



(11)

**EP 4 350 252 A1**

(12)

**EUROPEAN PATENT APPLICATION**  
published in accordance with Art. 153(4) EPC

(43) Date of publication:

**10.04.2024 Bulletin 2024/15**

(21) Application number: **21943071.7**

(22) Date of filing: **28.05.2021**

(51) International Patent Classification (IPC):

**F25B 39/00** <sup>(2006.01)</sup> **F25B 39/04** <sup>(2006.01)</sup>

**F28D 1/053** <sup>(2006.01)</sup> **F28F 1/02** <sup>(2006.01)</sup>

**F28F 17/00** <sup>(2006.01)</sup> **F25B 47/02** <sup>(2006.01)</sup>

(52) Cooperative Patent Classification (CPC):

**F28D 1/05391; F25B 39/00; F25B 39/04;**

**F25B 47/02; F28D 1/053; F28F 1/02; F28F 1/126;**

**F28F 17/00; F28F 1/022; F28F 1/40; F28F 2215/04**

(86) International application number:

**PCT/JP2021/020313**

(87) International publication number:

**WO 2022/249425 (01.12.2022 Gazette 2022/48)**

(84) Designated Contracting States:

**AL AT BE BG CH CY CZ DE DK EE ES FI FR GB  
GR HR HU IE IS IT LI LT LU LV MC MK MT NL NO  
PL PT RO RS SE SI SK SM TR**

Designated Extension States:

**BA ME**

Designated Validation States:

**KH MA MD TN**

(71) Applicant: **MITSUBISHI ELECTRIC  
CORPORATION**

**Chiyoda-ku**

**Tokyo 100-8310 (JP)**

(72) Inventors:

• **ONAKA, Yoji**

**Tokyo 100-8310 (JP)**

• **SAIKUSA, Tetsuji**

**Tokyo 100-8310 (JP)**

• **ADACHI, Rihito**

**Tokyo 100-8310 (JP)**

• **KISHIDA, Nanami**

**Tokyo 100-8310 (JP)**

• **GOMYO, Taisaku**

**Tokyo 100-8310 (JP)**

• **NAKAO, Yuki**

**Tokyo 100-8310 (JP)**

• **KASAKI, Shingo**

**Tokyo 100-8310 (JP)**

• **KIBE, Atsushi**

**Tokyo 100-8310 (JP)**

• **MORIMOTO, Hiroyuki**

**Tokyo 100-8310 (JP)**

(74) Representative: **Witte, Weller & Partner**

**Patentanwälte mbB**

**Postfach 10 54 62**

**70047 Stuttgart (DE)**

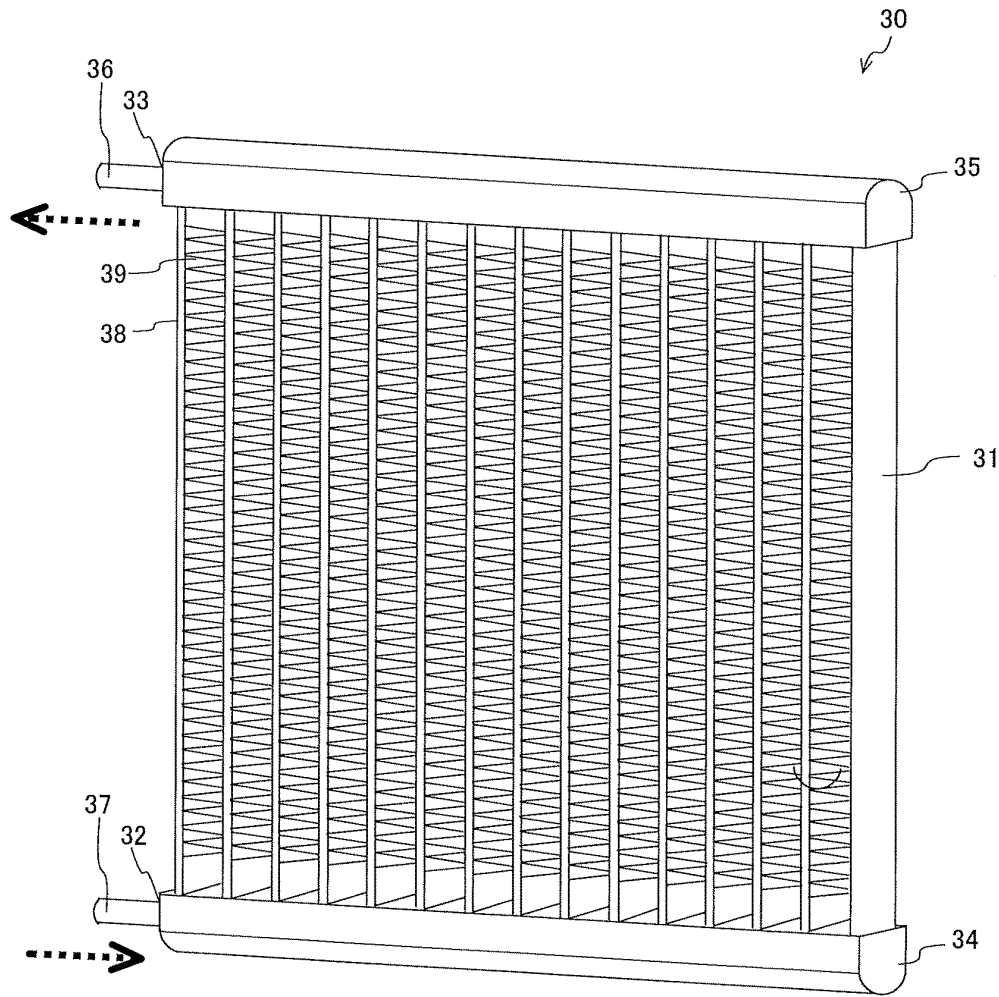
(54) **HEAT EXCHANGER, AIR CONDITIONER OUTDOOR UNIT EQUIPPED WITH HEAT EXCHANGER, AND AIR CONDITIONER EQUIPPED WITH AIR CONDITIONER OUTDOOR UNIT**

(57) A heat exchanger mounted in an outdoor unit of an air-conditioning apparatus includes at least one heat exchanger core and a hot-gas refrigerant inlet formed in lower part of the heat exchanger and in which refrigerant flows when the heat exchanger functions as a condenser. Each heat exchanger core includes flat tubes extending in an up-down direction and are placed along a direction of flow of air. When a total flow passage cross-sectional

area of each heat exchanger core is  $A \text{ [m}^2\text{]} = a \times N \text{ [m}^2\text{]}$ , where  $a \text{ [m}^2\text{]}$  is a flow passage cross-sectional area of each flat tube and  $N$  is a number of the flat tubes, a height of each of the heat exchanger cores is  $H \text{ [m]}$ , a differential pressure of a refrigerant flow passage is  $\Delta P_{\text{HEX}}$ , and a liquid head is  $\Delta P_{\text{HEAD}}$ ,  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}} = (5.94635 \times 10^{-4} \times A^{-175030})/(8.4303H + 0.8779) > 1$  is satisfied.

**EP 4 350 252 A1**

FIG. 2



**Description**

## Technical Field

5 **[0001]** The present disclosure relates to a heat exchanger including a plurality of flat tubes, an outdoor unit of an air-conditioning apparatus including the heat exchanger, and an air-conditioning apparatus including the outdoor unit of the air-conditioning apparatus.

## Background Art

10 **[0002]** Conventionally, there has been a heat exchanger including a plurality of flat tubes that extend in a vertical direction and that are arrayed at spacings in a horizontal direction, a plurality of fins that are each connected across a space between adjacent ones of the flat tubes and that transfer heat to the flat tubes, and headers provided at upper and lower ends of the plurality of flat tubes (see, for example, Patent Literature 1).

15 **[0003]** The heat exchanger of Patent Literature 1 is mounted in an outdoor unit of an air-conditioning apparatus that is capable of both a cooling operation and a heating operation. Moreover, in a case in which a heating operation is performed in a low-temperature environment in which the outside air temperature is low and the surface temperature of the heat exchanger is lower than or equal to 0 degrees C, frost forms on the heat exchanger. Therefore, when a particular or larger amount of frost forms on the heat exchanger, a defrosting operation is performed to melt the frost on the surface  
20 of the heat exchanger. In a defrosting operation, frost is removed by causing high-temperature and high-pressure gas refrigerant to flow in from one of the headers and flow to the flat tubes.

## Citation List

25 Patent Literature

**[0004]** Patent Literature 1: Japanese Unexamined Patent Application Publication No. 2018-96638

## Summary of Invention

30 Technical Problem

**[0005]** During a defrosting operation in a conventional heat exchanger such as that of Patent Literature 1, high-temperature and high-pressure gas refrigerant flows in from the lower header, flows as upward flows through the flat tubes, becomes cooler as it flows through the flat tubes, and increases its liquid phase as it flows further downstream. Moreover, there has undesirably been deterioration of defrosting performance due to the occurrence of liquid retention in which when the high-temperature and high-pressure gas refrigerant flows as upward flows through the flat tubes, refrigerant liquefied by the effect of gravity is undesirably retained without being able to go upward.

40 **[0006]** The present disclosure was made to solve such a problem and has as an object to provide a heat exchanger configured to reduce deterioration of defrosting performance, an outdoor unit of an air-conditioning apparatus including the heat exchanger, and an air-conditioning apparatus including the outdoor unit of the air-conditioning apparatus.

## Solution to Problem

45 **[0007]** A heat exchanger according to an embodiment of the present disclosure is a heat exchanger that is mounted in an outdoor unit of an air-conditioning apparatus. The heat exchanger includes one heat exchanger core or two or more heat exchanger cores each including a plurality of flat tubes that extend in an up-down direction and through which refrigerant flows as upward flows when the heat exchanger functions as a condenser. The two or more heat exchanger cores are placed along a direction of flow of air. In a case in which a total flow passage cross-sectional area of each of the heat exchanger cores is defined as  $A \text{ [m}^2\text{]} = a \times N \text{ [m}^2\text{]}$ , where  $a \text{ [m}^2\text{]}$  is a flow passage cross-sectional area of each of the flat tubes and  $N$  is a number of the flat tubes, a height of each of the heat exchanger cores is defined as  $H \text{ [m]}$ , a differential pressure of a refrigerant flow passage is defined as  $\Delta P_{\text{HEX}}$ , and a liquid head is defined as  $\Delta P_{\text{HEAD}}$ ,  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}} = (5.94635 \times 10^{-4} \times A^{-1.75030})/(8.4303H + 0.8779) > 1$  is satisfied.

55 **[0008]** Further, an outdoor unit of an air-conditioning apparatus according to an embodiment of the present disclosure includes the heat exchanger.

**[0009]** Further, an air-conditioning apparatus according to an embodiment of the present disclosure includes the outdoor unit of the air-conditioning apparatus, an indoor unit of an air-conditioning apparatus, and a refrigerant circuit that is constituted by the outdoor unit of the air-conditioning apparatus and the indoor unit of the air-conditioning apparatus

and through which refrigerant circulates.

#### Advantageous Effects of Invention

**[0010]** According to a heat exchanger according to an embodiment of the present disclosure, an outdoor unit of an air-conditioning apparatus including the heat exchanger, and an air-conditioning apparatus including the outdoor unit of the air-conditioning apparatus, the heat exchanger satisfies  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}} = (5.94635 \times 10^{-4} \times A^{-1.75030})/(8.4303H + 0.8779) > 1$ . This makes it possible to, when the heat exchanger functions as a condenser, suppress the occurrence of liquid retention in which when refrigerant flows as upward flows through the flat tubes, refrigerant liquefied by the effect of gravity is undesirably retained without being able to go upward, making it possible to reduce deterioration of defrosting performance.

#### Brief Description of Drawings

**[0011]**

[Fig. 1] Fig. 1 is a refrigerant circuit diagram of an air-conditioning apparatus including a heat exchanger according to Embodiment 1.

[Fig. 2] Fig. 2 is a perspective view of the heat exchanger according to Embodiment 1.

[Fig. 3] Fig. 3 is a front view of the heat exchanger according to Embodiment 1.

[Fig. 4] Fig. 4 is a diagram showing the flow passage cross-sectional area of a flat tube of the heat exchanger according to Embodiment 1.

[Fig. 5] Fig. 5 is a diagram showing a relationship between the total flow passage cross-sectional area of a heat exchanger core of a heat exchanger and  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$  according to experimental results.

[Fig. 6] Fig. 6 is a diagram showing a relationship between the height H of a heat exchanger core of a heat exchanger and  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$  according to the experimental results.

[Fig. 7] Fig. 7 is a diagram explaining deterioration of defrosting performance due to liquid retention in a heat exchanger.

[Fig. 8] Fig. 8 is a diagram explaining the heating capacity of the heat exchanger according to Embodiment 1 with passage of time.

[Fig. 9] Fig. 9 is a diagram explaining the heating capacity of a conventional heat exchanger with passage of time.

[Fig. 10] Fig. 10 is a diagram showing a relationship between H/L of a heat exchanger and a liquid head  $\Delta P_{\text{HEAD}}$  according to the experimental results.

[Fig. 11] Fig. 11 is a diagram showing a pressure distribution in the heat exchanger according to Embodiment 1.

[Fig. 12] Fig. 12 is a diagram showing a pressure distribution in a modification of the heat exchanger according to Embodiment 1.

[Fig. 13] Fig. 13 is a schematic view of a header flow passage of the heat exchanger according to Embodiment 1 and an area around the header flow passage.

[Fig. 14] Fig. 14 is a diagram explaining the heat exchanger performance of a heat exchanger according to Embodiment 2.

[Fig. 15] Fig. 15 is a diagram explaining the heat exchanger performance of a modification of the heat exchanger according to Embodiment 2.

[Fig. 16] Fig. 16 is a perspective view schematically showing a heat exchanger according to Embodiment 3.

[Fig. 17] Fig. 17 is an enlarged view of a bridging header of the heat exchanger according to Embodiment 3 and an area around the bridging header.

[Fig. 18] Fig. 18 is a diagram showing a relationship between the gap  $\delta$  of a heat exchanger and a differential pressure  $\Delta P_{2-3}$  according to experimental results.

[Fig. 19] Fig. 19 is a side schematic view of the heat exchanger according to Embodiment 3.

[Fig. 20] Fig. 20 is an enlarged refrigerant circuit diagram of an outdoor unit of an air-conditioning apparatus according to Embodiment 4.

[Fig. 21] Fig. 21 is an enlarged refrigerant circuit diagram of an outdoor unit of an air-conditioning apparatus according to Embodiment 5.

[Fig. 22] Fig. 22 is a cross-sectional schematic view of a flat tube of a heat exchanger according to Embodiment 6.

[Fig. 23] Fig. 23 is a cross-sectional schematic view of a flat tube of a modification of the heat exchanger according to Embodiment 6.

[Fig. 24] Fig. 24 is a side schematic view of the flat tube of the modification of the heat exchanger according to Embodiment 6.

[Fig. 25] Fig. 25 is a diagram showing a relationship between types of refrigerant that are used in a refrigerant circuit

of an air-conditioning apparatus according to Embodiment 7 and  $\Delta P_{\text{HEX}}/\Delta P_{\text{PHAD}}$ .

[Fig. 26] Fig. 26 is a front view of a heat exchanger according to Embodiment 8 of an air-conditioning apparatus.

[Fig. 27] Fig. 27 is a front view of a heat exchanger according to Embodiment 9 of an air-conditioning apparatus.

## Description of Embodiments

**[0012]** The following describes embodiments of the present disclosure with reference to the drawings. It should be noted that the present disclosure is not limited by the embodiments described below. Further, relationships in size between one constituent element and another in the following drawings may be different from actual ones.

### Embodiment 1.

#### <Configuration of Air-conditioning Apparatus 100>

**[0013]** Fig. 1 is a refrigerant circuit diagram of an air-conditioning apparatus 100 including a heat exchanger 30 according to Embodiment 1. In Fig. 1, the solid arrows indicate the flow of refrigerant during a cooling operation, and the dashed arrows indicate the flow of refrigerant during a heating operation.

**[0014]** As shown in Fig. 1, the heat exchanger 30 according to Embodiment 1 is mounted in an outdoor unit 10 of the air-conditioning apparatus 100, which includes the outdoor unit 10 and an indoor unit 20. The outdoor unit 10 includes a compressor 11, a flow switching device 12, and a fan 13 in addition to the heat exchanger 30. The indoor unit 20 includes an expansion device 21, an indoor heat exchanger 22, and an indoor fan 23.

**[0015]** Further, the air-conditioning apparatus 100 includes a refrigerant circuit 101 that is constituted by the outdoor unit 10 and the indoor unit 20 and through which refrigerant circulates. Specifically, the refrigerant circuit 101 is constituted by the compressor 11, the flow switching device 12, the heat exchanger 30, the expansion device 21, and the indoor heat exchanger 22 being connected by refrigerant pipes. The air-conditioning apparatus 100 is enabled by the flow switching device 12 to switch between a cooling operation and a heating operation.

**[0016]** The compressor 11 suctions low-temperature and low-pressure refrigerant, compresses the refrigerant thus suctioned, and discharges high-temperature and high-pressure refrigerant. The compressor 11 is constituted, for example, by an inverter compressor or other machines whose capacity, that is, rate of delivery per unit time, is controlled by varying operating frequency.

**[0017]** The flow switching device 12 is for example a four-way valve, and enables switching between a cooling operation and a heating operation to be done by switching the direction of flow of refrigerant. During a cooling operation, the flow switching device 12 makes switching to a state indicated by the solid lines in Fig. 1, so that a discharge side of the compressor 11 becomes connected to the heat exchanger 30. Further, during a heating operation, the flow switching device 12 makes switching to a state indicated by the dashed lines in Fig. 1, so that the discharge side of the compressor 11 becomes connected to the indoor heat exchanger 22.

