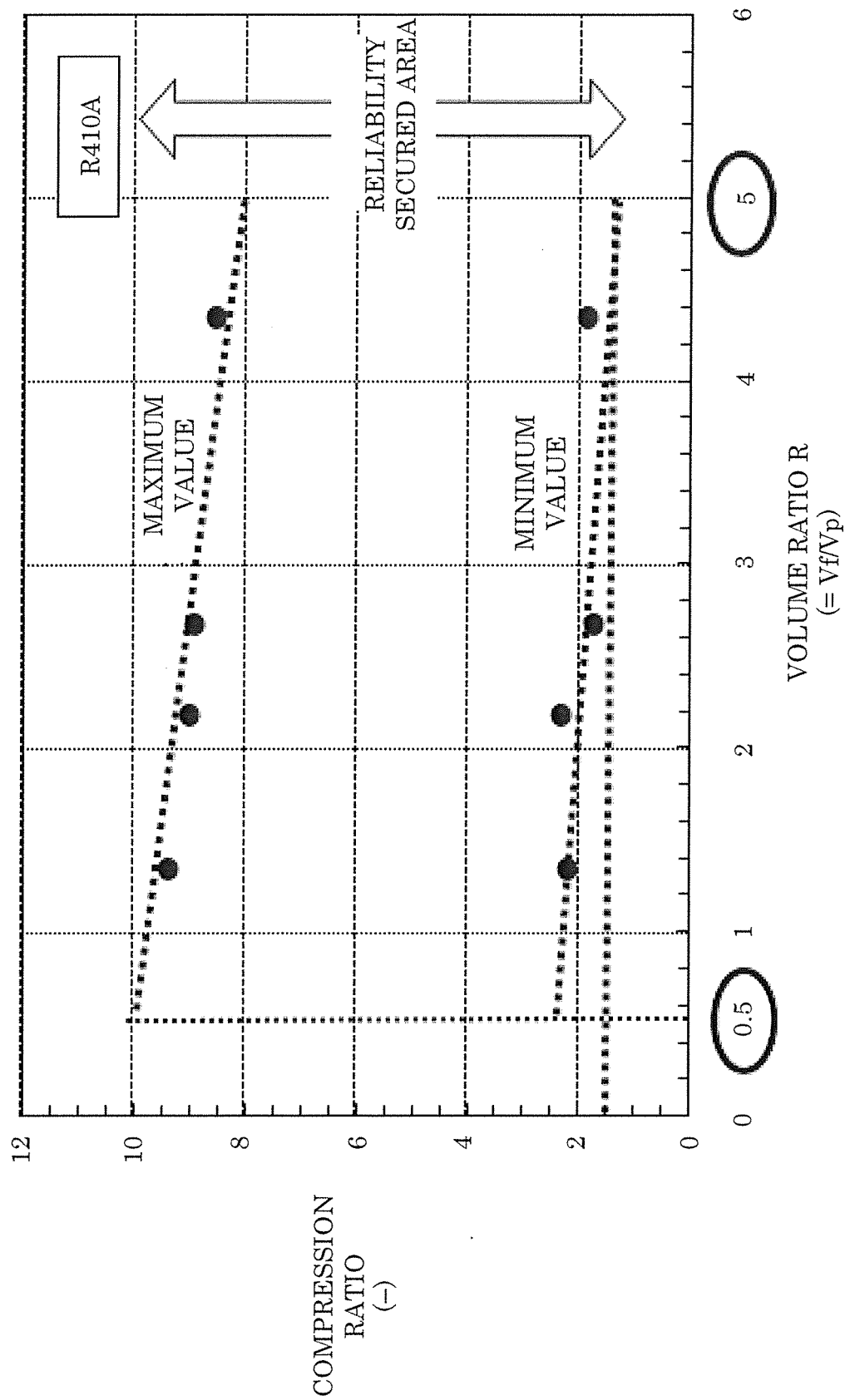


FIG. 5

