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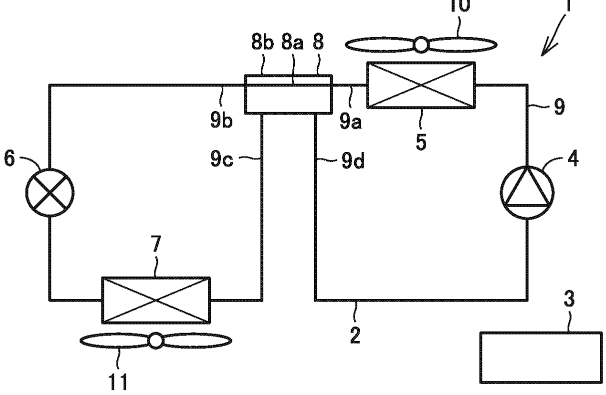
(54) REFRIGERATION CYCLE DEVICE

(57) A refrigeration cycle apparatus (1) of the present invention includes: a refrigerant circuit (2); and refrigerant. The refrigerant circuit (2) includes a compressor (4), a condenser (5), an expansion valve (6), an evaporator (7), and an internal heat exchanger (8). The refrigerant is a hydrocarbon refrigerant. The internal heat exchanger (8) includes: an inner pipe (8a); and an outer pipe (8b). The internal heat exchanger (8) is configured to cause heat exchange between the refrigerant flowing inside the inner pipe (8a) in a direction from the condenser (5) toward the expansion valve (6), and the refrigerant flowing inside the outer pipe (8b) and outside the inner pipe (8a) in a direction from the evaporator (7) toward the compressor (4). The refrigerant flowing inside the outer pipe (8b) and outside the inner pipe (8a) is entirely gas.

FIG.1



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Description

TECHNICAL FIELD

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

BACKGROUND ART

[0002] R32 refrigerant or R410A refrigerant has been conventionally used as refrigerant for a refrigeration cycle apparatus. There is known a refrigeration cycle apparatus in which R290 (propane) refrigerant having a global warming potential (GWP) smaller than that of R32 refrigerant or R410A refrigerant is used in a refrigerant circuit in order to reduce an influence on global warming. There is also known a refrigeration cycle apparatus including an internal heat exchanger for increasing a cooling capacity.

[0003] For example, Japanese Patent Laying-Open No. 2008-164245 (PTL 1) describes a refrigeration cycle apparatus including propane as refrigerant for use in a refrigerant circuit and including an internal heat exchanger. The refrigeration cycle apparatus described in this publication includes a compressor, a condenser, a heat exchanger, and an evaporator. The heat exchanger corresponds to the internal heat exchanger. The internal heat exchanger includes an inner pipe, and an outer pipe in which the inner pipe is inserted. The refrigerant delivered from the compressor through the condenser to the internal heat exchanger is delivered to the evaporator through the inner pipe in the heat exchanger. The refrigerant delivered to the evaporator returns to the compressor through the outer pipe in the internal heat exchanger. Heat exchange is performed between the refrigerant flowing through the inner pipe and the refrigerant flowing through the outer pipe in the internal heat exchanger.

CITATION LIST

PATENT LITERATURE

[0004] PTL 1: Japanese Patent Laying-Open No. 2008-164245

SUMMARY OF INVENTION

TECHNICAL PROBLEM

[0005] The publication above does not describe that the refrigerant flowing inside the outer pipe and outside the inner pipe of the internal heat exchanger is entirely gas refrigerant in the refrigeration cycle apparatus described in the publication above. When the refrigerant flowing inside the outer pipe and outside the inner pipe of the internal heat exchanger includes liquid refrigerant, it is difficult to increase a superheat degree of the refrigerant at an inlet of the compressor. Therefore, it is difficult to increase a coefficient of performance (COP), which is a ratio of consumed electric power to a capacity of a refrigeration cycle apparatus. When the refrigerant flowing inside the outer pipe and outside the inner pipe of the internal heat exchanger includes liquid refrigerant, it is difficult to reduce an amount of the refrigerant in the internal heat exchanger.

[0006] The present invention has been made in view of the above-described problem, and an object of the present invention is to provide a refrigeration cycle apparatus in which refrigerant having a small global warming potential can be used to increase a coefficient of performance of the refrigeration cycle apparatus and reduce an amount of the refrigerant in an internal heat exchanger.

SOLUTION TO PROBLEM

[0007] A refrigeration cycle apparatus of the present invention includes: a refrigerant circuit; and refrigerant. The refrigerant circuit includes a compressor, a condenser, an expansion valve, an evaporator, and an internal heat exchanger. The refrigerant flows in the refrigerant circuit in order of the compressor, the condenser, the internal heat exchanger, the expansion valve, the evaporator, and the internal heat exchanger. The refrigerant is a hydrocarbon refrigerant. The internal heat exchanger includes: an inner pipe connected to the condenser and the expansion valve; and an outer pipe connected to the evaporator and the compressor, the inner pipe being inserted in the outer pipe. The internal heat exchanger is configured to cause heat exchange between the refrigerant flowing inside the inner pipe in a direction from the condenser toward the expansion valve, and the refrigerant flowing inside the outer pipe and outside the inner pipe in a direction from the evaporator toward the compressor. The refrigerant flowing inside the outer pipe and outside the inner pipe is entirely gas.

ADVANTAGEOUS EFFECTS OF INVENTION

[0008] According to the refrigeration cycle apparatus of the present invention, the refrigerant is a hydrocarbon refrigerant and the refrigerant flowing inside the outer pipe and outside the inner pipe of the internal heat exchanger is entirely gas. Therefore, the refrigerant having a small global warming potential can be used. In addition, a coefficient of performance of the refrigeration cycle apparatus can be increased. Furthermore, an amount of the refrigerant in the internal heat exchanger can be reduced.

BRIEF DESCRIPTION OF DRAWINGS

[0009]

Fig. 1 is a configuration diagram showing a refrigeration cycle apparatus according to a first embodiment of the present invention.

Fig. 2 is a perspective view schematically showing a configuration of an internal heat exchanger of the refrigeration cycle apparatus according to the first embodiment of the present invention.

Fig. 3 is a cross-sectional view taken along line III-III in Fig. 2.

Fig. 4 is a graph showing a relationship between a suction SH and a theoretical COP of R290 refrigerant and R32 refrigerant.

Fig. 5 is a cross-sectional view schematically showing a flowing state of refrigerant in an internal heat exchanger in Comparative Example 1.

Fig. 6 is a cross-sectional view schematically showing a flowing state of refrigerant in an internal heat exchanger in Comparative Example 2.