**[0018]** The heat exchanger 30 exchanges heat between outdoor air and refrigerant. During a cooling operation, the heat exchanger 30 functions as a condenser configured to condense the refrigerant by rejecting the heat of the refrigerant to the outdoor air. Further, during a heating operation, the heat exchanger 30 functions as an evaporator configured to evaporate the refrigerant and cool the outdoor air with the resulting heat of vaporization.

**[0019]** The fan 13 is configured to supply outdoor air to the heat exchanger 30, and by controlling the rotation speed of the fan 13, the amount of air that is sent to the heat exchanger 30 is adjusted.

**[0020]** The expansion device 21 is for example an electronic expansion valve whose opening degree of expansion can be adjusted, and by adjusting the opening degree, the pressure of refrigerant flowing into the heat exchanger 30 or the indoor heat exchanger 22 is controlled. Although, in Embodiment 1, the expansion device 21 is provided in the indoor unit 20, the expansion device 21 may be provided in the outdoor unit 10 and is not limited in place of installation.

**[0021]** The indoor heat exchanger 22 exchanges heat between indoor air and refrigerant. During a cooling operation, the indoor heat exchanger 22 functions as an evaporator configured to evaporate the refrigerant and cool the outdoor air with the resulting heat of vaporization. Further, during a heating operation, the indoor heat exchanger 22 functions as a condenser configured to condense the refrigerant by rejecting the heat of the refrigerant to the outdoor air.

**[0022]** The indoor fan 23 is configured to supply indoor air to the indoor heat exchanger 22, and by controlling the rotation speed of the indoor fan 23, the amount of air that is sent to the indoor heat exchanger 22 is adjusted.

#### <Configuration of Heat Exchanger 30>

**[0023]** Fig. 2 is a perspective view of the heat exchanger 30 according to Embodiment 1. Fig. 3 is a front view of the heat exchanger 30 according to Embodiment 1. It should be noted that the dashed arrows in Fig. 2 and the white arrows in Fig. 3 indicate the flow of refrigerant during a cooling operation. Further, Fig. 3 shows the height H and width L of the

after-mentioned heat exchanger core 31.

**[0024]** As shown in Fig. 2, the heat exchanger 30 includes a heat exchanger core 31 including a plurality of flat tubes 38 and a plurality of fins 39. The flat tubes 38 are placed in parallel with one another at spacings in a horizontal direction (i.e. a right-left direction of Fig. 2) so that wind generated by the fan 13 flows, and refrigerant flows in a vertical direction through the flat tubes 38, which extend in a vertical direction (i.e. an up-down direction of Fig. 2). The fins 39 are each connected across a space between adjacent ones of the flat tubes 38 and transfer heat to the flat tubes 38. The fins 39 are intended to improve the efficiency of heat exchange between air and refrigerant. Usable examples of the fins 39 include, but are not limited to, corrugated fins. The fins 39 are dispensable because heat exchange between air and refrigerant takes place on the surfaces of the flat tubes 38.

**[0025]** A first header 34 is provided at a lower end of the heat exchanger core 31. Lower ends of the flat tubes 38 of the heat exchanger core 31 are directly inserted in the first header 34. Further, a second header 35 is provided at an upper end of the heat exchanger core 31. Upper ends of the flat tubes 38 of the heat exchanger core 31 are directly inserted in the second header 35.

**[0026]** A hot-gas refrigerant inlet 32 is formed at one end of the first header 34, and the hot-gas refrigerant inlet 32 is connected to the refrigerant circuit 101 of the air-conditioning apparatus 100 via a gas pipe 37. Therefore, the first header 34 is also called "gas header". The first header 34 causes high-temperature and high-pressure gas refrigerant (hereinafter also referred to as "hot-gas refrigerant") from the compressor 11 to flow into the heat exchanger 30 during a cooling operation and causes low-temperature and low-pressure gas refrigerant subjected to heat exchange in the heat exchanger 30 to flow out to the refrigerant circuit 101 during a heating operation. That is, the hot-gas refrigerant inlet 32 serves as a hot-gas refrigerant inflow port. Note here that the hot-gas refrigerant is not limited to single-phase gas refrigerant but may be two-phase gas-liquid refrigerant containing a gas phase of 0 degrees C or higher.

**[0027]** A liquid refrigerant outlet 33 is formed at one end of the second header 35, and the liquid refrigerant outlet 33 is connected to the refrigerant circuit 101 of the air-conditioning apparatus 100 via a liquid pipe 36. Therefore, the second header 35 is also called "liquid header". The second header 35 causes low-temperature and low-pressure two-phase refrigerant to flow into the heat exchanger 30 during a heating operation and causes low-temperature and high-pressure liquid refrigerant subjected to heat exchange in the heat exchanger 30 to flow out to the refrigerant circuit 101 during a cooling operation.

**[0028]** The plurality of flat tubes 38, the plurality of fins 39, the first header 34, and the second header 35 are all made of aluminum and joined to each other by brazing.

**[0029]** Next, the actions of the air-conditioning apparatus 100 during each operation are described with reference to Figs. 1 and 2.

#### <Cooling Operation>

**[0030]** High-temperature and high-pressure gas refrigerant discharged from the compressor 11 flows into the heat exchanger 30 via the flow switching device 12. The high-temperature and high-pressure gas refrigerant having flowed into the heat exchanger 30 condenses while rejecting heat by exchanging heat with outdoor air taken in by the fan 13 and turns into low-temperature and high-pressure liquid refrigerant that then flows out from the heat exchanger 30. The low-temperature and high-pressure liquid refrigerant having flowed out from the heat exchanger 30 is decompressed by the expansion device 21 into low-temperature and low-pressure two-phase gas-liquid refrigerant that then flows into the indoor heat exchanger 22. The low-temperature and low-pressure two-phase gas-liquid refrigerant having flowed into the indoor heat exchanger 22 evaporates while receiving heat by exchanging heat with indoor air taken in by the indoor fan 23, cools the indoor air, and turns into low-temperature and low-pressure gas refrigerant that then flows out from the indoor heat exchanger 22. The low-temperature and low-pressure gas refrigerant having flowed out from the indoor heat exchanger 22 is suctioned into the compressor 11 and turns again into high-temperature and high-pressure gas refrigerant.

#### <Heating Operation>

**[0031]** High-temperature and high-pressure gas refrigerant discharged from the compressor 11 flows into the indoor heat exchanger 22 via the flow switching device 12. The high-temperature and high-pressure gas refrigerant having flowed into the indoor heat exchanger 22 condenses while rejecting heat by exchanging heat with indoor air taken in by the indoor fan 23, heats the indoor air, and turns into low-temperature and high-pressure liquid refrigerant that then flows out from the indoor heat exchanger 22. The low-temperature and high-pressure liquid refrigerant having flowed out from the indoor heat exchanger 22 is decompressed by the expansion device 21 and turns into low-temperature and low-pressure two-phase gas-liquid refrigerant that then flows into the heat exchanger 30. The low-temperature and low-pressure two-phase gas-liquid refrigerant having flowed into the heat exchanger 30 evaporates while receiving heat by exchanging heat with outdoor air taken on by the fan 13 and turns into low-temperature and low-pressure gas refrigerant

that then flows out from the heat exchanger 30. The low-temperature and low-pressure gas refrigerant having flowed out from the heat exchanger 30 is suctioned into the compressor 11 and turns again into high-temperature and high-pressure gas refrigerant.

#### <Defrosting Operation>

**[0032]** In a case in which a heating operation is performed in a low-temperature environment in which the surface temperature of the flat tubes 38 and the fins 39 shown in Fig. 2 is lower than or equal to 0 degrees C, frost forms on the heat exchanger 30. When a particular or larger amount of frost forms on the heat exchanger 30, an air trunk of the heat exchanger 30 through which wind generated by the fan 13 passes is clogged. This causes deterioration of performance of the heat exchanger 30, resulting in deterioration of heating performance. Accordingly, in the case of deterioration of heating performance, a defrosting operation is performed to melt the frost on the surface of the heat exchanger 30.

**[0033]** In the defrosting operation, the fan 13 is suspended, and the flow switching device 12 is switched to the same state as it is during a cooling operation, so that high-temperature and high-pressure gas refrigerant flows into the heat exchanger 30. This causes the frost forming on the flat tubes 38 and the fins 39 to melt. Once the defrosting operation is started, the high-temperature and high-pressure gas refrigerant flows from the gas pipe 37 into each flat tube 38 via the first header 34. It should be noted that the refrigerant having flowed into each flat tube 38 turns into an upward flow that flows upward in a vertical direction. Then, the high-temperature refrigerant having flowed into the flat tubes 38 causes the frost forming on the flat tubes 38 and the fins 39 to melt and turn into water. The water produced by the frost melting is drained toward a lower level in the heat exchanger 30 along the flat tubes 38 or the fins 39. The defrosting operation is ended once the frost melts, and the heating operation is resumed. It should be noted that it is possible to use a known method to determine when to end the defrosting operation and resume the heating operation. For example, such a configuration may be set up to end the defrosting operation and resume the heating operation, for example, when a temperature detected by a temperature sensor (not illustrated) has reached a predetermined temperature or in a case in which the defrosting operation has been executed for a particular period of time.

**[0034]** Fig. 4 is a diagram showing the flow passage cross-sectional area of a flat tube 38 of the heat exchanger 30 according to Embodiment 1. Fig. 5 is a diagram showing a relationship between the total flow passage cross-sectional area of a heat exchanger core 31 of a heat exchanger and  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$  according to experimental results. Fig. 6 is a diagram showing a relationship between the height H of a heat exchanger core 31 of a heat exchanger and  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$  according to the experimental results. Fig. 7 is a diagram explaining deterioration of defrosting performance due to liquid retention in a heat exchanger. In Fig. 7, the white arrows indicate the flow of refrigerant during a defrosting operation.

**[0035]** In a case in which the total flow passage cross-sectional area of the heat exchanger core 31 is defined as A, the total flow passage cross-sectional area A is found by Formula (1) as follows:

$$A = a \times N \text{ [m}^2\text{]} \quad (1)$$

where a is the flow passage cross-sectional area [m<sup>2</sup>] of each of the flat tubes 38 (i.e. a shaded area of Fig. 4) and N is the number of flat tubes 38.

**[0036]** Further, in a case in which a differential pressure of a refrigerant flow passage (hereinafter referred to as "flow passage differential pressure") is defined as  $\Delta P_{\text{HEX}}$  and a liquid head is defined as  $\Delta P_{\text{HEAD}}$ ,  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$  is found by Formula (2) below. In Embodiment 1, the flow passage differential pressure  $\Delta P_{\text{HEX}}$  is the differential pressure of a flow passage through which hot-gas refrigerant flows as an upward flow during a defrosting operation and is the differential pressure across the upper and lower ends of a flat tube 38 in the heat exchanger core 31.

$$\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}} = (5.94635 \times 10^{-4} \times A^{-1.75030})/(8.4303H + 0.8779) \quad (2)$$

where A is the total flow passage cross-sectional area [m<sup>2</sup>] of the heat exchanger core 31 and H is the height [m] of the heat exchanger core 31.

**[0037]** Note here that the height H of the heat exchanger core 31 is a length between an upper end of the first header 34 and a lower end of the second header 35 and is the length of an exposed portion of a flat tube 38.

**[0038]** Formula (2) above is an empirical formula obtained by the inventors' numerical analyses and experimental results and, in a range of conditions under which the heat exchanger 30 is used in an outdoor unit 10 for building use, store use, and household use (hereinafter referred to as "use in buildings or other structures"), formulated using the total flow passage cross-sectional area A [m<sup>2</sup>] of the heat exchanger core 31, which is a shape parameter of the heat exchanger

30 dominated by the flow passage differential pressure  $\Delta P_{\text{HEX}}$ , and the height  $H$  [m] of the heat exchanger core 31, which is a shape parameter of the heat exchanger 30 dominated by the liquid head  $\Delta P_{\text{HEAD}}$ . This empirical formula is shown in Figs. 5 and 6. Fig. 5 is one obtained by making changes to the total flow passage cross-sectional area  $A$  of the heat exchanger core 31 with the height  $H$  of the heat exchanger core 31 fixed, and Fig. 6 is one obtained by making changes to the height  $H$  of the heat exchanger core 31 with the total flow passage cross-sectional area  $A$  of the heat exchanger core 31 fixed.

[0039] According to the inventors' experiments, as shown in Fig. 5,  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$  tends to decrease as the total flow passage cross-sectional area  $A$  [m<sup>2</sup>] of the heat exchanger core 31 increases. Further, as shown in Fig. 6,  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$  tends to decrease as the height  $H$  [m] of the heat exchanger core 31 increases. Moreover, according to the experimental results, it was found, as shown in Figs. 5 and 6, that when  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}} \leq 1$ , there is remarkable deterioration of defrosting performance due to the occurrence of liquid retention in which when hot-gas refrigerant having flowed into the first header 34 flows as upward flows through the flat tubes 38 of the heat exchanger core 31, refrigerant liquefied by the effect of gravity is undesirably retained in part of a hot-gas flow area without being able to go upward. As shown in Fig. 7, the occurrence of liquid retention undesirably forms, in a region in which the liquid retention has occurred (hereinafter referred to as "liquid retention area"), a lingering frost region in which frost forming on the flat tubes 38 and the fins 39 remains without melting even when a defrosting operation is performed.

[0040] In general, car air conditioners or other air conditioners utilize engine heat during heating and use heat pumps only during cooling. Therefore, most heat exchangers that are used in outdoor units of car air conditioners or other air conditioners and that use corrugated fins are heat exchangers dedicated to cooling and are therefore used for such purposes that a defrosting operation does not take place. Further, even in a case in which heat exchangers are used as heat pumps both during cooling and during heating, most of the heat exchangers are so small in size that the heights of heat exchanger cores are approximately 300 [mm]. On the other hand, in most heat exchangers that are used in outdoor units in buildings or other structures, the heights of heat exchanger cores are greater than or equal to 420 [mm], and in some of the heat exchangers, the heights of heat exchanger cores are greater than or equal to 800 [mm].

[0041] According to the inventors' study, it was found, as shown in Fig. 6, that attempting to apply, to outdoor units for use in buildings or other structures, heat exchangers that are used in outdoor units of car air conditioners or other air conditioners, for example, increasing the height  $H$  of a heat exchanger core 31 to approximately 420 [mm], causes  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$  to decrease by 43 [%] in comparison with a heat exchanger core 31 having a height of 300 [mm]. Moreover, it was found that this decrease leads to the occurrence of liquid retention in which it becomes hard for liquid refrigerant to flow through part of the heat exchanger. Further, according to the inventors' study, it was found that a heat exchanger core 31 having a height  $H$  of 490 [mm] is lower in  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$  by approximately 50 [%] than a heat exchanger core 31 having a height  $H$  of 300 [mm] and that a heat exchanger core 31 having a height  $H$  of 800 [mm] is lower in  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$  by approximately 65 [%] than a heat exchanger core 31 having a height  $H$  of 300 [mm].

[0042] Under conditions where  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$  is lower than or equal to 1, the liquid retention area increases as  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$  decreases. This suggests that a decrease in  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$  by 43 [%] to 65 [%] in comparison with a heat exchanger whose heat exchanger core 31 has a height  $H$  of 300 [mm] has an extraordinary effect on heat exchanger performance. According to the inventors' experiments, there were examples in which when  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$  became lower than or equal to 1, there were decreases in heat exchanger performance by approximately 30 [%] to 50 [%] in comparison with a heat exchanger whose heat exchanger core 31 has a height  $H$  of 300 [mm].

[0043] Conventionally, in a case in which the height  $H$  of a heat exchanger core 31 is greater than or equal to 420 [mm], there is undesirably remarkable deterioration of defrosting performance in a region of occurrence of liquid retention in which when hot-gas refrigerant flows into a first header 34 provided in a lower part of the heat exchanger and flows via the first header 34 and refrigerant flows as upward flows through flat tubes 38 extending in a vertical direction, refrigerant liquefied by the effect of gravity is retained without being able to go upward. To address this problem, Embodiment 1 is intended to remedy this problem to reduce deterioration of defrosting performance during a defrosting operation.

[0044] According to the experimental results, it was found, as shown in Figs. 5 and 6, that the occurrence of liquid retention can be suppressed when  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}} > 1$ . Accordingly, in Embodiment 1, the heat exchanger 30 is configured to satisfy  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}} > 1$  to suppress the occurrence of liquid retention to reduce deterioration of defrosting performance during a defrosting operation.

[0045] Fig. 8 is a diagram explaining the heating capacity of the heat exchanger 30 according to Embodiment 1 with passage of time. Fig. 9 is a diagram explaining the heating capacity of a conventional heat exchanger with passage of time.

[0046] In a case in which a heat exchanger is not configured to satisfy  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}} > 1$  as has conventionally been the case, liquid retention occurs during a defrosting operation, and in a liquid retention area in which the liquid retention has occurred, frost undesirably remains without being sufficiently removed. Therefore, as shown in Fig. 9, heating capacity during a heating operation deteriorates with passage of time.

[0047] On the other hand, in a case in which a heat exchanger 30 is configured to satisfy  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}} > 1$  as is the case in Embodiment 1, the occurrence of liquid retention during a defrosting operation is suppressed, so that lingering

frost can be reduced. Therefore, as shown in Fig. 8, deterioration of heating capacity during a heating operation can be reduced even with passage of time, so that the heating capacity during the heating operation can be improved.