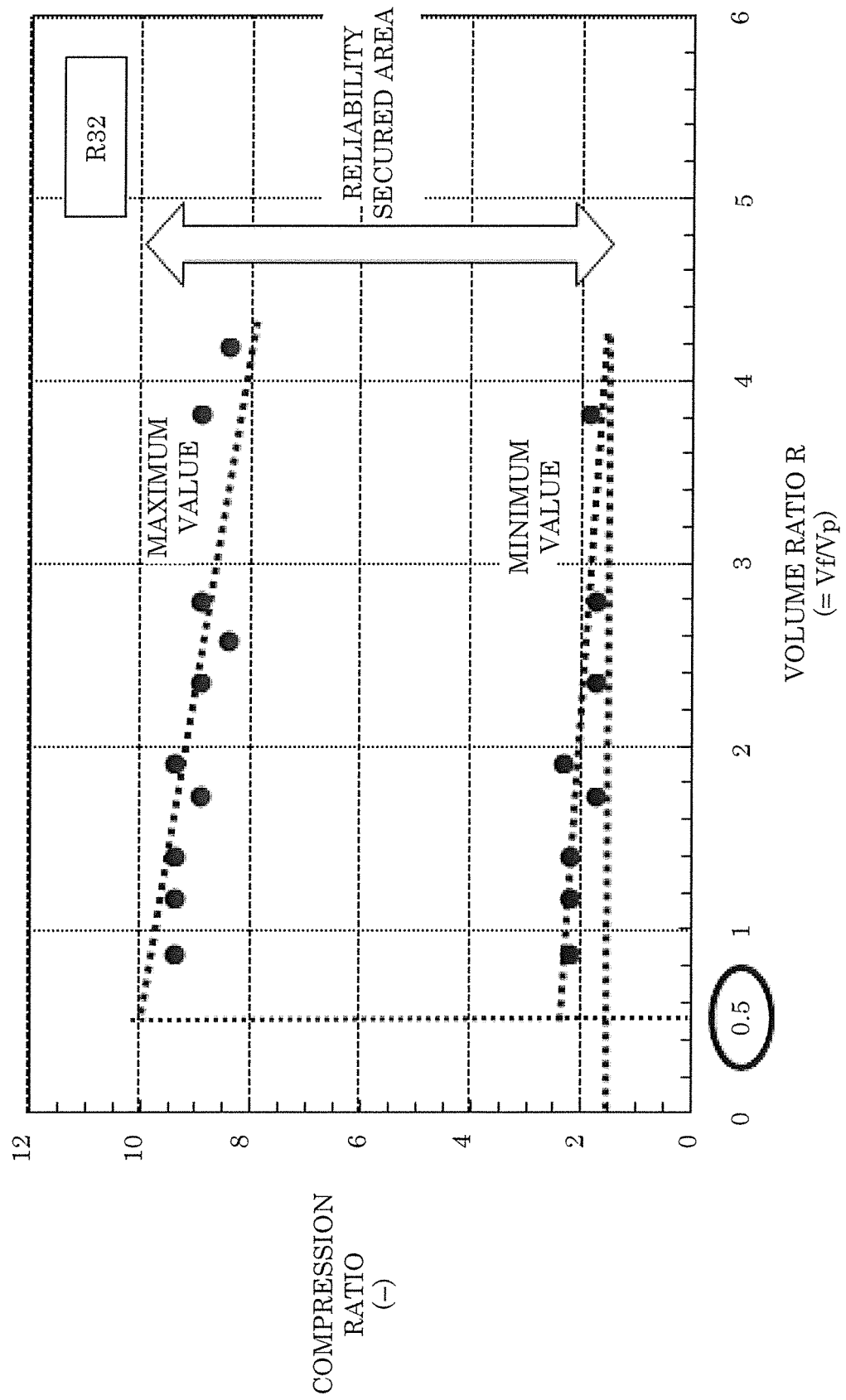


FIG. 6

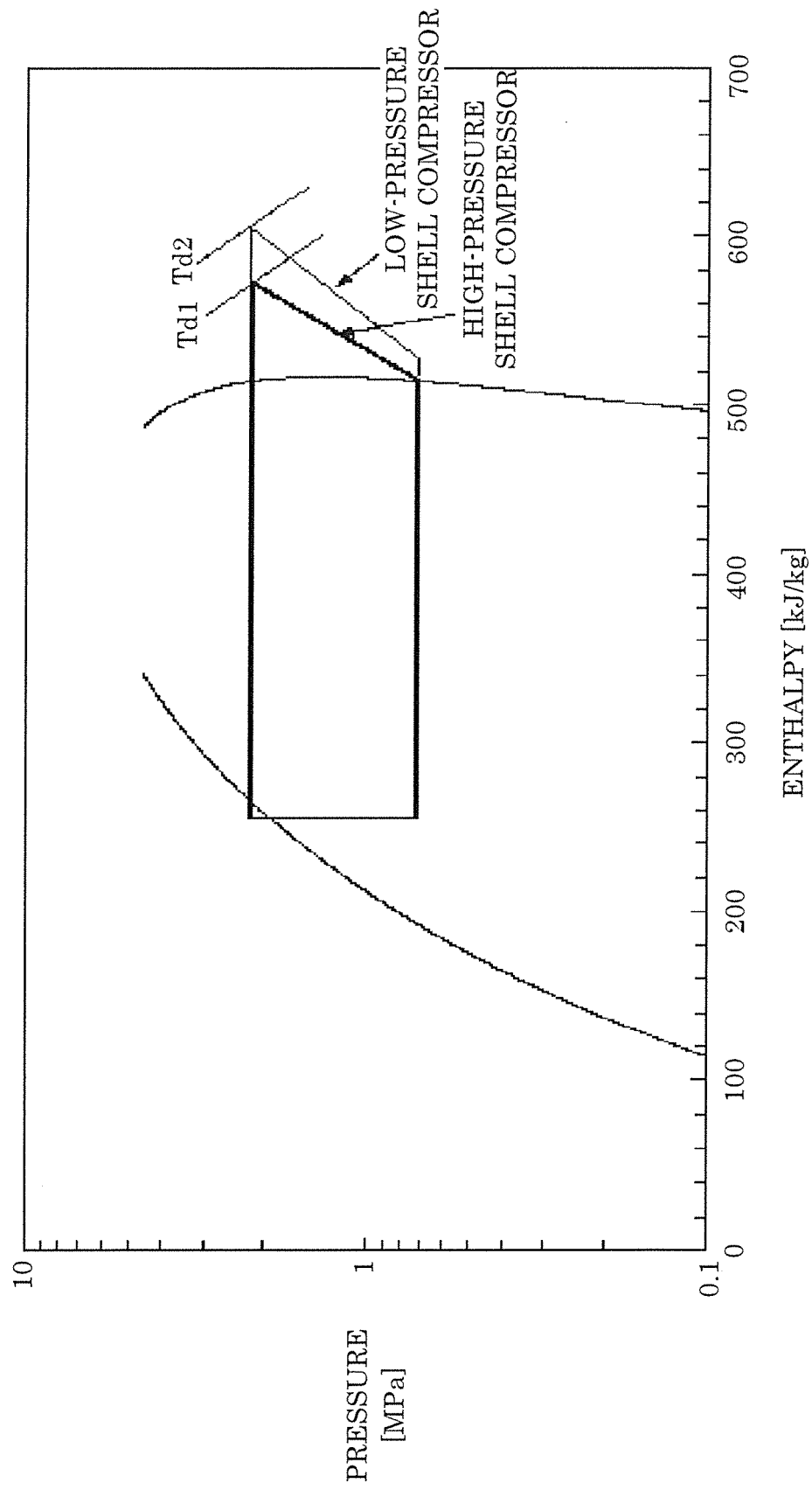