Fig. 7 is a cross-sectional view schematically showing a flowing state of the refrigerant in the internal heat exchanger of the refrigeration cycle apparatus according to the first embodiment of the present invention.

Fig. 8 is a partial cross-sectional view taken along line VIII-VIII in Fig. 7.

Fig. 9 is a cross-sectional view schematically showing a flowing state of refrigerant in an internal heat exchanger of a refrigeration cycle apparatus according to a second embodiment of the present invention.

Fig. 10 is a partial cross-sectional view taken along line X-X in Fig. 9.

DESCRIPTION OF EMBODIMENTS

[0010] Embodiments of the present invention will be described hereinafter with reference to the drawings. In the following description, the same or corresponding portions are denoted by the same reference characters and description thereof will not be repeated in principle.

First Embodiment

[0011] A configuration of a refrigeration cycle apparatus 1 according to a first embodiment of the present invention will be described with reference to Fig. 1. Fig. 1 is a configuration diagram showing the refrigeration cycle apparatus according to the first embodiment of the present invention. The refrigeration cycle apparatus according to the first embodiment of the present invention is, for example, an air conditioner. As shown in Fig. 1, refrigeration cycle apparatus 1 according to the first embodiment of the present invention includes a refrigerant circuit 2, a controller 3, a condenser fan 10, an evaporator fan 11, and refrigerant.

[0012] Refrigerant circuit 2 includes a compressor 4, a condenser 5, an expansion valve 6, an evaporator 7, and an internal heat exchanger 8. Compressor 4, condenser 5, expansion valve 6, evaporator 7, and internal heat exchanger 8 are connected by a pipe 9. Refrigerant circuit 2 is thus formed. Refrigerant circuit 2 is configured to circulate the refrigerant. Refrigerant circuit 2 is configured such that a refrigeration cycle is performed in which the refrigerant circulates in order of compressor 4, condenser 5, internal heat exchanger 8, expansion valve 6, evaporator 7, and internal heat exchanger 8 while changing its phase.

[0013] The refrigerant flows in refrigerant circuit 2 in order of compressor 4, condenser 5, internal heat exchanger 8, expansion valve 6, evaporator 7, and internal heat exchanger 8. The refrigerant is such that a coefficient of performance of the refrigerant becomes higher as a suction superheat degree (suction SH) of compressor 4 becomes higher. The refrigerant is, for example, a hydrocarbon refrigerant (HC refrigerant). Specifically, the refrigerant is, for example, propane (R290), isobutane (R600a), pentane (R601), butane (R600), ethane (R170), or propylene (R1270).

[0014] Controller 3 is configured to control refrigerant circuit 2. Controller 3 is configured to control instruments, devices and the like of refrigeration cycle apparatus 1 by performing computation, instruction and the like. Controller 3 is electrically connected to compressor 4, expansion valve 6, condenser fan 10, evaporator fan 11 and the like and is configured to control operations thereof.

[0015] Compressor 4 is configured to compress and discharge suctioned gaseous refrigerant. Compressor 4 is configured to be capacity-variable. Compressor 4 is configured such that a frequency is changed based on an instruction from controller 3 to thereby adjust a rotation speed and change a capacity. In compressor 4, refrigerator oil (lubricating oil) is used. The refrigerator oil is, for example, polyalkylene glycol (PAG)-based oil having an ether bond, polyol ester (POE)-based oil having an ester bond, or the like.

[0016] Condenser 5 is configured to condense the refrigerant compressed by compressor 4. Condenser 5 is connected to compressor 4 and internal heat exchanger 8. Condenser 5 includes a heat transfer tube through which the refrigerant flows. Condenser 5 is, for example, a fin-and-tube-type heat exchanger including a plurality of fins and a circular or flat heat transfer tube passing through the plurality of fins.

[0017] Expansion valve 6 is configured to expand and decompress the liquid refrigerant condensed by condenser 5. The liquid refrigerant condensed by condenser 5 is expanded and decompressed by expansion valve 6, and thus, the refrigerant enters a gas-liquid two-phase state at an outlet of expansion valve 6. Expansion valve 6 is connected to condenser 5 and evaporator 7. Expansion valve 6 is, for example, an electric expansion valve configured to adjust a flow rate of the refrigerant based on an instruction from controller 3. An amount of the refrigerant flowing through expansion valve 6 is adjusted by adjusting a degree of opening of expansion valve 6.

[0018] Evaporator 7 is configured to evaporate the refrigerant decompressed by expansion valve 6. Evaporator 7 is connected to expansion valve 6 and internal heat exchanger 8. Evaporator 7 includes a heat transfer tube through which the refrigerant flows. Evaporator 7 is, for example, a fin-and-tube-type heat exchanger including a plurality of fins and a circular or flat heat transfer tube passing through the plurality of fins.

[0019] Internal heat exchanger 8 is configured to cause heat exchange between the refrigerant on the outlet side of condenser 5 and the refrigerant on the outlet side of evaporator 7. In internal heat exchanger 8, heat exchange is performed between the refrigerant condensed by condenser 5 and the refrigerant evaporated by evaporator 7.

[0020] Pipe 9 connects compressor 4, condenser 5, expansion valve 6, evaporator 7, and internal heat exchanger 8. Pipe 9 forms a gas-side refrigerant path and a liquid-side refrigerant path. Pipe 9 includes a first pipe portion 9a, a second pipe portion 9b, a third pipe portion 9c, and a fourth pipe portion 9d. First pipe portion 9a is connected to condenser 5 and internal heat exchanger 8. Second pipe portion 9b is connected to internal heat exchanger 8 and expansion valve 6. Third pipe portion 9c is connected to evaporator 7 and internal heat exchanger 8. Fourth pipe portion 9d is connected to internal heat exchanger 8 and compressor 4.

[0021] In cooling, condenser fan 10 is provided in a not-shown outdoor unit. Condenser fan 10 is configured to forcibly deliver outdoor air to condenser 5. Condenser fan 10 is attached to condenser 5 and is configured to supply air as a heat exchange fluid to condenser 5. Condenser fan 10 is configured such that a rotation speed of condenser fan 10 is adjusted based on an instruction from controller 3 to thereby adjust an amount of air flowing around condenser 5 and adjust an amount of heat exchange between the air and the refrigerant.

[0022] Evaporator fan 11 is provided in a not-shown indoor unit. Evaporator fan 11 is configured to forcibly deliver indoor air to evaporator 7. Evaporator fan 11 is attached to evaporator 7 and is configured to supply air as a heat exchange fluid to evaporator 7. Evaporator fan 11 is configured such that a rotation speed of evaporator fan 11 is adjusted based on an instruction from controller 3 to thereby adjust an amount of air flowing around evaporator 7 and adjust an amount of heat exchange between the air and the refrigerant.