[0048] Further, as shown in Fig. 3, the heat exchanger 30 is configured to satisfy  $H/L > 1$  in a case in which the width of the heat exchanger core 31 is defined as  $L$  [m]. Note here that the width of the heat exchanger core 31 is the distance between the outer side parts of two of the plurality of flat tubes 38, which are arranged in a horizontal direction, that are placed furthest outward on both sides.

[0049] Fig. 10 is a diagram showing a relationship between  $H/L$  of a heat exchanger and  $\Delta P_{HEAD}$  according to the experimental results. It should be noted that Fig. 10 shows a relationship between  $H/L$  and the liquid head  $\Delta P_{HEAD}$  in a case in which a working fluid was passed through the heat exchanger.

[0050] In a heat exchanger, when the height  $H$  of a heat exchanger core is high and the aspect ratio of the heat exchanger increases in a height direction, that is, when  $H/L$  increases, the liquid head  $\Delta P_{HEAD}$  increases as shown in Fig. 10. Moreover, when the liquid head  $\Delta P_{HEAD}$  increases,  $\Delta P_{HEX}/\Delta P_{HEAD}$  decreases, whereby liquid retention occurs and there is undesirably remarkable deterioration of defrosting performance. However, even in such a high heat exchanger that is used in an outdoor unit in a building or other structures and in which  $H/L > 1$ , the occurrence of liquid retention can be suppressed, provided  $\Delta P_{HEX}/\Delta P_{HEAD} > 1$  is satisfied.

[0051] Fig. 11 is a diagram showing a pressure distribution in the heat exchanger 30 according to Embodiment 1. Fig. 12 is a diagram showing a pressure distribution in a modification of the heat exchanger 30 according to Embodiment 1. In Figs. 11 and 12, the white arrows and the black arrows indicate the flow of refrigerant during a defrosting operation.

[0052] In Embodiment 1, as shown in Fig. 11, the hot-gas refrigerant inlet 32 and the liquid refrigerant outlet 33 are provided at ends of the first header 34 and the second header 35 located on the same side as each other, respectively. On the other hand, in the modification according to Embodiment 1, the hot-gas refrigerant inlet 32 and the liquid refrigerant outlet 33 are provided at ends of the first header 34 and the second header 35 located opposite each other, respectively.

[0053] In a case in which the hot-gas refrigerant inlet 32 and the liquid refrigerant outlet 33 are provided at ends of the first header 34 and the second header 35 located on the same side as each other, respectively, as shown in Fig. 11,  $G_{1-4} > G_{2-3}$ , where  $G_{1-4}$  is the flow rate of refrigerant that flows from position (1) to position (4) in Fig. 11 and  $G_{2-3}$  is the flow rate of refrigerant that flows from position (2) to position (3) in Fig. 11. A reason for this is that under the effect of the pressure loss  $\Delta P_{1-2}$  of the first header 34 and the pressure loss  $\Delta P_{3-4}$  of the second header 35 due to the difference therebetween, the differential pressure  $\Delta P_{1-4}$  between position (1) and position (4) in Fig. 11 becomes relatively greater than the differential pressure  $\Delta P_{2-3}$  between position (2) and position (3) in Fig. 11. In other words, the flow passage differential pressure  $\Delta P_{HEX}$  decreases with distance from the hot-gas refrigerant inlet 32 and the liquid refrigerant outlet 33. Therefore, in a location away from the hot-gas refrigerant inlet 32 and the liquid refrigerant outlet 33, a region in which the flow passage differential pressure  $\Delta P_{HEX}$  decreases tends to form, and in the region,  $\Delta P_{HEX}/\Delta P_{HEAD}$  decreases, whereby liquid retention tends to occur.

[0054] On the other hand, when the hot-gas refrigerant inlet 32 and the liquid refrigerant outlet 33 are provided at ends of the first header 34 and the second header 35 located opposite each other, respectively, as shown in Fig. 12, the difference between the pressure loss  $\Delta P_{1-2}$  of the first header 34 and the pressure loss  $\Delta P_{3-4}$  of the second header 35 decreases. Therefore, under the diminishing effect of the pressure loss  $\Delta P_{1-2}$  of the first header 34 and the pressure loss  $\Delta P_{3-4}$  of the second header 35, the difference between the flow rate  $G_{1-4}$  of refrigerant that flows from position (1) to position (4) in Fig. 12 and the flow rate  $G_{2-3}$  of refrigerant that flows from position (2) to position (3) in Fig. 12 decreases. Accordingly, a region in which the flow passage differential pressure  $\Delta P_{HEX}$  decreases hardly forms. This suppresses the occurrence of liquid retention, making it possible to reduce lingering frost.

[0055] Fig. 13 is a schematic view of a header flow passage of the heat exchanger 30 according to Embodiment 1 and an area around the header flow passage. Although Fig. 13 shows the header flow passage of the first header 34 and the area around the header flow passage, the same configuration applies to a header flow passage of the second header 35 and an area around the header flow passage.

[0056] As noted above, by configuring the heat exchanger 30 to satisfy  $H/L > 1$ , the first header 34 and the second header 35 can be configured to be small in length with respect to the amount of heat exchange. This makes it possible to reduce pressure losses of working fluids that flow through the first header 34 and the second header 35. That is, the difference between the pressure loss  $\Delta P_{1-2}$  of the first header 34 and the pressure loss  $\Delta P_{3-4}$  of the second header 35 described in Figs. 11 and 12 can be decreased, and  $\Delta P_{2-3}$  can be decreased accordingly. This makes it possible to suppress the occurrence of liquid retention. In particular, in a heat exchanger 38 using flat tubes 38, the first and second headers 34 and 35 and the flat tubes 38 are usually joined to each other by brazing; therefore, as shown in Fig. 13, the flat tubes 38 are inserted into header flow passages formed inside the first header 34 and inside the second header 35. In this case, a phenomenon in which a working fluid expands and contracts at ends of the flat tubes 38 projecting into a header flow passage occurs in addition to a common friction loss of the header flow passage. This causes a big increase in pressure loss. According to the inventors' experiments, it became evident that there is a case in which a pressure loss due to expansion or contraction of a working fluid rather than a fluid resistance of friction occupies approximately 50% or more, and this effect becomes remarkable with an increase in the number of flat tubes 38 that are

inserted. In such a case, pressure losses in the first header 34 and the second header 35 can be reduced by configuring the heat exchanger 30 to satisfy  $H/L > 1$ .

[0057] As noted above, the heat exchanger 30 according to Embodiment 1 is a heat exchanger 30 that is mounted in an outdoor unit 10 of an air-conditioning apparatus 100. The heat exchanger 30 includes one heat exchanger core or two or more heat exchanger cores 31 each including a plurality of flat tubes 38 that extend in an up-down direction and through which refrigerant flows as upward flows when the heat exchanger 30 functions as a condenser. The two or more heat exchanger cores 31 are placed along a direction of flow of air. In a case in which a total flow passage cross-sectional area of each of the heat exchanger cores 31 is defined as  $A \text{ [m}^2\text{]} = a \times N \text{ [m}^2\text{]}$ , where  $a \text{ [m}^2\text{]}$  is a flow passage cross-sectional area of each of the flat tubes 38 and  $N$  is a number of the flat tubes 38, a height of each of the heat exchanger cores 31 is defined as  $H \text{ [m]}$ , a differential pressure of a refrigerant flow passage is defined as  $\Delta P_{\text{HEX}}$ , and a liquid head is defined as  $\Delta P_{\text{HEAD}}$ ,  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}} = (5.94635 \times 10^{-4} \times A^{-1.75030})/(8.4303H + 0.8779) > 1$  is satisfied.

[0058] According to the heat exchanger 30 according to Embodiment 1, the heat exchanger 30 satisfies  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}} = (5.94635 \times 10^{-4} \times A^{-1.75030})/(8.4303H + 0.8779) > 1$ . This makes it possible to, when the heat exchanger 30 functions as a condenser, suppress the occurrence of liquid retention in which when refrigerant flows as upward flows through the flat tubes 38, refrigerant liquefied by the effect of gravity is retained without being able to go upward, making it possible to reduce deterioration of defrosting performance.

[0059] Further, the heat exchanger 30 according to Embodiment 1 is configured to satisfy  $H/L > 1$  in a case in which a width of each of the heat exchanger cores 31 is defined as  $L \text{ [m]}$ .

[0060] The heat exchanger 30 according to Embodiment 1 is configured to satisfy  $H/L > 1$ . Therefore, the first header 34 and the second header 35 can be configured to be small in length with respect to the amount of heat exchange. This makes it possible to reduce pressure losses of working fluids that flow through the first header 34 and the second header 35, making it possible to suppress the occurrence of liquid retention.

[0061] Further, the heat exchanger 30 according to Embodiment 1 includes the one heat exchanger core 31, a first header 34 provided at a lower end of the heat exchanger core 31, and a second header 35 provided at an upper end of the heat exchanger core 35. Moreover, the heat exchanger 30 according to Embodiment 1 includes a hot-gas refrigerant inlet 32 formed at one end of the first header 34 and a liquid refrigerant outlet 33 that is formed at one end of the second header 35 located opposite the one end of the first header 34 and through which refrigerant flows out when the heat exchanger 30 functions as a condenser.

[0062] According to the heat exchanger 30 according to Embodiment 1, the hot-gas refrigerant inlet 32 and the liquid refrigerant outlet 33 are formed at ends of the first header 34 and the second header 35 located opposite each other. Therefore, the difference between the difference between the pressure loss  $\Delta P_{1-2}$  of the first header 34 and the pressure loss  $\Delta P_{3-4}$  of the second header 35 decreases. As a result of that, a region in which the flow passage differential pressure  $\Delta P_{\text{HEX}}$  decreases hardly forms. This suppresses the occurrence of liquid retention, making it possible to reduce lingering frost.

[0063] Further, an outdoor unit 10 according to Embodiment 1 of an air-conditioning apparatus 100 includes the heat exchanger 30.

[0064] Further, an air-conditioning apparatus 100 according to Embodiment 1 includes the outdoor unit 10 of the air-conditioning apparatus 100, an indoor unit 20 of the air-conditioning apparatus 100, and a refrigerant circuit 101 that is constituted by the outdoor unit 10 of the air-conditioning apparatus 100 and the indoor unit 20 of the air-conditioning apparatus 100 and through which refrigerant circulates.

[0065] The outdoor unit 10 according to Embodiment 1 of an air-conditioning apparatus 100 and the air-conditioning apparatus 100 according to Embodiment 1 bring about effects that are similar to those of the heat exchanger 30.

Embodiment 2.

[0066] The following describes Embodiment 2. A description of overlaps with Embodiment 1 is omitted. Components that are identical or equivalent to those of Embodiment 1 are given identical reference signs.

[0067] Fig. 14 is a diagram explaining the heat exchanger performance of a heat exchanger 30 according to Embodiment 2. Fig. 15 is a diagram explaining the heat exchanger performance of a modification of the heat exchanger 30 according to Embodiment 2. In Figs. 14 and 15, the white arrows and the black arrows indicate the flow of refrigerant during a defrosting operation. Further, Figs. 14 and 15 show the widths of different regions of a heat exchanger core 31 in the order of  $L_1, L_2, \dots$  from a downstream side. The term "downstream" here refers to the direction in which refrigerant having flowed in through the hot-gas refrigerant inlet 32 flows. The same applies below too.

[0068] In the heat exchanger 30 according to Embodiment 2, a partition plate 40 is provided in at least the first header 34. Fig. 14 shows a case in which one partition plate 40 is provided in the first header 34 and in which an odd number of partition plates 40 are provided. Further, Fig. 15 shows a case in which one partition plate 40 is provided in the first header 34, in which one partition plate 40 is provided in the second header 35, and in which an even number of partition plates 40 are provided. As shown in Fig. 14, in a case in which an odd number of partition plates 40 are provided, the

liquid refrigerant outlet 33 is provided at an end of the first header 34 opposite an end of the first header 34 at which the hot-gas refrigerant inlet 32 is formed. Further, as shown in Fig. 15, in a case in which an even number of partition plates 40 are provided, the liquid refrigerant outlet 33 is provided at an end of the second header 35 located opposite the end of the first header 34 at which the hot-gas refrigerant inlet 32 is formed.

**[0069]** This partition plate 40 is provided to divide a flow passage of the heat exchanger core 31 into a plurality of regions in a horizontal direction. Further, the partition plate 40 is provided so that a flow passage in each region of the heat exchanger core 31 is opposite in flow to a flow passage in an adjacent region. Moreover, assuming that  $L_1$  is the width of the furthest downstream region of the heat exchanger core 31, the heat exchanger 30 is configured to satisfy  $20 [\%] \leq L_1/L \leq 50 [\%]$ .

**[0070]** By providing the partition plate 40 in the first header 34, the flow passage cross-sectional area is reduced with the flow rate of refrigerant being the same. This increases the velocity of flow of refrigerant but causes an increase in pressure loss. Accordingly, in consideration of a balance between improvement in heat exchanger performance brought about by improvement in heat-transfer coefficient due to the increase in the velocity of flow of refrigerant and deterioration of heat exchanger performance caused by the increase in pressure loss, the heat exchanger 30 is configured to satisfy  $20 [\%] \leq L_1/L \leq 50 [\%]$  to attain a heat exchanger performance of 90 [%] or higher as shown in Figs. 14 and 15. Doing so makes it possible to make heat exchanger performance higher than in a case in which no partition plate 40 is provided in the first header 34. Furthermore, the increase in pressure loss suppresses the occurrence of liquid retention during a defrosting operation, making it possible to reduce lingering frost. As a result of that, defrosting performance during a defrosting operation can be improved. According to the inventors' experiments, it is found, as shown in Figs. 14 and 15, that a maximum of approximately 10 [%] improvement in heat exchanger performance can be brought about by configuring the heat exchanger 30 to satisfy  $20 [\%] \leq L_1/L \leq 50 [\%]$ .

**[0071]** As noted above, the heat exchanger 30 according to Embodiment 2 is a heat exchanger 30 including the one heat exchanger core 31, a first header 34 provided at a lower end of the heat exchanger core 31, and a second header 35 provided at an upper end of the heat exchanger core 35. Further, the heat exchanger 30 includes a partition plate 40 provided in at least the first header 34 and configured to divide a flow passage of the heat exchanger core 31 into a plurality of regions in a width direction. Moreover, in a case in which the width of the heat exchanger core 31 is defined as  $L$  [m] and a width of the furthest downstream region of the heat exchanger core 31 is defined as  $L_1$ , the heat exchanger 30 satisfies  $20 [\%] \leq L_1/L \leq 50 [\%]$ .

**[0072]** The heat exchanger 30 according to Embodiment 2 is configured to satisfy  $20 [\%] \leq L_1/L \leq 50 [\%]$ . This makes it possible to make heat exchanger performance higher than in a case in which no partition plate 40 is provided in the first header 34. Furthermore, an increase in pressure loss suppresses the occurrence of liquid retention during a defrosting operation, making it possible to reduce lingering frost. As a result of that, defrosting performance during a defrosting operation can be improved.

### Embodiment 3.

**[0073]** The following describes Embodiment 3. A description of overlaps with Embodiments 1 and 2 is omitted. Components that are identical or equivalent to those of Embodiments 1 and 2 are given identical reference signs.

**[0074]** Fig. 16 is a perspective view schematically showing a heat exchanger 30 according to Embodiment 3. Fig. 17 is an enlarged view of a bridging header 50 of the heat exchanger 30 according to Embodiment 3 and an area around the bridging header 50. To avoid complicated illustration, Fig. 16 illustrates only flat tubes 38 at ends of heat exchanger cores 31. In Fig. 16, the black arrows indicate the flow of air that passes through the heat exchanger 30, and the dashed arrows and the white arrows indicate the flow of refrigerant during a defrosting operation. Further, in Fig. 17, the white arrows indicate the flow of refrigerant

**[0075]** As shown in Fig. 16, in the heat exchanger 30 according to Embodiment 3, two heat exchanger cores 31 are arranged side by side in the direction of flow of air. Moreover, upper ends of both of the two heat exchanger cores 31 arranged in the direction of flow of air are connected to the bridging header 50. Further, a lower end of one of the two heat exchanger cores 31 that is located on a leeward side is connected to the first header 34, and a lower end of one of the two heat exchanger cores 31 that is located on a windward side is connected to the second header 35. Moreover, during a defrosting operation, hot-gas refrigerant having flowed into the first header 34 flows as upward flows through the flat tubes 38 of the leeward heat exchanger core 31, turns back at the bridging header 50, flows as downward flows through the flat tubes 38 of the windward heat exchanger core 31, and then flows out from the second header 35. That is, the bridging header 50 is provided at one end of each of the two adjacent heat exchanger cores 31 and configured so that refrigerant converging from the flat tubes 38 of the leeward heat exchanger cores 31 is distributed to the flat tubes 38 of the windward heat exchanger cores 31.

**[0076]** By thus arranging the two heat exchanger cores 31 in the direction of flow of air and providing the bridging header 50 at one end of each of the two heat exchanger cores 31, a refrigerant flow passage represented by the sum of the heights of the two heat exchanger cores 31 can be lengthened. Therefore, the flow passage differential pressure

$\Delta P_{\text{HEX}}$  can be increased. As a result of that,  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$  can be increased, and liquid retention is suppressed. This makes it possible to improve defrosting performance during a defrosting operation. In Embodiment 3, the flow passage differential pressure  $\Delta P_{\text{HEX}}$  is the differential pressure  $P_{1-4}$  between a lower end of a flat tube 38 of the leeward heat exchanger core 31 and a lower end of a flat tube 38 of the windward heat exchanger core 31 (i.e. the differential pressure between position (1) and position (4) in Fig. 16).