FIG. 7

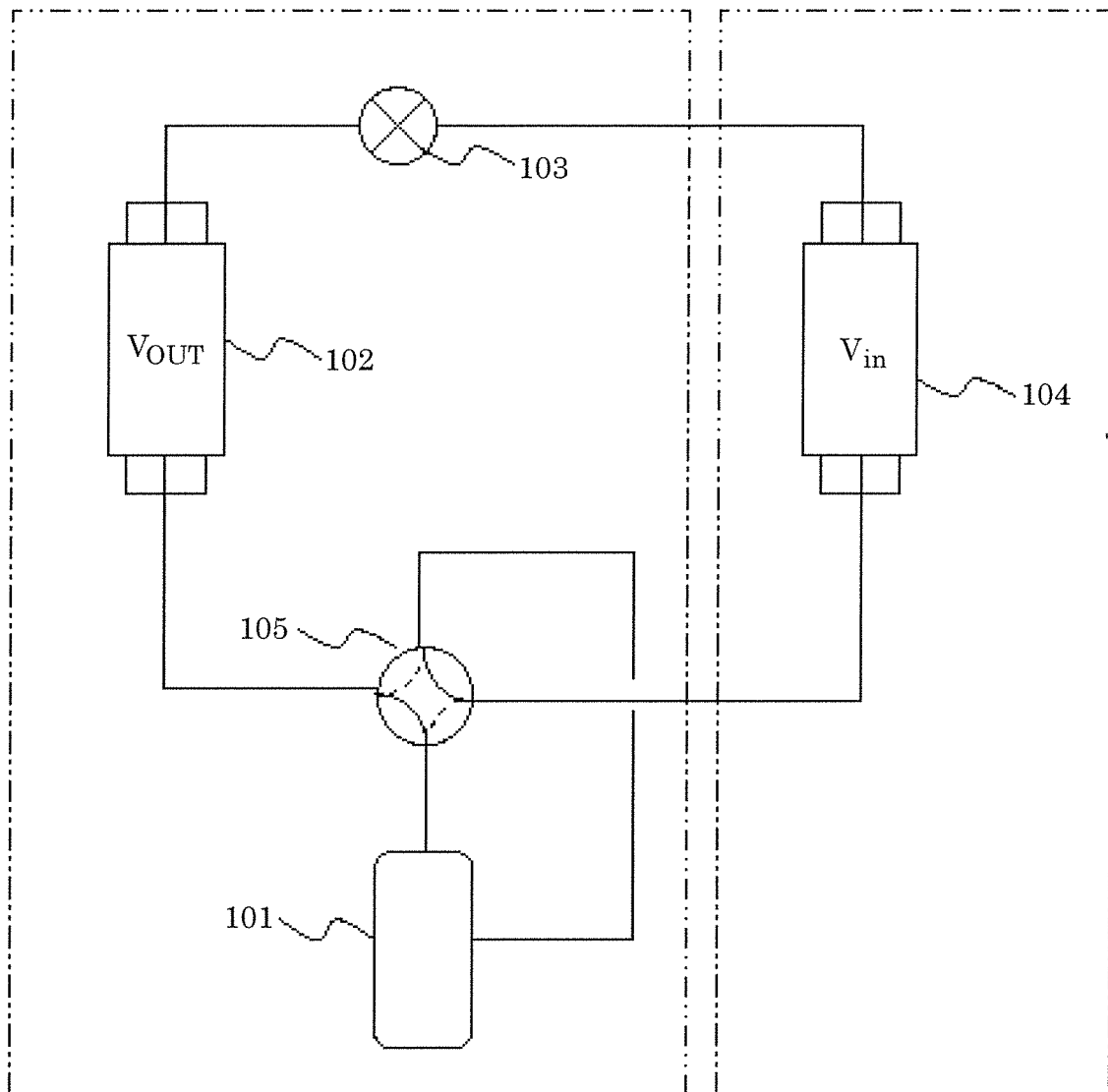