[0023] A configuration of internal heat exchanger 8 will be described in detail with reference to Figs. 1 to 3.

[0024] As shown in Figs. 2 and 3, internal heat exchanger 8 is a double-pipe-type heat exchanger. Internal heat exchanger 8 includes an inner pipe 8a and an outer pipe 8b. Inner pipe 8a has a pipe shape. Outer pipe 8b has a pipe shape. Inner pipe 8a is inserted in outer pipe 8b. That is, inner pipe 8a is arranged within outer pipe 8b. A gap GP is provided between an outer circumferential surface of inner pipe 8a and an inner circumferential surface of outer pipe 8b. Gap GP may have an uniform dimension over an entire circumference in an outer circumferential direction of inner pipe 8a.

[0025] As shown in Figs. 1 to 3, inner pipe 8a is connected to condenser 5 and expansion valve 6. Inner pipe 8a is connected to condenser 5 with first pipe portion 9a being interposed, and is connected to expansion valve 6 with second pipe portion 9b being interposed. Inner pipe 8a is configured such that the high-pressure-side refrigerant flows therethrough. Outer pipe 8b is connected to evaporator 7 and compressor 4. Outer pipe 8b is connected to evaporator 7 with third pipe portion 9c being interposed, and is connected to compressor 4 with fourth pipe portion 9d being interposed. Outer pipe 8b is configured such that the low-pressure-side refrigerant flows therethrough.

[0026] Internal heat exchanger 8 is configured to cause heat exchange between the refrigerant flowing inside inner pipe 8a in a direction from condenser 5 toward expansion valve 6 and the refrigerant flowing inside outer pipe 8b and outside inner pipe 8a in a direction from evaporator 7 toward compressor 4. Internal heat exchanger 8 is configured to cause heat exchange, via a wall surface of inner pipe 8a, between the refrigerant flowing inside inner pipe 8a and the refrigerant flowing inside outer pipe 8b and outside inner pipe 8a. Internal heat exchanger 8 is configured to cause heat exchange, via the wall surface of inner pipe 8a, between the refrigerant flowing inside inner pipe 8a and the refrigerant flowing through gap GP.

[0027] In internal heat exchanger 8, the refrigerant flowing inside outer pipe 8b and outside inner pipe 8a is entirely gas. The refrigerant flowing through gap GP is entirely gas. The refrigerant flowing inside outer pipe 8b and outside inner pipe 8a is entirely in a dry state.

[0028] Next, an operation of refrigeration cycle apparatus 1 will be described with reference to Figs. 1 to 3. During a refrigeration cycle operation, the gaseous refrigerant compressed by compressor 4 is discharged from compressor 4 and delivered to condenser 5 through pipe 9 serving as the gas-side refrigerant path. In condenser 5, heat is released from the refrigerant flowing through the heat transfer tube to the air, and the refrigerant is thereby condensed. Thereafter, the refrigerant is delivered to internal heat exchanger 8 through first pipe portion 9a serving as the liquid-side refrigerant path. The refrigerant delivered to internal heat exchanger 8 through first pipe portion 9a flows through inner pipe 8a of internal heat exchanger 8, and then, is delivered to expansion valve 6 through second pipe portion 9b.

[0029] In expansion valve 6, the liquid refrigerant is decompressed to the refrigerant in a gas-liquid two-phase state. The refrigerant decompressed by expansion valve 6 is delivered to evaporator 7 through pipe 9 serving as the liquid-side refrigerant path. Thereafter, the refrigerant takes in heat from the air and evaporates in evaporator 7, and then, is delivered to internal heat exchanger 8 through third pipe portion 9c serving as the gas-side refrigerant path. The refrigerant delivered to internal heat exchanger 8 through third pipe portion 9c flows through outer pipe 8b of internal heat exchanger 8, and then, returns to compressor 4 through fourth pipe portion 9d.

[0030] In internal heat exchanger 8, heat exchange is performed between the refrigerant on the outlet side of condenser 5 (high-pressure-side refrigerant) flowing through inner pipe 8a and the refrigerant on the outlet side of evaporator 7 (low-pressure-side refrigerant) flowing through outer pipe 8b. Since a degree of dryness of the refrigerant at the outlet of evaporator 7 can be reduced by internal heat exchanger 8, the heat transfer performance of evaporator 7 is improved. As a result, a coefficient of performance (COP) of refrigeration cycle apparatus 1 is improved.

[0031] Next, functions and effects of refrigeration cycle apparatus 1 according to the first embodiment of the present invention will be described in comparison with Comparative Example 1 and Comparative Example 2.

[0032] Here, in refrigeration cycle apparatus 1 according to the first embodiment of the present invention, the R290 refrigerant is used as one example of the refrigerant. Comparative Example 1 is different from refrigeration cycle apparatus 1 according to the first embodiment of the present invention in that the refrigerant is R32. The R32 refrigerant has a global warming potential (GWP) greater than that of the R290 refrigerant. In addition, Comparative Example 1 is different from the refrigeration cycle apparatus according to the first embodiment of the present invention in that the low-pressure-side refrigerant flows through inner pipe 8a and the high-pressure-side refrigerant flows through outer pipe 8b in internal heat exchanger 8. That is, in Comparative Example 1, inner pipe 8a is connected to evaporator 7 and compressor 4 and outer pipe 8b is connected to condenser 5 and expansion valve 6 in internal heat exchanger 8.

[0033] Fig. 4 is a graph showing a relationship between a theoretical coefficient of performance (hereinafter, referred to as "theoretical COP") and a suction superheat degree (suction SH) of compressor 4 when each of the R290 refrigerant and the R32 refrigerant is used as the refrigerant for refrigerant circuit 2. The coefficient of performance (COP) is a ratio of consumed electric power to a capacity of refrigeration cycle apparatus 1.

[0034] Referring to Fig. 4, the theoretical COP of the R32 refrigerant decreases as the suction superheat degree (suction SH) of compressor 4 increases. In contrast, the theoretical COP of the R290 refrigerant increases as the suction superheat degree (SH) of compressor 4 increases. This is because the R290 refrigerant and the R32 refrigerant are different in properties. That is, as the suction superheat degree (suction SH) of compressor 4 increases, the coefficient of performance of the R290 refrigerant becomes superior to that of the R32 refrigerant.