**[0077]** Further, as shown in Fig. 17, assuming that  $\delta$  is the gap between an upper end of a flat tube 38 inserted in the bridging header 50 and a wall portion 51 of the bridging header 50 that faces the upper end of the flat tube 38, the upper end of the flat tube 38 is inserted in the bridging header 50 so that  $\delta \leq 3$  [mm], preferably  $\delta \leq 1$  [mm].

**[0078]** By so doing, the differential pressure  $P_{2-3}$  in the bridging header 50 (i.e. the differential pressure between position (2) and position (3) in Fig. 16) is increased, so the flow passage differential pressure  $\Delta P_{\text{HEX}} = P_{1-4}$  increases. This makes it possible to suppress the occurrence of liquid retention.

**[0079]** Figs. 18 is a diagram showing a relationship between the gap  $\delta$  of a heat exchanger and a differential pressure  $\Delta P_{2-3}$  according to experimental results. It should be noted that Fig. 18 shows an example of the differential pressure  $P_{2-3}$  in the bridging header 50 with variations in the gap  $\delta$  between the upper end of the flat tube 38 and the wall portion 51 of the bridging header 50 based on the inventors' simulations.

**[0080]** As shown in Fig. 18, reducing the gap  $\delta$  causes the differential pressure  $\Delta P_{2-3}$  to exponentially increase. Usually, increasing the gap  $\delta$  brings about improvement in heat exchanger performance, but in consideration of the occurrence of liquid retention, it is preferable that  $\delta \leq 3$  [mm], more preferably  $\delta \leq 1$  [mm]. Doing so makes it possible to suppress the occurrence of liquid retention even in consideration of the effect of a pressure loss, making it possible to improve heat exchanger performance.

**[0081]** Fig. 19 is a side schematic view of the heat exchanger 30 according to Embodiment 3. In Fig. 19, the black arrow indicates the flow of air that passes through the heat exchanger 30, and the white arrows indicate the flow of refrigerant.

**[0082]** In general, during a defrosting operation, the fan 13 stops to reduce leakage of heat from the heat exchanger 30 to the air, but due to the effect of external wind or other currents, an air current that pushes into the heat exchanger 30 may be generated. In such a case, as shown in Fig. 19, the first header 34, in which the hot-gas refrigerant inlet 32 is formed, is placed on the leeward side, and the second header 35, in which the liquid refrigerant outlet 33 is formed, is placed on the windward side. Doing so reduces the difference between the temperature of refrigerant in the flat tubes 38 of the leeward heat exchanger core 31 through which the refrigerant flows as upward flows that move upward in a vertical direction and the temperature of air, thus making it possible to suppress the occurrence of liquid retention in the leeward heat exchanger core 31. Further, as shown in Fig. 19, the flow of refrigerant through the leeward heat exchanger core 31 and the flow of refrigerant through the windward heat exchanger core 31 are opposite to each other. Therefore, even in a case in which an air current that pushes into the heat exchanger 30 is generated due to the effect of external wind or other currents, the temperature of air can be raised by the flat tubes 38 of the windward heat exchanger core 31. Moreover, this reduces the difference between the temperature of refrigerant in the flat tubes 38 of the leeward heat exchanger core 31 and the temperature of air, thus making it possible to suppress the occurrence of liquid retention in the leeward heat exchanger core 31.

**[0083]** Although the heat exchanger 30 according to Embodiment 3 is configured such that two heat exchanger cores 31 are arranged side by side in the direction of flow of air, the heat exchanger 30 according to Embodiment 3 is not limited to this configuration but may be configured such that three or more heat exchanger cores 31 are arranged side by side in the direction of flow of air. In that case, the heat exchanger 30 is configured to include bridging headers 50 whose number is smaller by 1 than the number of heat exchanger cores 31. For example, in a case in which three heat exchanger cores 31 are arranged side by side in the direction of flow of air, the first header 34 is provided at a lower end of one of the heat exchanger cores 31 that is located on the furthest leeward side, and the second header 35 is provided at an upper end of one of the heat exchanger cores 31 that is located on the furthest windward side. Moreover, a bridging header 50 is provided at an upper end of the furthest leeward heat exchanger core 31 and an upper end of a middle one of the heat exchanger cores 31 that is adjacent to the furthest leeward heat exchanger core 31, and a bridging header 50 is provided at a lower end of the middle heat exchanger core 31 and a lower end of the furthest windward heat exchanger core 31 that is adjacent to the middle heat exchanger core 31.

**[0084]** As noted above the heat exchanger 30 according to Embodiment 3 is a heat exchanger 30 including the two or more heat exchanger cores 31 placed along the direction of flow of air, a first header 34 provided at a lower end of one of the heat exchanger cores 31 that is located on a furthest leeward side, a second header 35 provided at an upper or lower end of one of the heat exchanger cores 31 that is located at a furthest windward side, a hot-gas refrigerant inlet 32 formed at one end of the first header 34, and a liquid refrigerant outlet 33 formed at one end of the second header 35 located on the same side as the one end of the first header 34. Moreover, the heat exchanger 30 includes a bridging header 50 provided at upper or lower ends of two adjacent ones of the heat exchanger cores 31 and configured so that refrigerant converging from the flat tubes 38 of one of the heat exchanger cores 31 that is located on a leeward side is distributed to the flat tubes 38 of one of the heat exchanger cores 31 that is located on a windward side.

**[0085]** The heat exchanger 30 according to Embodiment 3 includes a bridging header 50 configured so that refrigerant converging from the flat tubes 38 of one of the heat exchanger cores 31 that is located on a leeward side is distributed to the flat tubes 38 of one of the heat exchanger cores 31 that is located on a windward side. This makes it possible to lengthen a refrigerant flow passage represented by the sum of the heights of the two or more heat exchanger cores 31, thus making it possible to increase the flow passage differential pressure  $\Delta P_{\text{HEX}}$ . As a result of that,  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$  can be increased, and liquid retention is suppressed. This makes it possible to improve defrosting performance during a defrosting operation.

**[0086]** Further, in the heat exchanger 30 according to Embodiment 3, an upper or lower end of each of the flat tubes 38 of the two adjacent heat exchanger cores 31 is inserted in the bridging header 50. Moreover, in a case in which a gap between the upper or lower end of the flat tube 38 and a wall portion 51 of the bridging header 50 that faces the upper or lower end is defined as  $\delta$ , the heat exchanger 30 satisfies  $\delta \leq 3$  [mm].

**[0087]** The heat exchanger 30 according to Embodiment 3 is configured to satisfy  $\delta \leq 3$  [mm]. This makes it possible to suppress the occurrence of liquid retention even in consideration of the effect of a pressure loss, making it possible to improve heat exchanger performance.

Embodiment 4.

**[0088]** The following describes Embodiment 4. A description of overlaps with Embodiments 1 to 3 is omitted. Components that are identical or equivalent to those of Embodiments 1 to 3 are given identical reference signs.

**[0089]** Fig. 20 is an enlarged refrigerant circuit diagram of an outdoor unit 10 according to Embodiment 4 of an air-conditioning apparatus 100 with the outdoor unit 10 including heat exchangers 30. In Fig. 20, the white arrows indicate the flow of refrigerant during a defrosting operation.

**[0090]** As shown in Fig. 20, the outdoor unit 10 according to Embodiment 4 of an air-conditioning apparatus 100 includes a plurality of heat exchangers 30a to 30c. It should be noted that each of the heat exchangers 30a to 30c is any of the heat exchangers 30 described in Embodiments 1 to 3. Further, the number of heat exchangers 30a to 30c that the outdoor unit 10 of an air-conditioning apparatus 100 includes is not limited to 3 but may be at least 2.

**[0091]** An outlet side of the heat exchanger 30a and an outlet side of the heat exchanger 30b are configured to merge at a first merging section 63a. Further, an outlet side of the first merging section 63a and an outlet side of the heat exchanger 30c are configured to merge at a second merging section 63b. Further, a first expansion device 62a is provided in one of the refrigerant pipes that is located between the first merging section 63a and the second merging section 63b. Further, a second expansion device 62b is provided in one of the refrigerant pipes that is located between an outlet of the heat exchanger 30c and the second merging section 63b. Further, a first open-close valve 61a is provided in one of the refrigerant pipes that is located between a branch point on inlet sides of the heat exchangers 30a to 30c and an inlet of the heat exchanger 30c. Further, a second open-close valve 61b is provided in one of the refrigerant pipes that connects the inlet of the heat exchanger 30c with one of the refrigerant pipes that is located between the first merging section 63a and the first expansion device 62a. It should be noted that the first open-close valve 61a and the second open-close valve 61b are not valves that simply open and shut but may be valves whose opening degrees can be adjusted. In the following, the first expansion device 62a and the second expansion device 62b are collectively referred to as "expansion devices", and the first open-close valve 61a and the second open-close valve 61b are collectively referred to as "open-close valves".

**[0092]** Further, the air-conditioning apparatus 100 includes a controller 70 configured to control the expansion devices, the open-close valves, or other devices. The controller 70 is constituted by dedicated hardware or a central processing unit (CPU; also referred to as "processing unit", "arithmetic device", "microprocessor", or "processor") configured to execute programs that are stored in a storage unit (not illustrated). It should be noted that the controller 70 may be provided in the outdoor unit 10 or may be provided in the indoor unit 20.

**[0093]** In a case in which the controller 70 is the dedicated hardware, the controller 70 corresponds to a single circuit, a complex circuit, an application specific integrated circuit (ASIC), a field-programmable gate array (FPGA), or a combination thereof. Functional parts that the controller 70 implements may be implemented by each separate piece of hardware or may be implemented by one piece of hardware.

**[0094]** In a case in which the controller 70 is the CPU, functions that the controller 70 executes may be implemented by software, firmware, or a combination of the software and the firmware. The software and the firmware are described as programs and stored in the storage unit. The CPU implements the functions of the controller 70 by reading out and executing the programs stored in the storage unit. Note here that the storage unit is configured to store various types of information and includes data-rewritable nonvolatile semiconductor memory such as a flash memory, an EPROM, or an EEPROM.

**[0095]** Some of the functions of the controller 70 may be implemented by the dedicated hardware, and others of the functions of the controller 70 may be implemented by the software or the firmware.

**[0096]** Moreover, the flow passage differential pressure  $\Delta P_{\text{HEX}}$  is made greater than or equal to the liquid head  $\Delta P_{\text{HEAD}}$

by configuring the heat exchangers 30a to 30c so that during a defrosting operation, a flow of refrigerant through one or more of the heat exchangers 30a to 30c is in series with a flow of refrigerant through the other of the heat exchangers 30a to 30c and a flow of refrigerant through rest of the heat exchangers 30a to 30c is in parallel with the flow of refrigerant through the other of the heat exchangers 30a to 30c. Specifically, the controller 70 causes the second open-close valve 61b to be opened and causes the first open-close valve 61a to be shut. Further, the heat exchangers 30a to 30c are configured so that when the heat exchangers 30a to 30c function as evaporators, such as during a heating operation, a flow of refrigerant through each of the heat exchangers 30a to 30c is in parallel with a flow of refrigerant through the other of the heat exchangers 30a to 30c. Specifically, the controller 70 causes the second open-close valve 61b to be shut and causes the first open-close valve 61a to be opened.

**[0097]** By thus configuring the heat exchangers 30a to 30c so that during a defrosting operation, a flow of refrigerant through one or more of the heat exchangers 30a to 30c is in series with a flow of refrigerant through the other of the heat exchangers 30a to 30c and a flow of refrigerant through rest of the heat exchangers 30a to 30c is in parallel with the flow of refrigerant through the other of the heat exchangers 30a to 30c, the flow passage cross-sectional area is reduced with the flow rate of refrigerant being the same. Therefore, the velocity of flow of refrigerant through a flow passage through which hot-gas refrigerant flows upward increases, and the flow passage differential pressure  $\Delta P_{\text{HEX}}$  can be increased. This suppresses liquid retention, making it possible to improve defrosting performance during a defrosting operation. Further, by configuring the heat exchangers 30a to 30c so that when the heat exchangers 30a to 30c function as evaporators, a flow of refrigerant through each of the heat exchangers 30a to 30c is in parallel with a flow of refrigerant through the other of the heat exchangers 30a to 30c, the flow passage cross-sectional area is increased with the flow rate of refrigerant being the same. This reduces a pressure loss, thus making it possible to improve heating capacity.

**[0098]** As noted above, the air-conditioning apparatus 100 according to Embodiment 4 includes an outdoor unit 10 including a plurality of heat exchangers 30a to 30c and a controller 70 configured so that during a defrosting operation, one of more of the plurality of heat exchangers 30a to 30c are in series with the other of the heat exchangers 30a to 30c and configured so that when the heat exchangers 30a to 30c function as evaporators, a flow of the refrigerant through each of the heat exchangers 30a to 30c is in parallel with a flow of the refrigerant through the other of the heat exchangers 30a to 30c.

**[0099]** According to the air-conditioning apparatus 100 according to Embodiment 4, by configuring the heat exchangers 30a to 30c so that during a defrosting operation, a flow of refrigerant through one or more of the heat exchangers 30a to 30c is in series with a flow of refrigerant through the other of the heat exchangers 30a to 30c and a flow of refrigerant through rest of the heat exchangers 30a to 30c is in parallel with the flow of refrigerant through the other of the heat exchangers 30a to 30c, the flow passage cross-sectional area is reduced with the flow rate of refrigerant being the same. Therefore, the velocity of flow of refrigerant through a flow passage through which hot-gas refrigerant flows upward increases, and the flow passage differential pressure  $\Delta P_{\text{HEX}}$  can be increased. This suppresses liquid retention, making it possible to improve defrosting performance during a defrosting operation. Further, by configuring the heat exchangers 30a to 30c so that when the heat exchangers 30a to 30c function as evaporators, a flow of refrigerant through each of the heat exchangers 30a to 30c is in parallel with a flow of refrigerant through the other of the heat exchangers 30a to 30c, the flow passage cross-sectional area is increased with the flow rate of refrigerant being the same. This reduces a pressure loss, thus making it possible to improve heating capacity.

Embodiment 5.

**[0100]** The following describes Embodiment 5. A description of overlaps with Embodiments 1 to 4 is omitted. Components that are identical or equivalent to those of Embodiments 1 to 4 are given identical reference signs.

**[0101]** Fig. 21 is an enlarged refrigerant circuit diagram of an outdoor unit 10 according to Embodiment 5 of an air-conditioning apparatus 100 with the outdoor unit 10 including heat exchangers 30. In Fig. 21, the white arrows indicate the flow of refrigerant during a defrosting operation.

**[0102]** As shown in Fig. 21, the outdoor unit 10 according to Embodiment 5 of an air-conditioning apparatus 100 includes a plurality of heat exchangers 30a to 30c. It should be noted that each of the heat exchangers 30a to 30c is any of the heat exchangers 30 described in Embodiments 1 to 3. Further, the number of heat exchangers 30a to 30c that the outdoor unit 10 of an air-conditioning apparatus 100 includes is not limited to 3 but may be at least 2.

**[0103]** In Embodiment 5, a third open-close valve 61c is provided in one of the refrigerant pipes that is located between a branch point on the inlet sides of the heat exchangers 30a to 30b and an inlet of the heat exchanger 30b. A description of other components is omitted, as they are identical to those of the refrigerant circuit 101 described in Embodiment 4.

**[0104]** Moreover, the flow passage differential pressure  $\Delta P_{\text{HEX}}$  is made greater than or equal to the liquid head  $\Delta P_{\text{HEAD}}$  by configuring the heat exchangers 30a to 30c so that during a defrosting operation, a flow of refrigerant through one or more of the heat exchangers 30a to 30c is in series with a flow of refrigerant through the other of the heat exchangers 30a to 30c and a flow of refrigerant through rest of the heat exchangers 30a to 30c is in parallel with the flow of refrigerant

through the other of the heat exchangers 30a to 30c. Specifically, the controller 70 causes the second open-close valve 61b and the third open-close valve 61c to be opened and causes the first open-close valve 61a to be shut. Further, by fully closing an expansion device or an open-close valve, refrigerant is stopped from flowing through at least one of the heat exchangers 30a to 30c configured in parallel with one another and the other of the heat exchangers 30a to 30c is preferentially subjected to the defrosting operation. Moreover, by switching from fully closing one expansion device or one open-close valve to fully closing another expansion device or another open-close valve, switching from subjecting one of the heat exchangers 30a to 30c to the defrosting operation to subjecting another of the heat exchangers 30a to 30c to the defrosting operation is done. The timing of switching from fully closing one expansion device or one open-close valve to fully closing another expansion device or another open-close valve and switching from preferentially subjecting one of the heat exchangers 30a to 30c to the defrosting operation to preferentially subjecting another of the heat exchangers 30a to 30c to the defrosting operation is when a predetermined period of time has elapsed, based on a temperature sensed by a temperature sensor, such as a thermistor, provided on the outlet side of each of the heat exchangers 30a to 30c, or other timings.

**[0105]** Further, the heat exchangers 30a to 30c are configured so that when the heat exchangers 30a to 30c function as evaporators, such as during a heating operation, a flow of refrigerant through each of the heat exchangers 30a to 30c is in parallel with a flow of refrigerant through the other of the heat exchangers 30a to 30c. Specifically, the controller 70 causes the second open-close valve 61b to be shut and causes the first open-close valve 61a and the third open-close valve 61c to be opened.