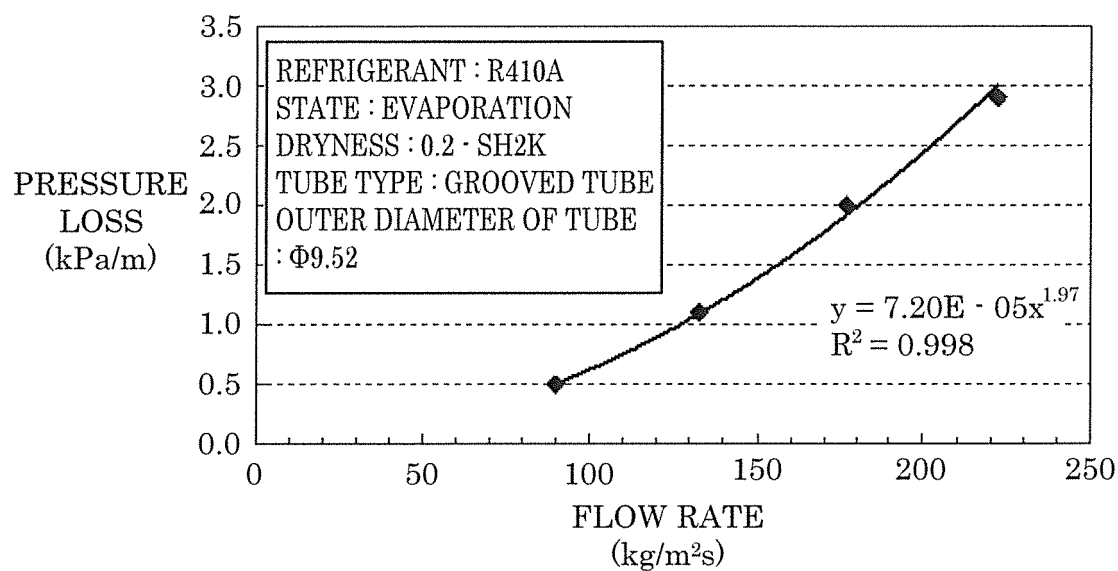
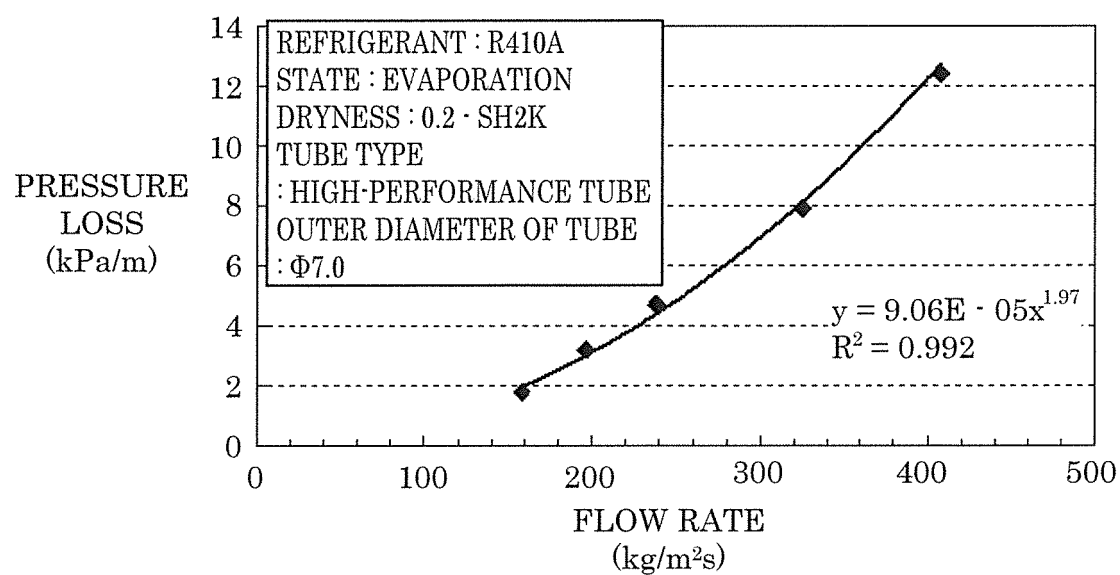