[0035] Due to the properties of the R32 refrigerant, the coefficient of performance of the R32 refrigerant is higher when the suction superheat degree (suction SH) of compressor 4 is zero than when the suction superheat degree (suction SH) of compressor 4 is higher than zero. Therefore, in order to increase the coefficient of performance, the low-pressure-side refrigerant is brought into a wet state in internal heat exchanger 8 so as to prevent the suction superheat degree (suction SH) of compressor 4 from becoming higher than zero.

[0036] Fig. 5 is a cross-sectional view showing a flowing state of the refrigerant in internal heat exchanger 8 in Comparative Example 1. Referring to Fig. 5, in internal heat exchanger 8 in Comparative Example 1, refrigerant R1 flowing through inner pipe 8a is low-pressure-side refrigerant, and refrigerant R2 flowing through outer pipe 8b is high-pressure-side refrigerant. Low-pressure-side refrigerant R1 flowing through inner pipe 8a is in a gas-liquid two-phase state. Low-pressure-side refrigerant R1 flowing through inner pipe 8a forms an annular flow. That is, as for the low-pressure-side refrigerant flowing through inner pipe 8a, gas refrigerant Ra flows through a central portion of inner pipe 8a, and liquid refrigerant Rb flows through an outer portion along the wall surface of inner pipe 8a. Since liquid refrigerant Rb comes into contact with the wall surface of inner pipe 8a serving as a heat transfer surface, the heat transfer performance increases. However, since the refrigerant in Comparative Example 1 is the R32 refrigerant, the global warming potential of the refrigerant is greater than that of the R290 refrigerant. Therefore, in Comparative Example 1, the global warming potential of the refrigerant cannot be reduced.

[0037] Fig. 6 is a cross-sectional view showing a flowing state of the refrigerant in internal heat exchanger 8 in Comparative Example 2. Referring to Fig. 6, in internal heat exchanger 8 in Comparative Example 2, refrigerant R1 flowing through inner pipe 8a is low-pressure-side refrigerant, and refrigerant R2 flowing through outer pipe 8b is high-pressure-side refrigerant. In this respect, Comparative Example 2 is different from refrigeration cycle apparatus 1 according to the first embodiment of the present invention. The refrigerant in Comparative Example 2 is propane (R290).

[0038] When the superheat degree of the refrigerant at the outlet of evaporator 7 is zero (SH=0), the performance of evaporator 7 is theoretically high. However, due to properties of the propane (R290) refrigerant, the coefficient of performance becomes superior as the suction superheat degree (suction SH) of compressor 4 becomes higher. Therefore, in order to increase the suction superheat degree (suction SH) of compressor 4 while keeping the superheat degree of the refrigerant at the outlet of evaporator 7 zero (SH=0), the superheat degree of the refrigerant at the low-pressure-side inlet of internal heat exchanger 8 may only be zero.

[0039] When refrigeration cycle apparatus 1 including propane (R290) used as the refrigerant is operated such that the coefficient of performance of refrigeration cycle apparatus 1 becomes higher, the superheat degree of the refrigerant at the outlet of evaporator 7 becomes near zero. In this case, the superheat degree at the low-pressure-side outlet of internal heat exchanger 8, i.e., at the inlet of compressor 4 becomes higher than or equal to zero. In addition, the refrigerant at the low-pressure-side inlet of internal heat exchanger 8 is gas. In this case, refrigerant R1 flowing through inner pipe 8a of internal heat exchanger 8 does not include liquid refrigerant, and thus, refrigerator oil 20 is likely to precipitate on an inner surface of the wall surface of inner pipe 8a. When refrigerator oil 20 is precipitated on the wall surface of inner pipe 8a of internal heat exchanger 8, refrigerator oil 20 precipitated on the wall surface of inner pipe 8a of internal heat exchanger 8 serves as a thermal resistance, and thus, the heat transfer performance of internal heat exchanger 8 decreases.

[0040] Figs. 7 and 8 are cross-sectional views showing a flowing state of the refrigerant in internal heat exchanger 8 of refrigeration cycle apparatus 1 according to the first embodiment of the present invention. Referring to Figs. 7 and 8, in internal heat exchanger 8 of refrigeration cycle apparatus 1 according to the first embodiment of the present invention, refrigerant R1 flowing through inner pipe 8a is high-pressure-side refrigerant, and refrigerant R2 flowing through outer pipe 8b is low-pressure-side refrigerant.

[0041] The wall surface of inner pipe 8a serves as a heat transfer surface where heat exchange is performed between high-pressure-side refrigerant R1 flowing through inner pipe 8a and low-pressure-side refrigerant R2 flowing through outer pipe 8b in internal heat exchanger 8. In addition to the wall surface of inner pipe 8a serving as the heat transfer surface where heat exchange is performed between the low-pressure-side refrigerant flowing through outer pipe 8b and the high-pressure-side refrigerant flowing through inner pipe 8a, there exists a wall surface of outer pipe 8b serving as a heat transfer surface where heat exchange is performed between the low-pressure-side refrigerant flowing through outer pipe 8b and the air outside outer pipe 8b. Therefore, an area of the wall surface on which refrigerator oil 20 is precipitated is larger in refrigeration cycle apparatus 1 according to the first embodiment of the present invention than in Comparative Example 2. Thus, an oil amount of the refrigerator oil precipitated on the wall surface of inner pipe 8a serving as the heat transfer surface decreases. Therefore, the refrigerator oil precipitated on the wall surface of inner pipe 8a serves as a thermal resistance, and thus, a reduction in heat transfer performance of internal heat exchanger 8 can be suppressed.

[0042] That is, in refrigeration cycle apparatus 1 according to the first embodiment of the present invention, the propane (R290) refrigerant is used, and the high-pressure-side refrigerant flows through inner pipe 8a of internal heat exchanger 8 and the low-pressure-side refrigerant flows through outer pipe 8b of internal heat exchanger 8. Furthermore, the refrigerant at the low-pressure-side inlet of internal heat exchanger 8 is in a dry state. That is, the superheat degree of the refrigerant at the low-pressure-side inlet of internal heat exchanger 8 is zero. Therefore, a reduction in heat transfer performance caused by precipitation of the refrigerator oil in internal heat exchanger 8 is suppressed. Thus, the operation with a high coefficient of performance can be achieved in refrigeration cycle apparatus 1.