**[0106]** Thus, the heat exchangers 30a to 30c are configured so that during a defrosting operation, a flow of refrigerant through one or more of the heat exchangers 30a to 30c is in series with a flow of refrigerant through the other of the heat exchangers 30a to 30c and a flow of refrigerant through rest of the heat exchangers 30a to 30c is in parallel with the flow of refrigerant through the other of the heat exchangers 30a to 30c. By so doing, the flow passage cross-sectional area is reduced with the flow rate of refrigerant being the same. Therefore, the velocity of flow of refrigerant through a flow passage through which hot-gas refrigerant flows upward increases, and the flow passage differential pressure  $\Delta P_{\text{HEX}}$  can be increased. Furthermore, by preferentially subjecting, to the defrosting operation, at least one of the heat exchangers 30a to 30c configured so that flows of the refrigerant through the heat exchangers 30a to 30c are in parallel with one another and then sequentially switching from preferentially subjecting one of the heat exchangers 30a to 30c to the defrosting operation to preferentially subjecting another of the heat exchangers 30a to 30c to the defrosting operation, lingering frost can be reduced. This further suppresses liquid retention, making it possible to further improve defrosting performance during a defrosting operation. Further, by configuring the heat exchangers 30a to 30c so that when the heat exchangers 30a to 30c function as evaporators, a flow of refrigerant through each of the heat exchangers 30a to 30c is in parallel with a flow of refrigerant through the other of the heat exchangers 30a to 30c, the flow passage cross-sectional area is increased with the flow rate of refrigerant being the same. This reduces a pressure loss, thus making it possible to improve heating capacity.

**[0107]** As noted above, in the air-conditioning apparatus 100 according to Embodiment 5, the controller 70 is configured so that during a defrosting operation, a flow of the refrigerant through one or more of the plurality of heat exchangers 30a to 30c is in series with a flow of the refrigerant through the other of the heat exchangers 30a to 30c and configured so that a flow of the refrigerant through rest of the heat exchangers 30a to 30c is in parallel with the flow of the refrigerant through the other of the heat exchangers 30a to 30c, and in a case in which there are a plurality of the heat exchangers 30a to 30c configured so that during a defrosting operation, flows of the refrigerant through the heat exchangers 30a to 30c are in parallel with one another, the refrigerant is stopped from flowing through at least one of the heat exchangers 30a to 30c and the other of the heat exchangers 30a to 30c is preferentially subjected to the defrosting operation.

**[0108]** According to the air-conditioning apparatus 100 according to Embodiment 5, by preferentially subjecting, to the defrosting operation, at least one of the heat exchangers 30a to 30c configured so that flows of refrigerant through the heat exchangers 30a to 30c are in parallel with one another and then sequentially switching from preferentially subjecting one of the heat exchangers 30a to 30c to the defrosting operation to preferentially subjecting another of the heat exchangers 30a to 30c to the defrosting operation, lingering frost can be reduced. This further suppresses liquid retention, making it possible to further improve defrosting performance during a defrosting operation.

#### Embodiment 6.

**[0109]** The following describes Embodiment 6. A description of overlaps with Embodiments 1 to 5 is omitted. Components that are identical or equivalent to those of Embodiments 1 to 5 are given identical reference signs.

**[0110]** Fig. 22 is a cross-sectional schematic view of a flat tube 38 of a heat exchanger 30 according to Embodiment 6. Fig. 23 is a cross-sectional schematic view of a flat tube 38 of a modification of the heat exchanger 30 according to Embodiment 6. Fig. 24 is a side schematic view of the flat tube 38 of the modification of the heat exchanger 30 according to Embodiment 6.

**[0111]** As shown in Fig. 22, the flat tube 38 of the heat exchanger 30 is provided with a plurality of partition posts 38a

inside. These partition posts 38a are placed along a direction parallel with the length of a cross-section of the flat tube 38, extend along a direction parallel with the length of the flat tube 38, and divide the inside of the flat tube 38 into a plurality of spaces. Furthermore, the flat tube 38 is a grooved flat tube provided with a plurality of inward projecting portions 38b each provided between adjacent ones of the partition posts 38a. These projecting portions 38b extend along a direction parallel with the length of the flat tube 38. Further, as shown in Figs. 23 and 24, the flat tube 38 of the heat exchanger 30 is an end-shrunk flat tube having a distal portion 38c subjected to tube shrinking so that the outer diameter of the flat tube 38 decreases toward a distal end.

[0112] Thus, the flat tube 38 of the heat exchanger 30 is made a grooved flat tube or an end-shrunk flat tube. By so doing, the flow passage cross-sectional area is reduced with the flow rate of refrigerant being the same. Therefore, the velocity of flow of refrigerant through a flow passage through which hot-gas refrigerant flows upward increases, and the flow passage differential pressure  $\Delta P_{\text{HEX}}$  can be increased. This suppresses liquid retention, making it possible to improve defrosting performance during a defrosting operation.

[0113] As noted above, in the heat exchanger 30 according to Embodiment 6, each of the flat tubes 38 is provided with a plurality of partition posts 38a configured to partition an internal flow passage and an inward projecting portion 38b provided between adjacent ones of the partition posts 38a, or each of the flat tubes 38 has a distal portion 38c subjected to tube shrinking so that an outer diameter of the flat tube 38 decreases toward a distal end.

[0114] According to the heat exchanger 30 according to Embodiment 6, each of the flat tubes 38 has a plurality of projecting portions 38b each formed along a flow passage between adjacent ones of the partition posts 38a, or each of the flat tubes 38 has a distal portion 38c subjected to tube shrinking so that an outer diameter of the flat tube 38 decreases toward a distal end. By thus making the flat tube 38 a grooved flat tube or an end-shrunk flat tube, the flow passage cross-sectional area is reduced with the flow rate of refrigerant being the same. Therefore, the velocity of flow of refrigerant through a flow passage through which hot-gas refrigerant flows upward increases, and the flow passage differential pressure  $\Delta P_{\text{HEX}}$  can be increased. As a result of that, liquid retention is suppressed, and defrosting performance during a defrosting operation can be improved.

#### Embodiment 7.

[0115] The following describes Embodiment 7. A description of overlaps with Embodiments 1 to 6 is omitted. Components that are identical or equivalent to those of Embodiments 1 to 6 are given identical reference signs.

[0116] Fig. 25 is a diagram showing a relationship between types of refrigerant that are used in a refrigerant circuit 101 of an air-conditioning apparatus 100 according to Embodiment 7 and  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$ .

[0117] As shown in Fig. 25, it is found that all of the pure refrigerants HFO1123, HFO1132(E), R1234yf, R1234ze(E), R1234ze(Z), R1233zd(E), propane (R290), and fluoroethane (R161) are higher in  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$  than the pure refrigerants R32 and R410A.

[0118] Accordingly, Embodiment 7 uses any of the pure refrigerants HFO1123, HFO1132(E), R1234yf, R1234ze(E), R1234ze(Z), R1233zd(E), propane (R290), and fluoroethane (R161) as refrigerant that circulates through the refrigerant circuit 101 of the air-conditioning apparatus 100.

[0119] By thus using any of the foregoing pure refrigerants as refrigerant that circulates through the refrigerant circuit 101 of the air-conditioning apparatus 100,  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$  can be improved. This makes it possible to suppress the occurrence of liquid retention, making it possible to improve heat exchanger performance.

[0120] As noted above, in the air-conditioning apparatus 100 according to Embodiment 7, the refrigerant is a pure refrigerant selected from the group consisting of HFO1123, HFO1132(E), R1234yf, R1234ze(E), R1234ze(Z), R1233zd(E), propane (R290), and fluoroethane (R161).

[0121] The air-conditioning apparatus 100 according to Embodiment 7 uses any of the foregoing pure refrigerants as refrigerant that circulates through the refrigerant circuit 101, thus making it possible to improve  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}}$ . This suppresses the occurrence of liquid retention, making it possible to improve heat exchanger performance.

#### Embodiment 8.

[0122] The following describes Embodiment 8. A description of overlaps with Embodiments 1 to 7 is omitted. Components that are identical or equivalent to those of Embodiments 1 to 7 are given identical reference signs.

[0123] Fig. 26 is a front view of a heat exchanger 30 according to Embodiment 8 of an air-conditioning apparatus 100. In Fig. 26, the white arrows indicate the flow of refrigerant during a cooling operation. Further, Fig. 26 shows the height H and width L of a heat exchanger core 31 and shows the widths of different regions of the heat exchanger core 31 in the order of  $L_1$ ,  $L_2$ , ... from a downstream side.

[0124] During a cooling operation, the heat exchanger 30 according to Embodiment 8 functions as a condenser configured to condense the refrigerant by rejecting the heat of the refrigerant to the outdoor air. As shown in Fig. 26, the heat exchanger 30 includes a heat exchanger core 31 including a plurality of flat tubes 38 and a plurality of fins 39. The

flat tubes 38 are placed in parallel with one another at spacings in a horizontal direction (i.e. a right-left direction of Fig. 26) so that wind generated by the fan 13 flows, and refrigerant flows in a vertical direction through the flat tubes 38, which extend in a vertical direction (i.e. an up-down direction of Fig. 26). The fins 39 are each connected across a space between adjacent ones of the flat tubes 38 and transfer heat to the flat tubes 38. The fins 39 are intended to improve the efficiency of heat exchange between air and refrigerant. Usable examples of the fins 39 include, but are not limited to, corrugated fins. The fins 39 are dispensable because heat exchange between air and refrigerant takes place on the surfaces of the flat tubes 38.

[0125] A first header 34 is provided at a lower end of the heat exchanger core 31. Lower ends of the flat tubes 38 of the heat exchanger core 31 are directly inserted in the first header 34. Further, a second header 35 is provided at an upper end of the heat exchanger core 31. Upper ends of the flat tubes 38 of the heat exchanger core 31 are directly inserted in the second header 35.

[0126] A hot-gas refrigerant inlet 32 is formed at one end of the second header 35, and the hot-gas refrigerant inlet 32 is connected to the refrigerant circuit 101 of the air-conditioning apparatus 100 via a gas pipe 37. Further, a liquid refrigerant outlet 33 is formed at the other end of the second header 35, and the liquid refrigerant outlet 33 is connected to the refrigerant circuit 101 of the air-conditioning apparatus 100 via a liquid pipe 36. The second header 35 causes high-temperature and high-pressure gas refrigerant from the compressor 11 to flow into the heat exchanger 30 during a cooling operation and causes low-temperature and high-pressure gas refrigerant subjected to heat exchange in the heat exchanger 30 to flow out to the refrigerant circuit 101. Further, the second header 35 causes low-temperature and low-pressure two-phase refrigerant to flow into the heat exchanger 30 during a heating operation and causes low-temperature and low-pressure liquid refrigerant subjected to heat exchange in the heat exchanger 30 to flow out to the refrigerant circuit 101.

[0127] The plurality of flat tubes 38, the plurality of fins 39, the first header 34, and the second header 35 are all made of aluminum and joined to each other by brazing.

[0128] In the heat exchanger 30 according to Embodiment 8, as shown in Fig. 26, a partition plate 40 is provided in the second header 35. This partition plate 40 is provided to divide a flow passage of the heat exchanger core 31 into a plurality of regions in a horizontal direction. Further, the partition plate 40 is provided so that a flow passage in each region of the heat exchanger core 31 is opposite in flow to a flow passage in an adjacent region. In Embodiment 8, the flow passage of the heat exchanger core 31 is divided by the partition plate 40 into two regions  $T_1$  and  $T_2$ . Further, by providing the partition plate 40 in the second header 35, a hot-gas refrigerant merging region  $M_1$  is formed in the first header 34.

[0129] Moreover, hot-gas refrigerant having flowed into the second header 35 flows as downward flows through ones of the flat tubes 38 of the heat exchanger core 31 that are placed in the region  $T_1$ , merges in the merging region  $M_1$  of the first header 34, flows as upward flows through ones of the flat tubes 38 of the heat exchanger core 31 that are placed in the region  $T_2$ , and then flows out from the second header 35. That is, the region  $T_1$  is a downward flow region, and the region  $T_2$  is an upward flow region. Further, the merging region  $M_1$  of the first header 34 serves as a hot-gas refrigerant inflow port for the upward flow region.

[0130] In the region  $T_2$ , there is an occurrence of liquid retention in which when hot-gas refrigerant flows as upward flows through the flat tubes 38 of the heat exchanger core 31, refrigerant liquefied by the effect of gravity is undesirably retained without being able to go upward. To address this problem, the heat exchanger 30 according to Embodiment 8 is configured such that in a case in which the total flow passage cross-sectional area of the heat exchanger core 31 in the region  $T_2$  is defined as  $A_r [\text{m}^2] = a \times N_r [\text{m}^2]$ , where  $N_r$  is the number of flat tubes 38 of the heat exchanger core 31 that are placed in the region  $T_2$ , which is an upward flow region, the height of the heat exchanger core 31 is defined as  $H [\text{m}]$ , a differential pressure of a refrigerant flow passage is defined as  $\Delta P_{\text{HEX}}$ , and a liquid head is defined as  $\Delta P_{\text{HEAD}}$ ,  $\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}} = (5.94635 \times 10^{-4} \times A_r^{-1.75030})/(8.4303H + 0.8779) > 1$  is satisfied. The heat exchanger 30 thus configured makes it possible to suppress the occurrence of liquid retention in which when refrigerant having flowed in from the hot-gas refrigerant inlet 32 formed in an upper part of the heat exchanger 30 flows as upward flows through the flat tubes 38 of the heat exchanger core 31 in the region  $T_2$ , refrigerant liquefied by the effect of gravity is retained without being able to go upward, making it possible to reduce deterioration of defrosting performance. Further, by providing the partition plate 40 in the second header 35, the flow passage cross-sectional area is reduced with the flow rate of refrigerant being the same. Therefore, the velocity of flow of refrigerant increases, and the flow passage differential pressure  $\Delta P_{\text{HEX}}$  can be increased. This suppresses liquid retention, making it possible to improve defrosting performance during a defrosting operation.

Embodiment 9.

[0131] The following describes Embodiment 9. A description of overlaps with Embodiments 1 to 8 is omitted. Components that are identical or equivalent to those of Embodiments 1 to 8 are given identical reference signs.

[0132] Fig. 27 is a front view of a heat exchanger 30 according to Embodiment 9 of an air-conditioning apparatus 100.

In Fig. 27, the white arrows indicate the flow of refrigerant during a cooling operation. Further, Fig. 27 shows the height H and width L of a heat exchanger core 31 and shows the widths of different regions of the heat exchanger core 31 in the order of  $L_1, L_2, \dots$  from a downstream side.

**[0133]** During a cooling operation, the heat exchanger 30 according to Embodiment 9 functions as a condenser configured to condense the refrigerant by rejecting the heat of the refrigerant to the outdoor air. As shown in Fig. 27, the heat exchanger 30 includes a heat exchanger core 31 including a plurality of flat tubes 38 and a plurality of fins 39. The flat tubes 38 are placed in parallel with one another at spacings in a horizontal direction (i.e. a right-left direction of Fig. 27) so that wind generated by the fan 13 flows, and refrigerant flows in a vertical direction through the flat tubes 38, which extend in a vertical direction (i.e. an up-down direction of Fig. 27). The fins 39 are each connected across a space between adjacent ones of the flat tubes 38 and transfer heat to the flat tubes 38. The fins 39 are intended to improve the efficiency of heat exchange between air and refrigerant. Usable examples of the fins 39 include, but are not limited to, corrugated fins. The fins 39 are dispensable because heat exchange between air and refrigerant takes place on the surfaces of the flat tubes 38.

**[0134]** A first header 34 is provided at a lower end of the heat exchanger core 31. Lower ends of the flat tubes 38 of the heat exchanger core 31 are directly inserted in the first header 34. Further, a second header 35 is provided at an upper end of the heat exchanger core 31. Upper ends of the flat tubes 38 of the heat exchanger core 31 are directly inserted in the second header 35.

**[0135]** A hot-gas refrigerant inlet 32 is formed at one end of the second header 35, and the hot-gas refrigerant inlet 32 is connected to the refrigerant circuit 101 of the air-conditioning apparatus 100 via a gas pipe 37. The second header 35 causes high-temperature and high-pressure gas refrigerant from the compressor 11 to flow into the heat exchanger 30 during a cooling operation and causes low-temperature and high-pressure gas refrigerant subjected to heat exchange in the heat exchanger 30 to flow out to the refrigerant circuit 101 during a heating operation.

**[0136]** Further, a liquid refrigerant outlet 33 is formed at one end of the first header 34 located opposite the one end of the second header 35, and the liquid refrigerant outlet 33 is connected to the refrigerant circuit 101 of the air-conditioning apparatus 100 via a liquid pipe 36. The first header 34 causes low-temperature and low-pressure two-phase refrigerant to flow into the heat exchanger 30 during a heating operation and causes low-temperature and high-pressure liquid refrigerant subjected to heat exchange in the heat exchanger 30 to flow out to the refrigerant circuit 101 during a cooling operation.

**[0137]** The plurality of flat tubes 38, the plurality of fins 39, the first header 34, and the second header 35 are all made of aluminum and joined to each other by brazing.

**[0138]** In the heat exchanger 30 according to Embodiment 9, as shown in Fig. 27, partition plates 40 are provided in the first header 34 and the second header 35. These partition plate 40 are provided to divide a flow passage of the heat exchanger core 31 into a plurality of regions in a horizontal direction. Further, the partition plates 40 are provided so that a flow passage in each region of the heat exchanger core 31 is opposite in flow to a flow passage in an adjacent region. In Embodiment 9, the flow passage of the heat exchanger core 31 is divided by the two partition plates 40 into three regions  $T_1, T_2$ , and  $T_3$ . Further, by providing the partition plates 40 in the first header 34 and the second header 35, hot-gas refrigerant merging regions  $M_1$  and  $M_2$  are formed in the first header 34 and the second header 35, respectively.