FIG. 8

(a)



(b)



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2015/004163

A. CLASSIFICATION OF SUBJECT MATTER

F25B1/00(2006.01)i, F25B1/02(2006.01)i, F25B1/04(2006.01)i, F25B39/00(2006.01)i, F25B39/02(2006.01)i, F25B39/04(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)

F25B1/00, F25B1/02, F25B1/04, F25B39/00, F25B39/02, F25B39/04

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.
X Y	WO 2013/051059 A1 (Mitsubishi Electric Corp.), 11 April 2013 (11.04.2013), paragraphs [0011] to [0032], [0043] to [0046], [0052], [0095]; fig. 4 & US 2014/0345310 A1 paragraphs [0028] to [0053], [0064] to [0067], [0073], [0089]; fig. 4 & EP 2765371 A1 & CN 103842747 A	1-2 3
Y	JP 2001-248922 A (Daikin Industries, Ltd.), 14 September 2001 (14.09.2001), claims; paragraphs [0011] to [0015]; fig. 1, 6 & US 2002/0189280 A1 paragraphs [0020] to [0024]; claims; fig. 1, 6 & EP 1243877 A1 & AU 1893601 A & CN 1415062 A & KR 10-0445810 B	1-3

☒ 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
16 October 2015 (16.10.15)

Date of mailing of the international search report
27 October 2015 (27.10.15)

Name and mailing address of the ISA/
Japan Patent Office
3-4-3, Kasumigaseki, Chiyoda-ku,
Tokyo 100-8915, Japan

Authorized officer

Telephone No.

INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2015/004163

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 2008-082653 A (Mitsubishi Electric Corp.), 10 April 2008 (10.04.2008), paragraphs [0021], [0046] to [0050]; fig. 3 (Family: none)	1-3
Y	JP 2010-002111 A (Mitsubishi Electric Corp.), 07 January 2010 (07.01.2010), paragraph [0015]; fig. 9 (Family: none)	3

Form PCT/ISA/210 (continuation of second sheet) (July 2009)

REFERENCES CITED IN THE DESCRIPTION

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- JP 2001248922 A [0008]

Non-patent literature cited in the description

- **MAMORU HOFUKU ; TWO OTHERS ; HITACHI CABLE, LTD.** Development of grooved tube which provides enhanced evaporation performances. *ou-hatsu seinou wo koujoushita naimen mizo tsuki kuda no kaihatsu*, January 2007, 59 [0009]
- Daigae reibai you kouseinou naimen mizo tsuki kuda HF taipu no kaihatsu. **YASUTOSHI MORI**. Furukawa denkou jihou (Furukawa Electric News Reports). Furukawa Electric Co., Ltd, 12 July 2000, 8 [0009]