[0043] In refrigeration cycle apparatus 1 according to the first embodiment of the present invention, the refrigerant is a hydrocarbon refrigerant (HC refrigerant). Therefore, the refrigerant having a small global warming potential (GWP) can be used. In addition, the refrigerant flowing inside outer pipe 8b and outside inner pipe 8a of internal heat exchanger 8 is entirely gas. Therefore, the superheat degree of the refrigerant at the inlet of compressor 4 can be increased, as compared with the case in which the refrigerant flowing inside outer pipe 8b and outside inner pipe 8a of internal heat exchanger 8 includes liquid refrigerant. Thus, the coefficient of performance (COP) of refrigeration cycle apparatus 1 can be increased. Furthermore, the superheat degree of the refrigerant at the outlet of outer pipe 8b of internal heat exchanger 8 can be increased, as compared with the case in which the refrigerant flowing inside outer pipe 8b and outside inner pipe 8a of internal heat exchanger 8 includes liquid refrigerant. Therefore, the amount of the refrigerant in internal heat exchanger 8 can be reduced.

[0044] In refrigeration cycle apparatus 1 according to the first embodiment of the present invention, the refrigerant is an HC refrigerant. Therefore, the global warming potential (GWP) of the refrigerant can be reduced.

[0045] In refrigeration cycle apparatus 1 according to the first embodiment of the present invention, expansion valve 6 is an electric expansion valve configured to adjust a flow rate of the refrigerant. Therefore, the flow rate of the refrigerant can be adjusted by the electric expansion valve.

Second Embodiment

[0046] Refrigeration cycle apparatus 1 according to a second embodiment of the present invention has the same configuration, operation and effect as those of above-described refrigeration cycle apparatus 1 according to the first embodiment of the present invention, unless otherwise stated.

[0047] Referring to Figs. 9 and 10, refrigeration cycle apparatus 1 according to the second embodiment of the present invention is different in a configuration of outer pipe 8b of internal heat exchanger 8 from refrigeration cycle apparatus 1 according to the first embodiment of the present invention.

[0048] In refrigeration cycle apparatus 1 according to the second embodiment of the present invention, a groove 30 is provided in an inner surface of outer pipe 8b of internal heat exchanger 8. Groove 30 may be provided over an entire circumference of the inner surface of outer pipe 8b of internal heat exchanger 8. Groove 30 may be configured to be serrated. Inner pipe 8a of internal heat exchanger 8 is not provided with groove 30. That is, no groove is provided in an inner surface and an outer surface of inner pipe 8a of internal heat exchanger 8.

[0049] Since only outer pipe 8b of internal heat exchanger 8 is provided with groove 30, refrigerator oil 20 is likely to precipitate in groove 30, which is a portion that does not contribute to heat transfer between the refrigerant flowing through inner pipe 8a and the refrigerant flowing through outer pipe 8b in internal heat exchanger 8. As a result, a reduction in heat transfer performance caused by the refrigerator oil precipitated on the wall surface of inner pipe 8a can be suppressed, as compared with the first embodiment.

[0050] In refrigeration cycle apparatus 1 according to the present embodiment, groove 30 is provided in the inner surface of outer pipe 8b of internal heat exchanger 8. Since groove 30 results in an increase in heat transfer area of outer pipe 8b, refrigerator oil 20 is likely to precipitate in groove 30. Therefore, a reduction in heat transfer performance caused by the refrigerator oil precipitated on the wall surface of inner pipe 8a can be suppressed.

[0051] In refrigeration cycle apparatus 1 according to the present embodiment, groove 30 is configured to be serrated. Therefore, the refrigerator oil is likely to precipitate on the bottom of the serrated configuration.

[0052] It should be understood that the embodiments disclosed herein are illustrative and non-restrictive in every respect. The scope of the present invention is defined by the terms of the claims, rather than the description above, and is intended to include any modifications within the scope and meaning equivalent to the terms of the claims.

REFERENCE SIGNS LIST

[0053] 1 refrigeration cycle apparatus; 2 refrigerant circuit; 3 controller; 4 compressor; 5 condenser; 6 expansion valve; 7 evaporator; 8 internal heat exchanger; 8a inner pipe; 8b outer pipe; 9 pipe; 10 condenser fan; 11 evaporator fan; 20 refrigerator oil; 30 groove.

Claims

1. A refrigeration cycle apparatus comprising:

a refrigerant circuit comprising a compressor, a condenser, an expansion valve, an evaporator, and an internal heat exchanger; and
refrigerant flowing in the refrigerant circuit in order of the compressor, the condenser, the internal heat exchanger, the expansion valve, the evaporator, and the internal heat exchanger,
the refrigerant being a hydrocarbon refrigerant,
the internal heat exchanger comprising:

an inner pipe connected to the condenser and the expansion valve; and
an outer pipe connected to the evaporator and the compressor, the inner pipe being inserted in the outer pipe,

the internal heat exchanger being configured to cause heat exchange between

the refrigerant flowing inside the inner pipe in a direction from the condenser toward the expansion valve, and
the refrigerant flowing inside the outer pipe and outside the inner pipe in a direction from the evaporator toward the compressor,

the refrigerant flowing inside the outer pipe and outside the inner pipe being entirely gas.

2. The refrigeration cycle apparatus according to claim 1, wherein
a groove is provided in an inner surface of the outer pipe.

3. The refrigeration cycle apparatus according to claim 2, wherein
the groove is configured to be serrated.

4. The refrigeration cycle apparatus according to any one of claims 1 to 3, wherein
the refrigerant is an HC refrigerant.

5. The refrigeration cycle apparatus according to any one of claims 1 to 4, wherein
the expansion valve is an electric expansion valve configured to adjust a flow rate of the refrigerant.