**[0139]** Moreover, hot-gas refrigerant having flowed into the second header 35 flows as downward flows through ones of the flat tubes 38 of the heat exchanger core 31 that are placed in the region  $T_1$ , merges in the merging region  $M_1$  of the first header 34, and flows as upward flows through ones of the flat tubes 38 of the heat exchanger core 31 that are placed in the region  $T_2$ . After that, the hot-gas refrigerant merges in the merging region  $M_2$  of the second header 35, flows as downward flows through ones of the flat tubes 38 of the heat exchanger core 31 that are placed in the region  $T_3$ , and then flows out from the first header 34. That is, the region  $T_1$  and  $T_3$  are downward flow regions, and the region  $T_2$  is an upward flow region. Further, the merging region  $M_1$  of the first header 34 serves as a hot-gas refrigerant inflow port for the upward flow region.

**[0140]** In the region  $T_2$ , there is an occurrence of liquid retention in which when hot-gas refrigerant flows as upward flows through the flat tubes 38 of the heat exchanger core 31, refrigerant liquefied by the effect of gravity is undesirably retained without being able to go upward. To address this problem, the heat exchanger 30 according to Embodiment 9 is configured such that in a case in which the total flow passage cross-sectional area of the heat exchanger core 31 in the region  $T_2$  is defined as  $A_T [m^2] = a \times N_T [m^2]$ , where  $N_T$  is the number of flat tubes 38 of the heat exchanger core 31 that are placed in the region  $T_2$ , which is an upward flow region, the height of the heat exchanger core 31 is defined as H [m], a differential pressure of a refrigerant flow passage is defined as  $\Delta P_{HEX}$ , and a liquid head is defined as  $\Delta P_{HEAD}$ ,  $\Delta P_{HEX}/\Delta P_{HEAD} = (5.94635 \times 10^{-4} \times A_T^{-1.75030})/(8.4303H + 0.8779) > 1$  is satisfied. The heat exchanger 30 thus configured makes it possible to suppress the occurrence of liquid retention in which when refrigerant having flowed in from the hot-gas refrigerant inlet 32 formed in an upper part of the heat exchanger 30 flows as upward flows through the flat tubes 38 of the heat exchanger core 31 in the region  $T_2$ , refrigerant liquefied by the effect of gravity is retained without being able to go upward, making it possible to reduce deterioration of defrosting performance. Further, by providing the partition plates 40 in the first header 34 and the second header 35, the flow passage cross-sectional area is reduced with the

flow rate of refrigerant being the same. Therefore, the velocity of flow of refrigerant increases, and the flow passage differential pressure  $\Delta P_{\text{HEX}}$  can be increased. This suppresses liquid retention, making it possible to improve defrosting performance during a defrosting operation.

## 5 Reference Signs List

[0141] 10: outdoor unit, 11: compressor, 12: flow switching device, 13: fan, 20: indoor unit, 21: expansion device, 22: indoor heat exchanger, 23: indoor fan, 30: heat exchanger, 30a: heat exchanger, 30b: heat exchanger, 30c: heat exchanger, 31: heat exchanger core, 32: hot-gas refrigerant inlet, 33: liquid refrigerant outlet, 34: first header, 35: second header, 36: liquid pipe, 37: gas pipe, 38 flat tube, 38a: partition post, 38b: projecting portion, 38c: distal portion, 39: fin, 40: partition plate, 50: bridging header, 51: wall portion, 61a: first open-close valve, 61b: second open-close valve, 61c: third open-close valve, 62a: first expansion device, 62b: second expansion device, 63a: first merging section, 63b: second merging section, 70: controller, 100: air-conditioning apparatus, 101: refrigerant circuit

15

## Claims

1. A heat exchanger that is mounted in an outdoor unit of an air-conditioning apparatus, the heat exchanger comprising one heat exchanger core or two or more heat exchanger cores each including a plurality of flat tubes that extend in an up-down direction and through which refrigerant flows as upward flows when the heat exchanger functions as a condenser, the two or more heat exchanger cores being placed along a direction of flow of air, wherein in a case in which a total flow passage cross-sectional area of each of the heat exchanger cores is defined as  $A \text{ [m}^2\text{]} = a \times N \text{ [m}^2\text{]}$ , where  $a \text{ [m}^2\text{]}$  is a flow passage cross-sectional area of each of the flat tubes and  $N$  is a number of the flat tubes, a height of each of the heat exchanger cores is defined as  $H \text{ [m]}$ , a differential pressure of a refrigerant flow passage is defined as  $\Delta P_{\text{HEX}}$ , and a liquid head is defined as  $\Delta P_{\text{HEAD}}$ ,

25

$$\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}} = (5.94635 \times 10^{-4} \times A^{-1.75030})/(8.4303H + 0.8779) > 1 \text{ is satisfied.}$$

30

is satisfied.

2. The heat exchanger of claim 1, further comprising a hot-gas refrigerant inlet formed in a lower part thereof when the heat exchanger functions as a condenser.

3. The heat exchanger of claim 1, further comprising a hot-gas refrigerant merging region formed in a lower part thereof when the heat exchanger functions as a condenser.

4. The heat exchanger of claim 2, comprising:

the one heat exchanger core;  
a first header provided at a lower end of the heat exchanger core; and  
a second header provided at an upper end of the heat exchanger core,  
wherein the hot-gas refrigerant inlet is formed at one end of the first header,  
the heat exchanger further comprising a liquid refrigerant outlet that is formed at one end of the second header located opposite the one end of the first header and through which refrigerant flows out when the heat exchanger functions as a condenser.

5. The heat exchanger of any one of claims 1 to 3, comprising:

the one heat exchanger core;  
a first header provided at a lower end of the heat exchanger core;  
a second header provided at an upper end of the heat exchanger core; and  
a partition plate provided in at least the first header and configured to divide a flow passage of the heat exchanger core into a plurality of regions in a width direction,  
wherein in a case in which a width of the heat exchanger core is defined as  $L \text{ [m]}$  and a width of a furthest downstream region of the heat exchanger core is defined as  $L_1$ ,

55

$$20 [\%] \leq L_1/L \leq 50 [\%]$$

is satisfied.

6. The heat exchanger of claim 2, comprising:

the two or more heat exchanger cores placed along the direction of flow of air;  
a first header provided at a lower end of one of the heat exchanger cores that is located on a furthest leeward side; and  
a second header provided at an upper or lower end of one of the heat exchanger cores that is located at a furthest windward side,  
wherein the hot-gas refrigerant inlet is formed at one end of the first header,  
the heat exchanger further comprising:

a liquid refrigerant outlet that is formed at one end of the second header located on a same side as the one end of the first header and through which refrigerant flows out when the heat exchanger functions as a condenser; and  
a bridging header provided at upper or lower ends of two adjacent ones of the heat exchanger cores and configured so that refrigerant converging from the flat tubes of one of the heat exchanger cores that is located on a leeward side is distributed to the flat tubes of one of the heat exchanger cores that is located on a windward side.

7. The heat exchanger of claim 6, wherein

an upper or lower end of each of the flat tubes of the two adjacent heat exchanger cores is inserted in the bridging header, and  
in a case in which a gap between the upper or lower end of the flat tube and a wall portion of the bridging header that faces the upper or lower end is defined as  $\delta$ ,  
 $\delta \leq 3$  [mm] is satisfied.

8. The heat exchanger of any one of claims 1 to 3, wherein in a case in which a width of each of the heat exchanger cores is defined as L [m],  $H/L > 1$  is satisfied.

9. The heat exchanger of any one of claims 1 to 3, wherein  $H \geq 0.42$  [m] is satisfied.

10. The heat exchanger of any one of claims 1 to 9, wherein each of the flat tubes is provided with a plurality of partition posts configured to partition an internal flow passage and an inward projecting portion provided between adjacent ones of the partition posts.

11. The heat exchanger of any one of claims 1 to 9, wherein each of the flat tubes has a distal portion subjected to tube shrinking so that an outer diameter of the flat tube decreases toward a distal end.

12. A heat exchanger that is mounted in an outdoor unit of an air-conditioning apparatus, the heat exchanger comprising one heat exchanger core or two or more heat exchanger cores each including a plurality of flat tubes that extend in an up-down direction and through which refrigerant flows as upward flows when the heat exchanger functions as a condenser, the two or more heat exchanger cores being placed along a direction of flow of air,  
wherein in a case in which a total flow passage cross-sectional area of each of the heat exchanger cores in an upward flow region is defined as  $A_r$  [m<sup>2</sup>] =  $a \times N_r$  [m<sup>2</sup>], where  $a$  [m<sup>2</sup>] is a flow passage cross-sectional area of each of the flat tubes and  $N_r$  is a number of the flat tubes through which hot-gas refrigerant flows as upward flows in the heat exchanger, a height of each of the heat exchanger cores is defined as H [m], a differential pressure of a refrigerant flow passage is defined as  $\Delta P_{\text{HEX}}$ , and a liquid head is defined as  $\Delta P_{\text{HEAD}}$ ,

$$\Delta P_{\text{HEX}}/\Delta P_{\text{HEAD}} = (5.94635 \times 10^{-4} \times A_r^{-1.75030})/(8.4303H + 0.8779) > 1$$

is satisfied.

13. An outdoor unit of an air-conditioning apparatus comprising the heat exchanger of any one of claims 1 to 12.

14. An air-conditioning apparatus comprising:

5           the outdoor unit of the air-conditioning apparatus of claim 13;  
           an indoor unit of an air-conditioning apparatus; and  
           a refrigerant circuit that is constituted by the outdoor unit of the air-conditioning apparatus and the indoor unit  
           of the air-conditioning apparatus and through which refrigerant circulates.

10   15. The air-conditioning apparatus of claim 14, wherein the outdoor unit of the air-conditioning apparatus includes a  
       plurality of the heat exchangers,  
       the air-conditioning apparatus further comprising a controller configured so that during a defrosting operation, a flow  
       of the refrigerant through one or more of the plurality of heat exchangers is in series with a flow of the refrigerant  
       through an other of the heat exchangers and configured so that when the heat exchangers function as evaporators,  
       a flow of the refrigerant through each of the heat exchangers is in parallel with a flow of the refrigerant through an  
       other of the heat exchangers.

16. The air-conditioning apparatus of claim 15, wherein

20           the controller is configured so that during a defrosting operation, a flow of the refrigerant through one or more  
           of the plurality of heat exchangers is in series with a flow of the refrigerant through an other of the heat exchangers  
           and configured so that a flow of the refrigerant through rest of the heat exchangers is in parallel with the flow  
           of the refrigerant through the other of the heat exchangers, and  
           in a case in which there are a plurality of the heat exchangers configured so that during a defrosting operation,  
       25       flows of the refrigerant through the heat exchangers are in parallel with one another, the refrigerant is stopped  
           from flowing through at least one of the heat exchangers and an other of the heat exchangers is preferentially  
           subjected to the defrosting operation.

30   17. The air-conditioning apparatus of any one of claims 14 to 16, wherein the refrigerant is a pure refrigerant selected  
       from the group consisting of HFO1123, HFO1132(E), R1234yf, R1234ze(E), R1234ze(Z), R1233zd(E), propane  
       (R290), and fluoroethane (R161).

35

40

45

50

55

FIG. 1

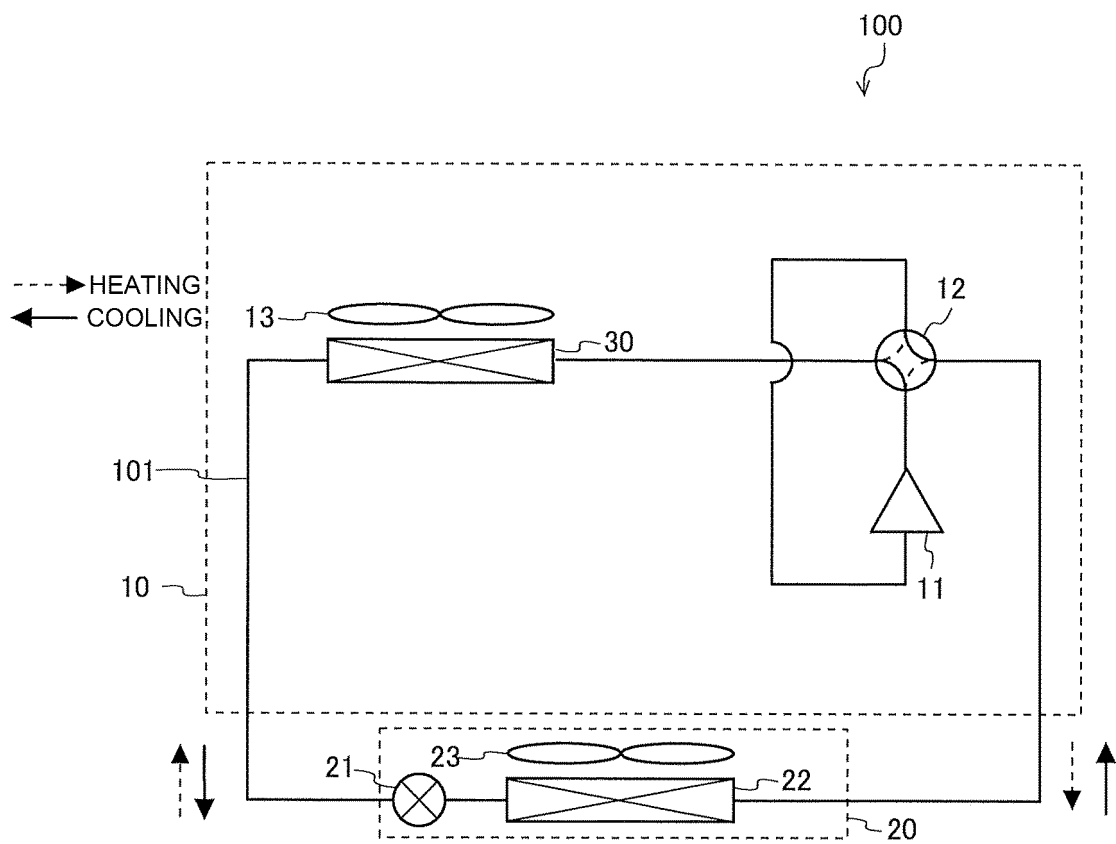


FIG. 2

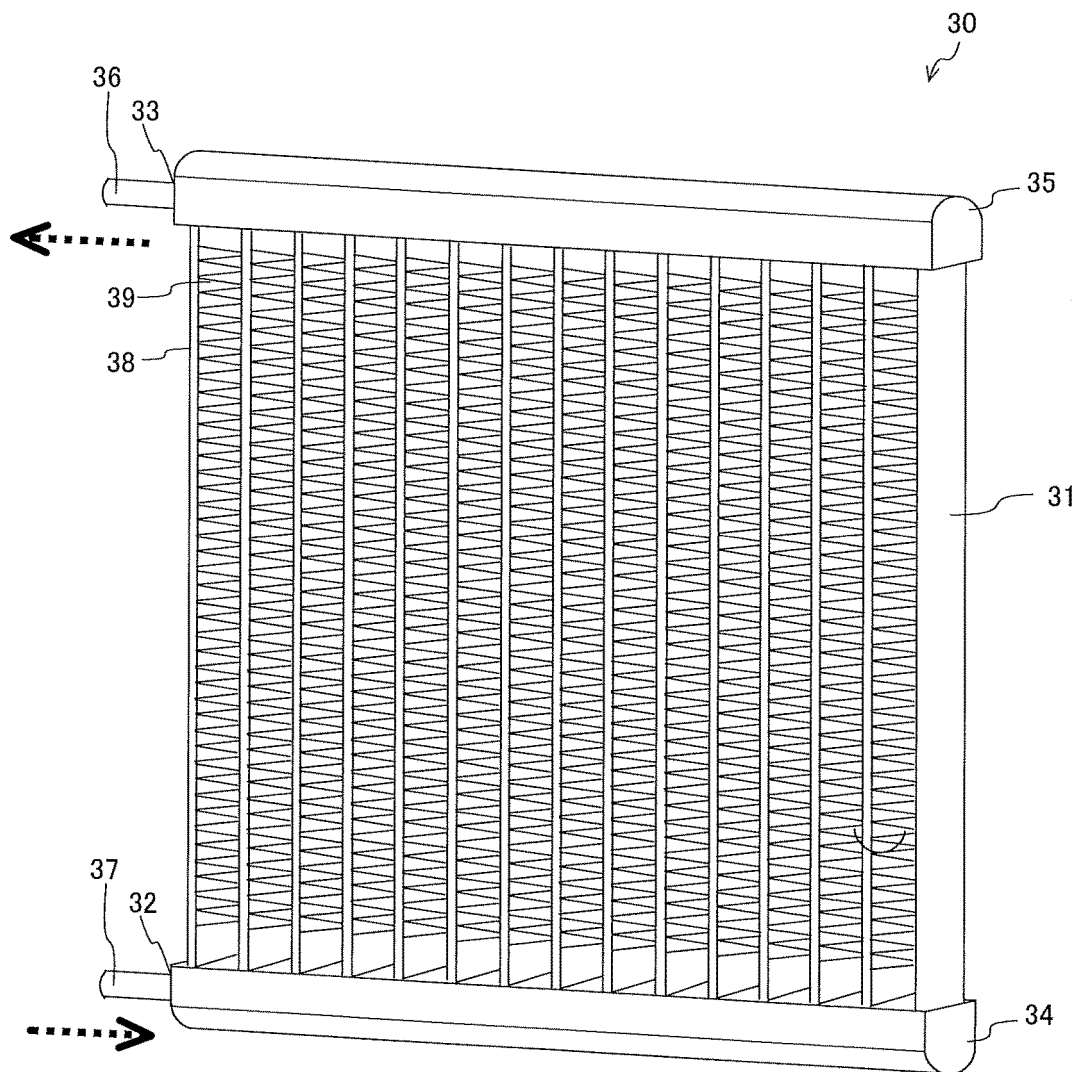


FIG. 3

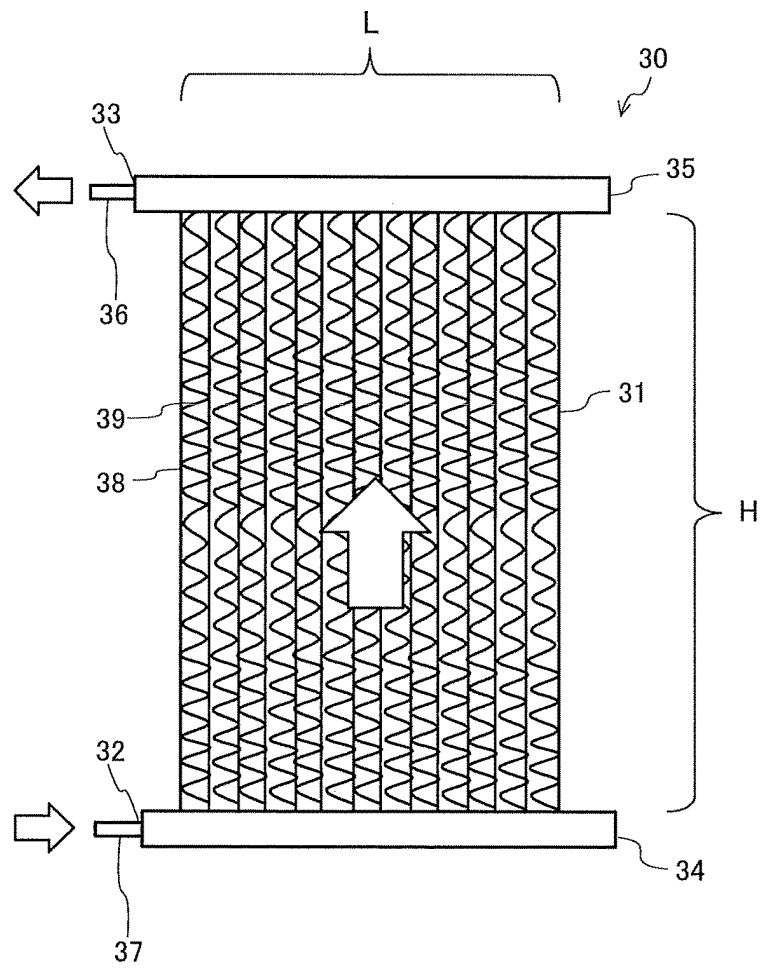


FIG. 4

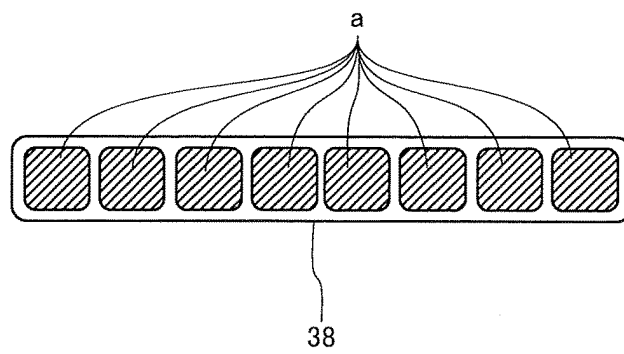


FIG. 5

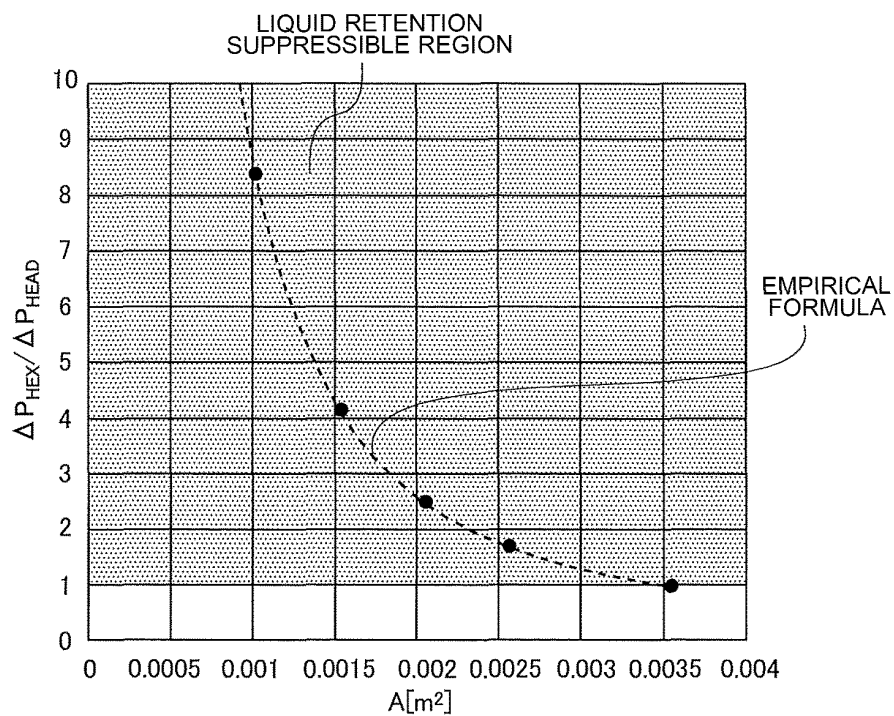


FIG. 6

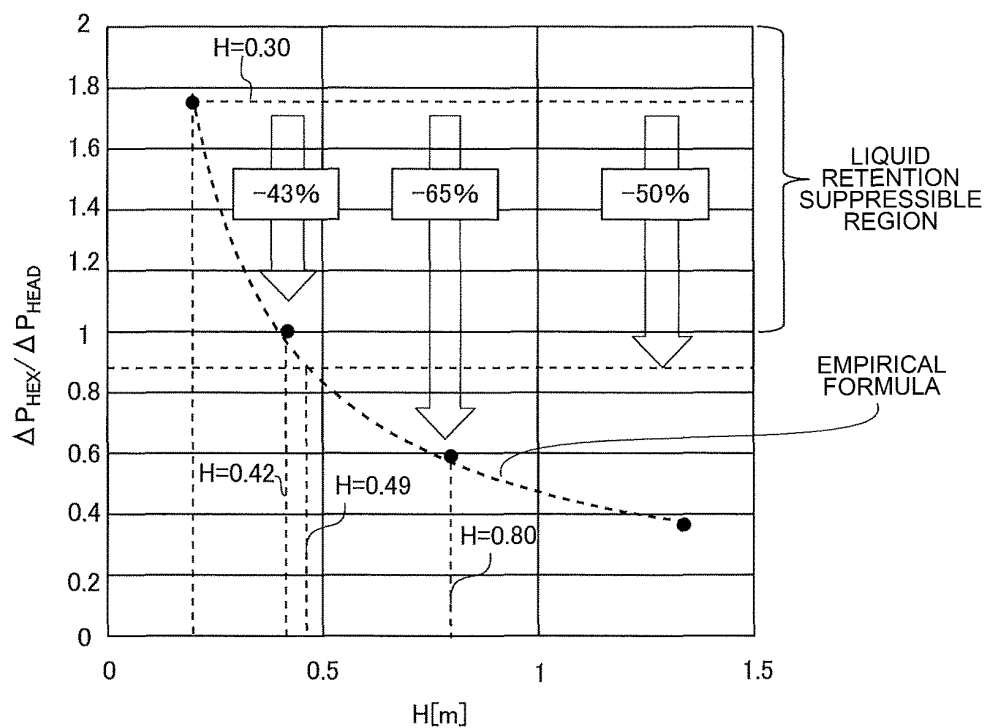


FIG. 7

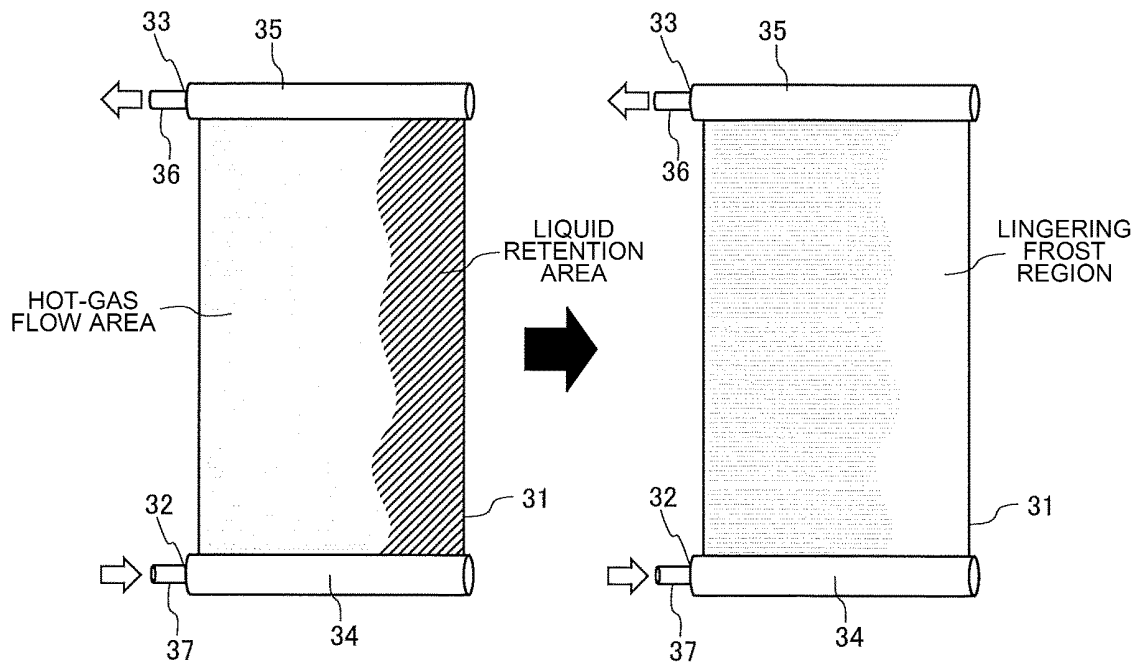


FIG. 8

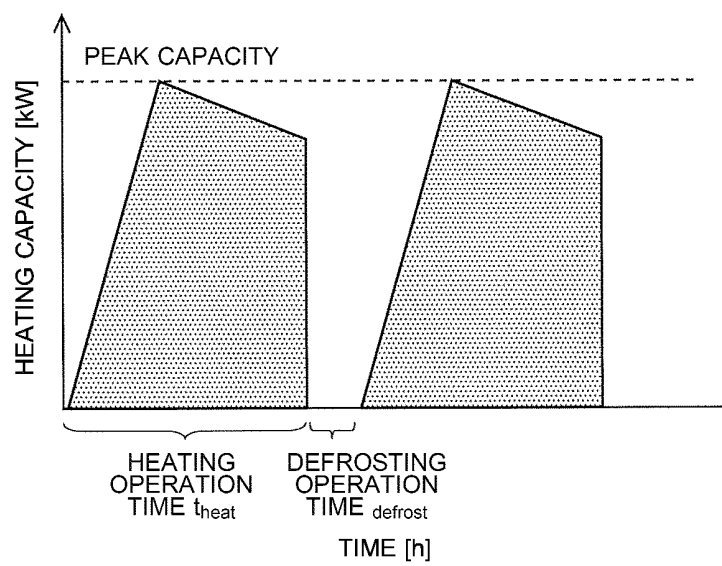


FIG. 9

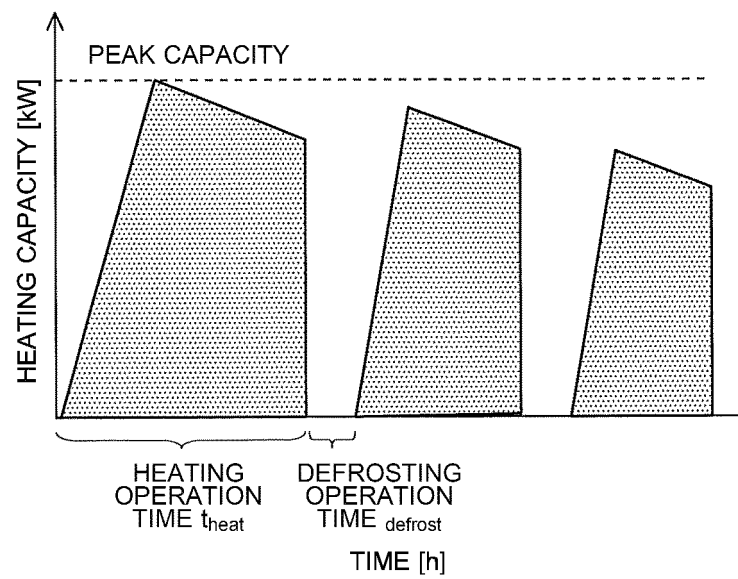


FIG. 10

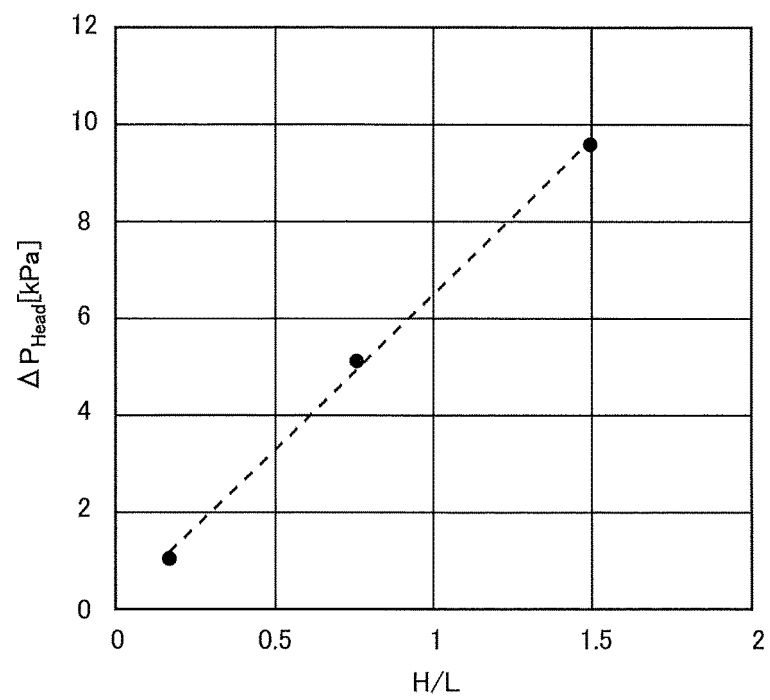