FIG.1

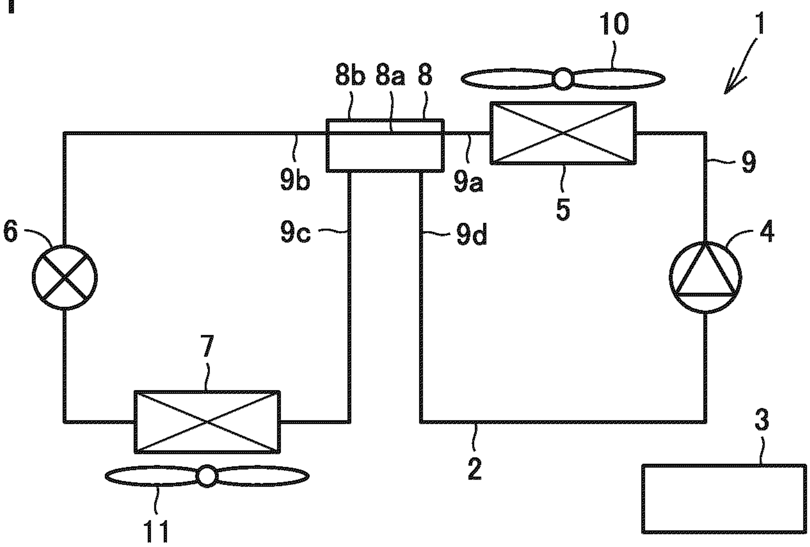


FIG.2

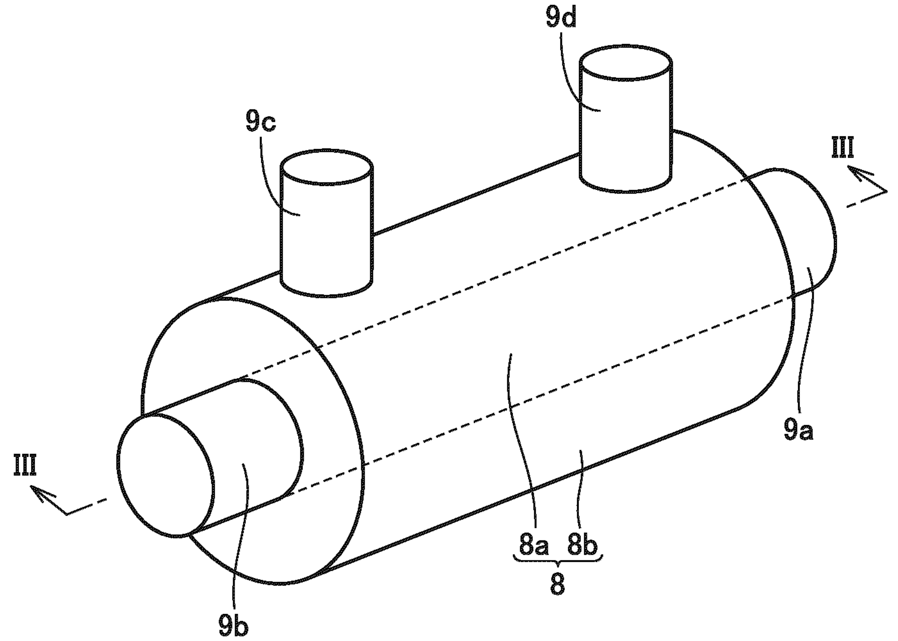


FIG.3

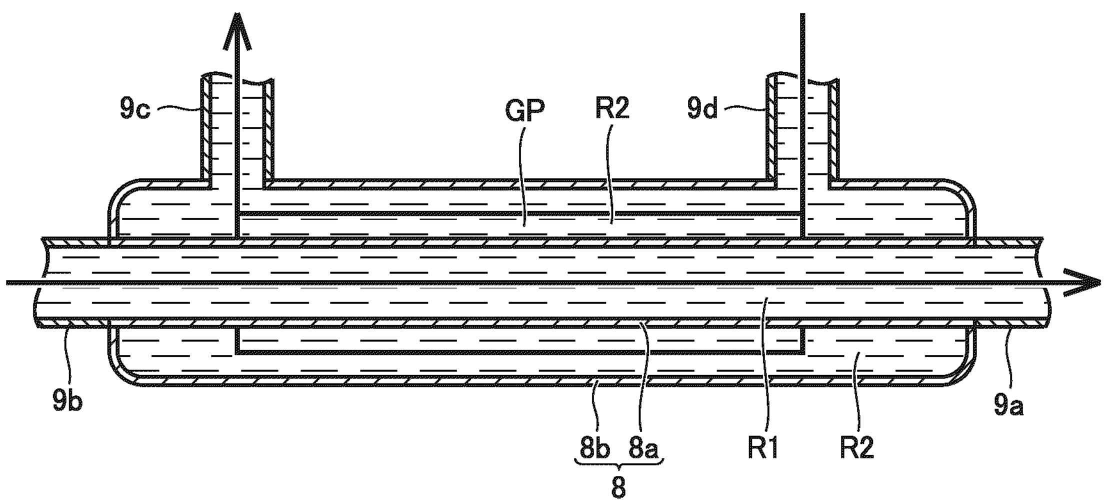
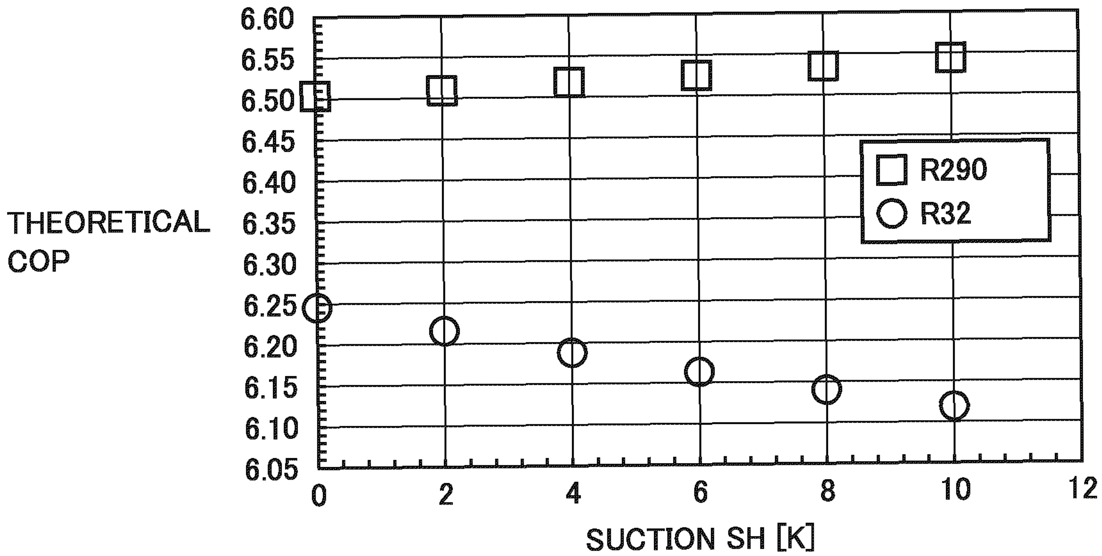


FIG.4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THEORETICAL COP
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FIG.5

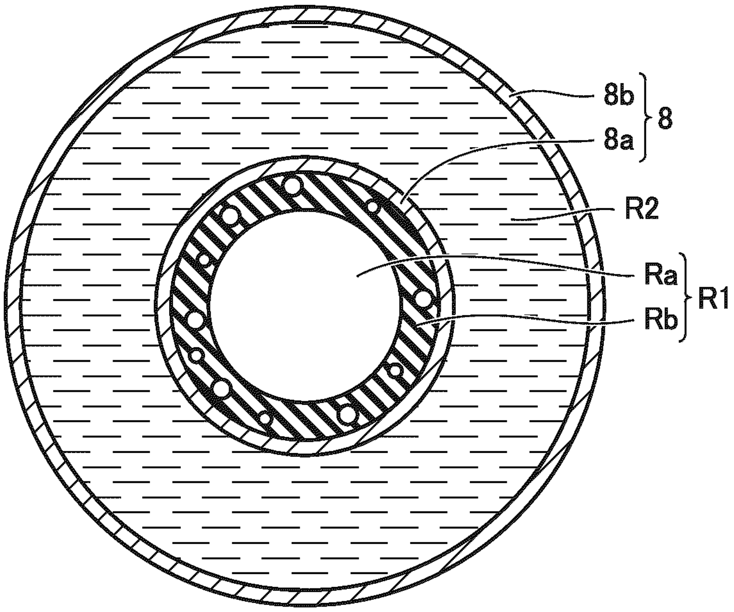


FIG.6

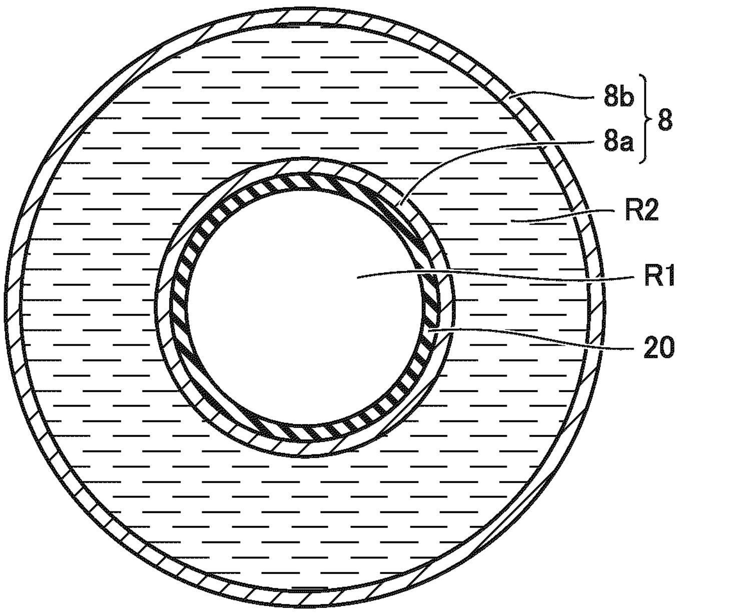


FIG.7

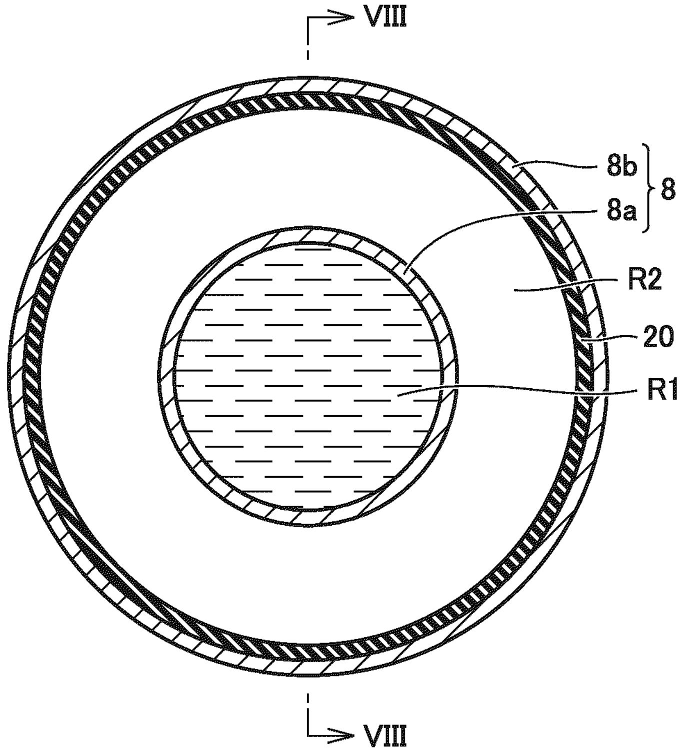


FIG.8

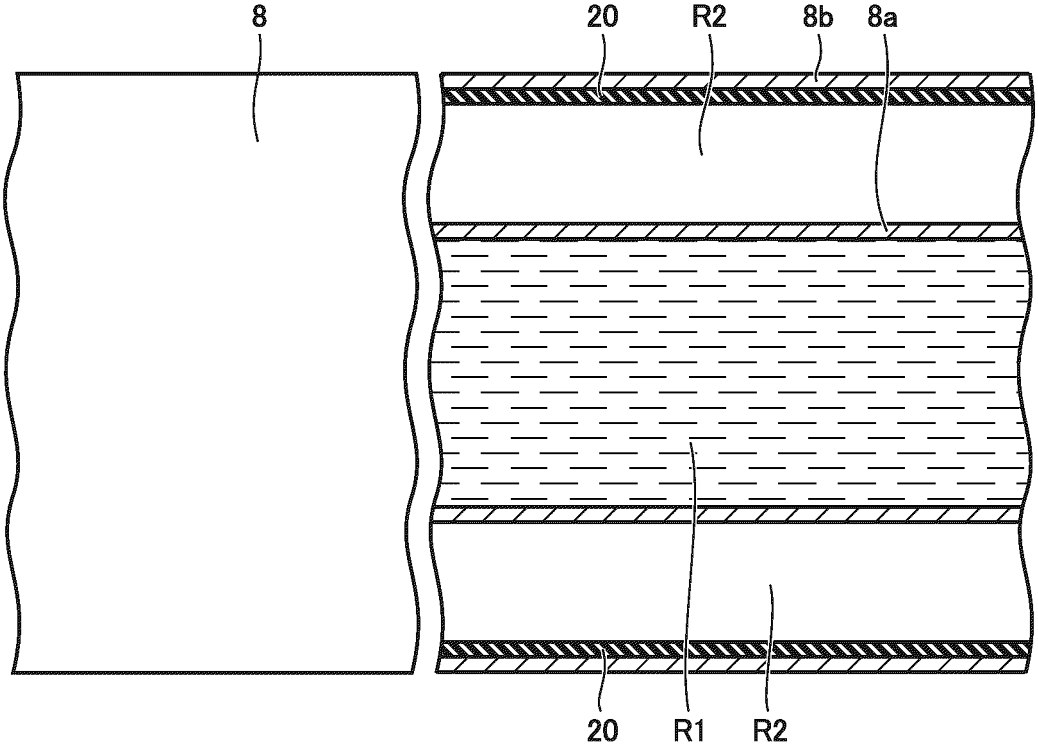


FIG.9

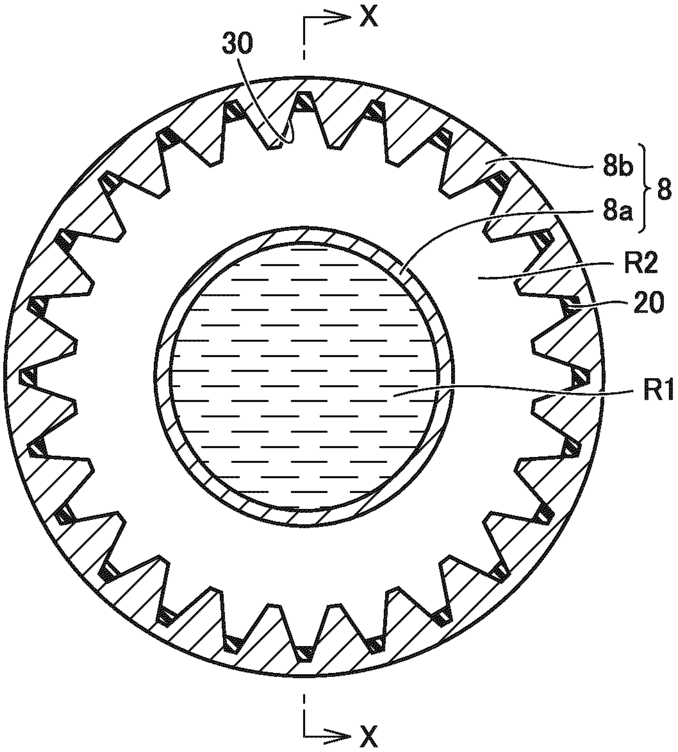
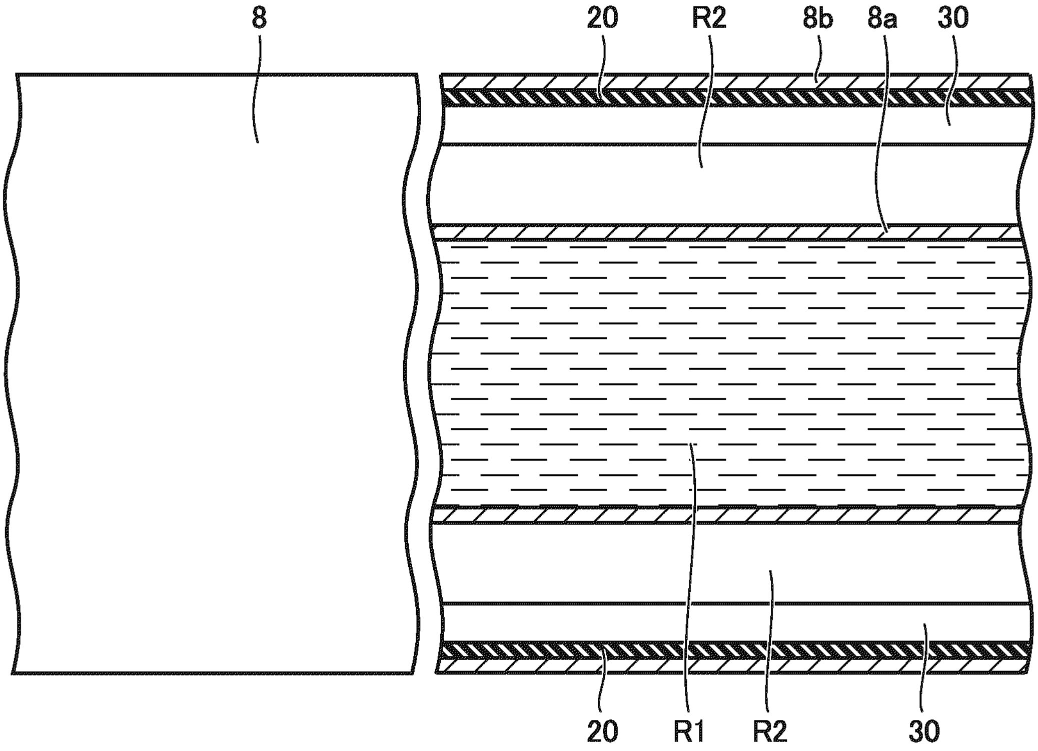


FIG.10



INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2019/000356

A. CLASSIFICATION OF SUBJECT MATTER
Int.Cl. F25B1/00(2006.01)i, F28D7/10(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)
Int.Cl. F25B1/00, F28D7/10

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Published examined utility model applications of Japan	1922-1996
Published unexamined utility model applications of Japan	1971-2019
Registered utility model specifications of Japan	1996-2019
Published registered utility model applications of Japan	1994-2019

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 2008-164245 A (KOBELCO & MATERIALS COPPER TUBE CO., LTD.) 17 July 2008, paragraphs [0031], [0055]-[0056], fig. 3 & KR 10-2008-0063150 A & CN 101210783 A	1-5
Y	JP 2014-178110 A (MITSUBISHI ELECTRIC CORP.) 25 September 2014, paragraphs [0059]-[0069] (Family: none)	1-5
Y	JP 2004-12127 A (MITSUBISHI ELECTRIC CORP.) 15 January 2004, paragraphs [0065]-[0066], [0074]-[0077] (Family: none)	1-5

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

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"O" document referring to an oral disclosure, use, exhibition or other means
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"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
30 January 2019 (30.01.2019)	12 February 2019 (12.02.2019)

Name and mailing address of the ISA/	Authorized officer
Japan Patent Office 3-4-3, Kasumigaseki, Chiyoda-ku, Tokyo 100-8915, Japan	Telephone No.

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INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2019/000356

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 2000-179959 A (DENSO CORP.) 30 June 2000, paragraphs [0013], [0029], fig. 1 & US 6233969 B1, column 2, line 64 to column 3, line 3, column 5, lines 51-58, fig. 1 & DE 19958226 A1	1-5
Y	KR 20-0420568 Y1 04 July 2006, fig. 1 (Family: none)	1-5
A	JP 2010-127498 A (NIPPON SOKEN, INC.) 10 June 2010, entire text, all drawings (Family: none)	1-5
A	JP 2003-314927 A (MATSUSHITA ELECTRIC INDUSTRIAL CO., LTD.) 06 November 2003, entire text, all drawings (Family: none)	1-5

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REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 2008164245 A **[0003] [0004]**