FIG. 11

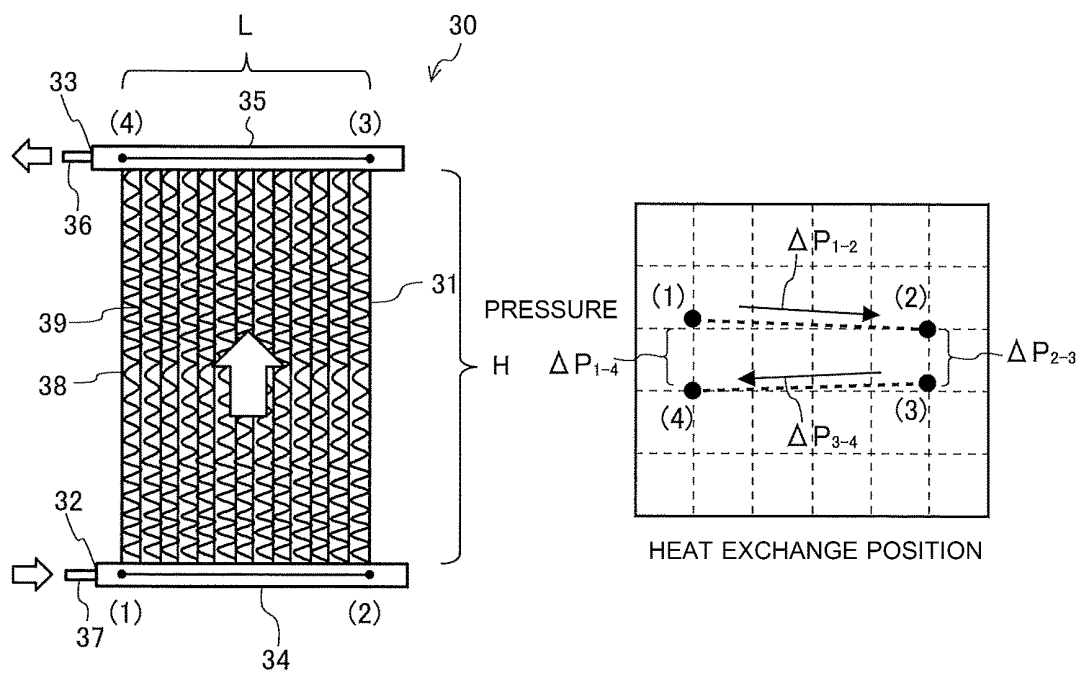


FIG. 12

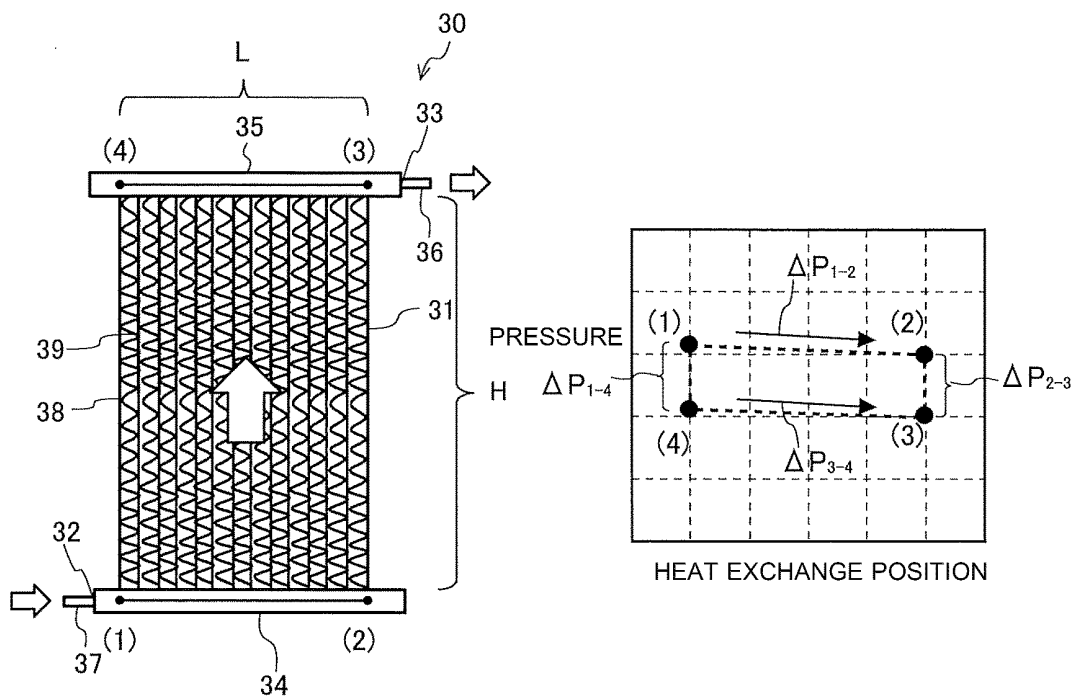


FIG. 13

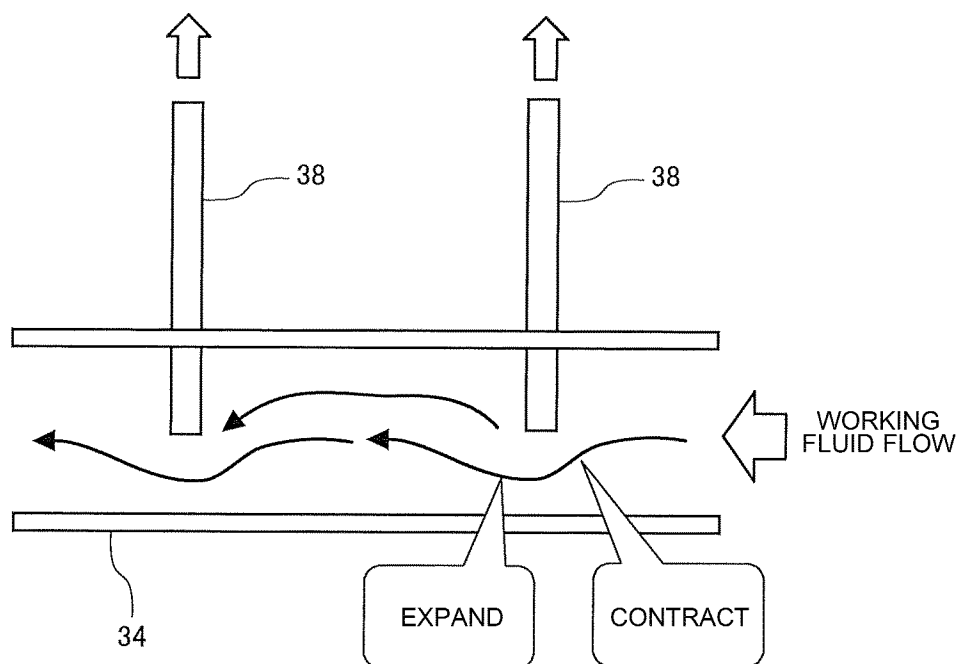


FIG. 14

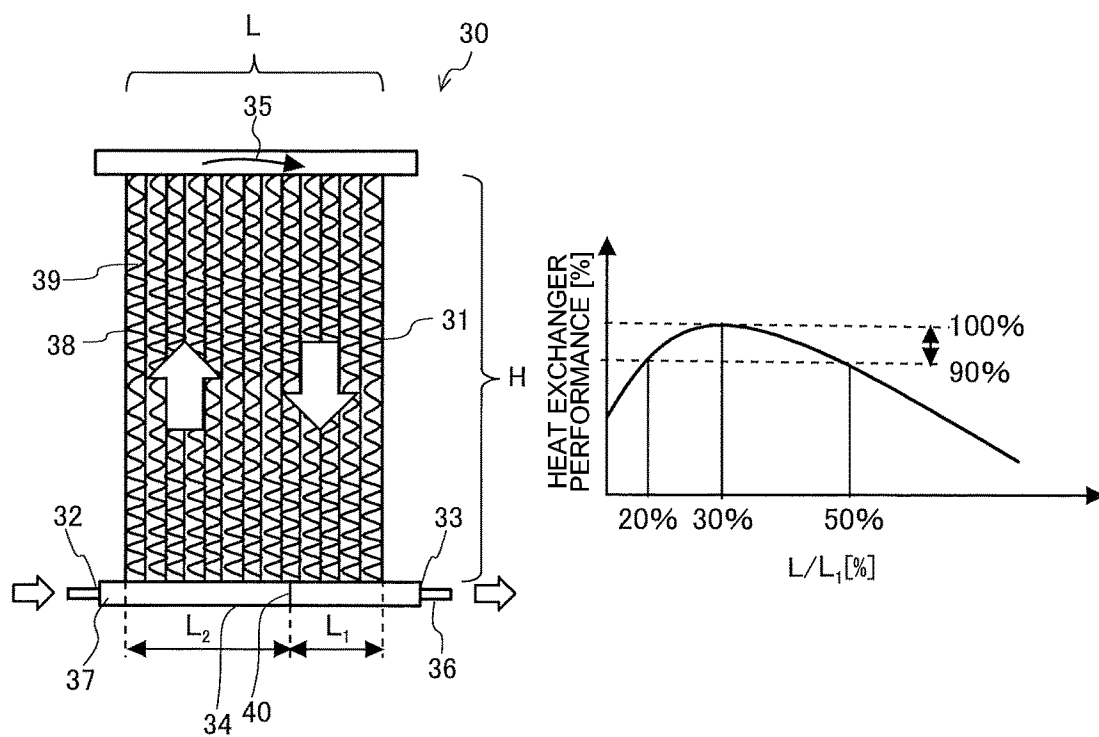


FIG. 15

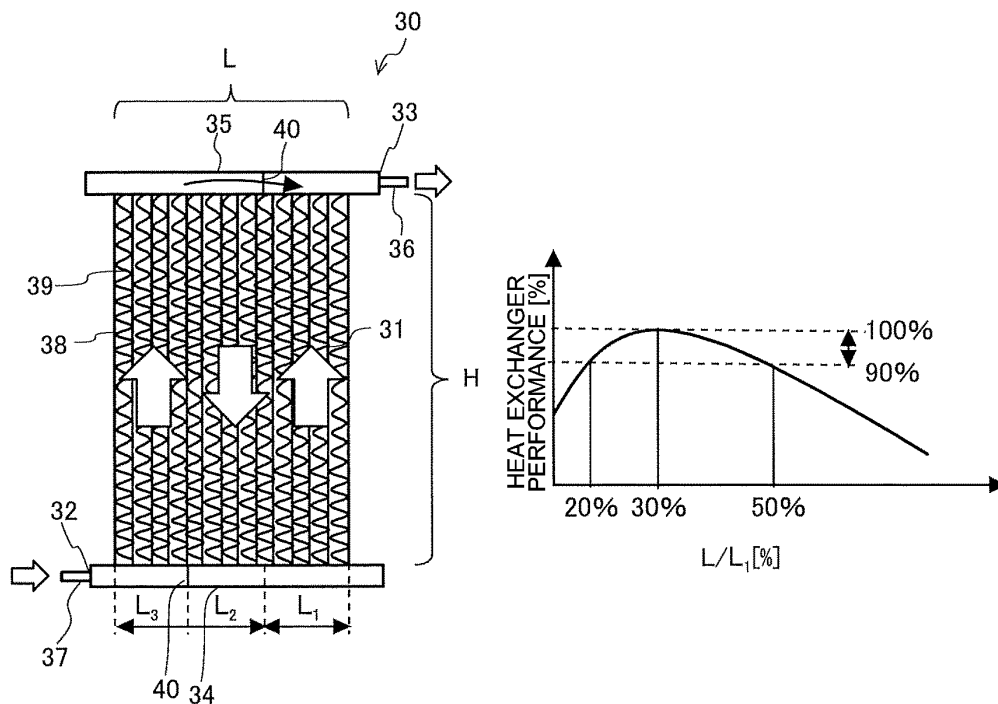


FIG. 16

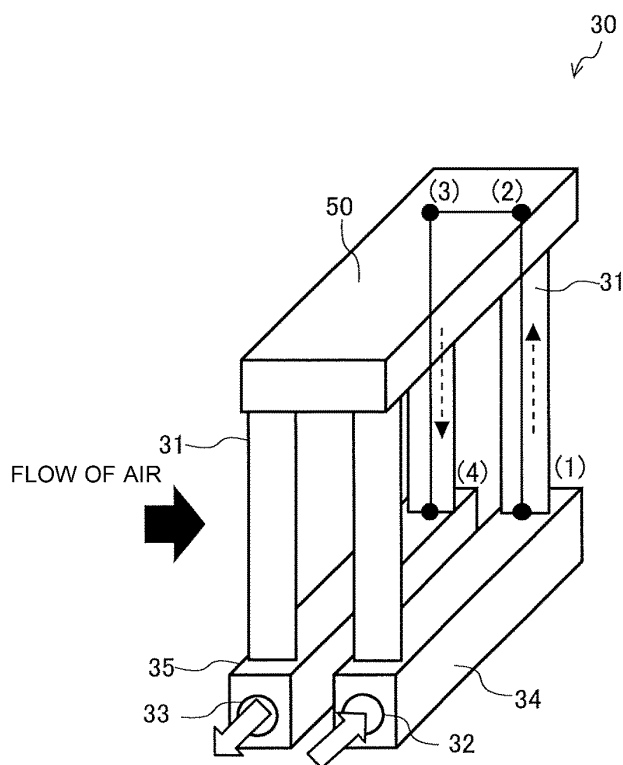


FIG. 17

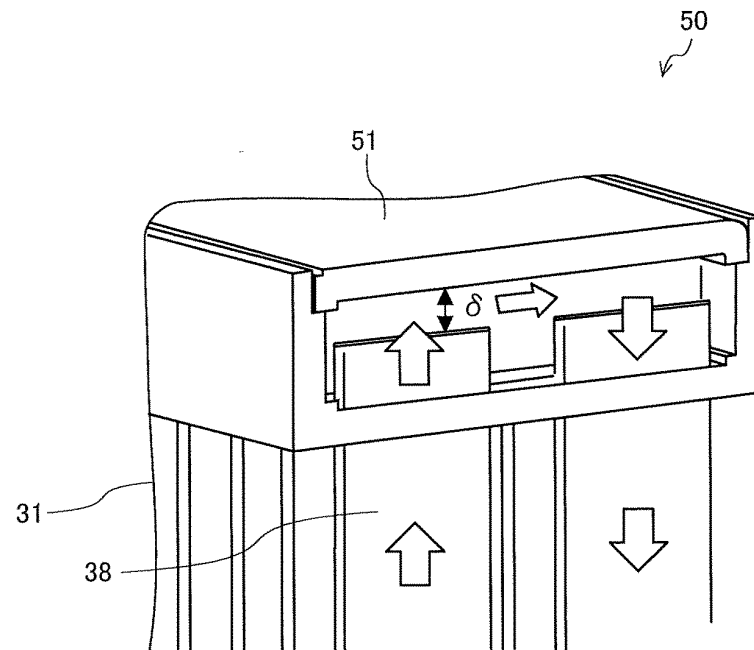


FIG. 18

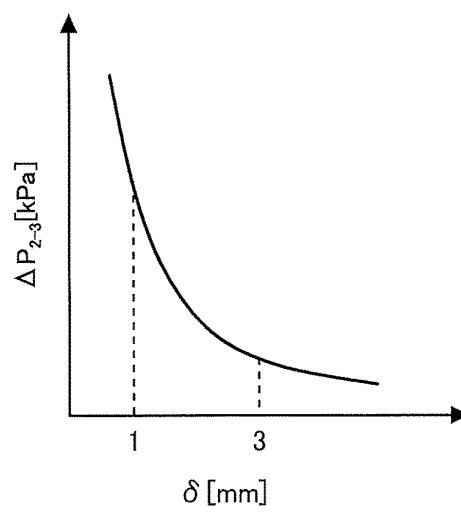


FIG. 19

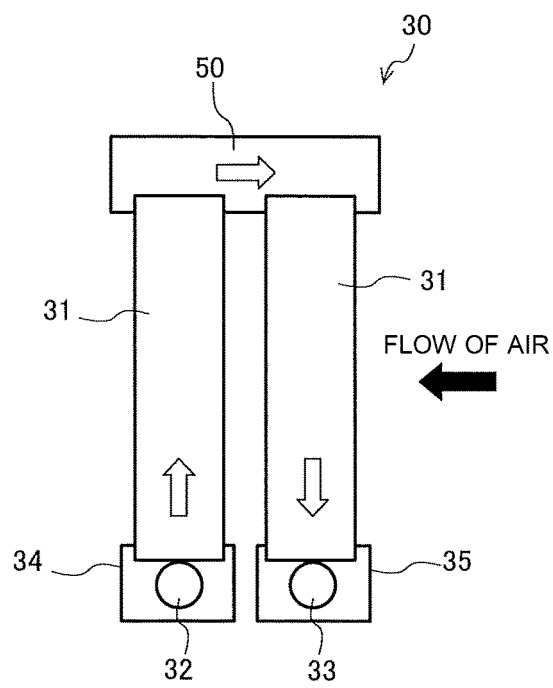


FIG. 20

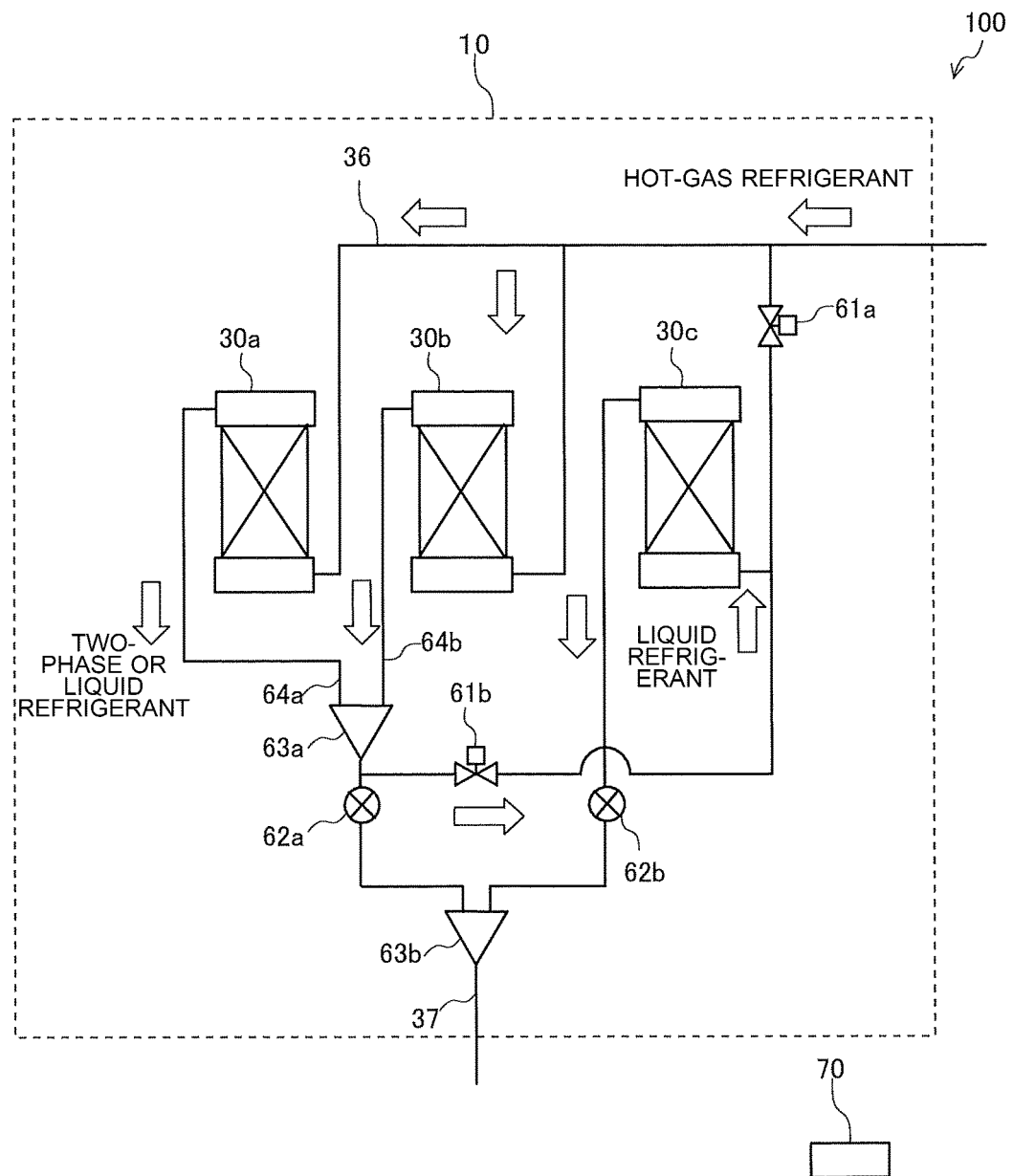


FIG. 21

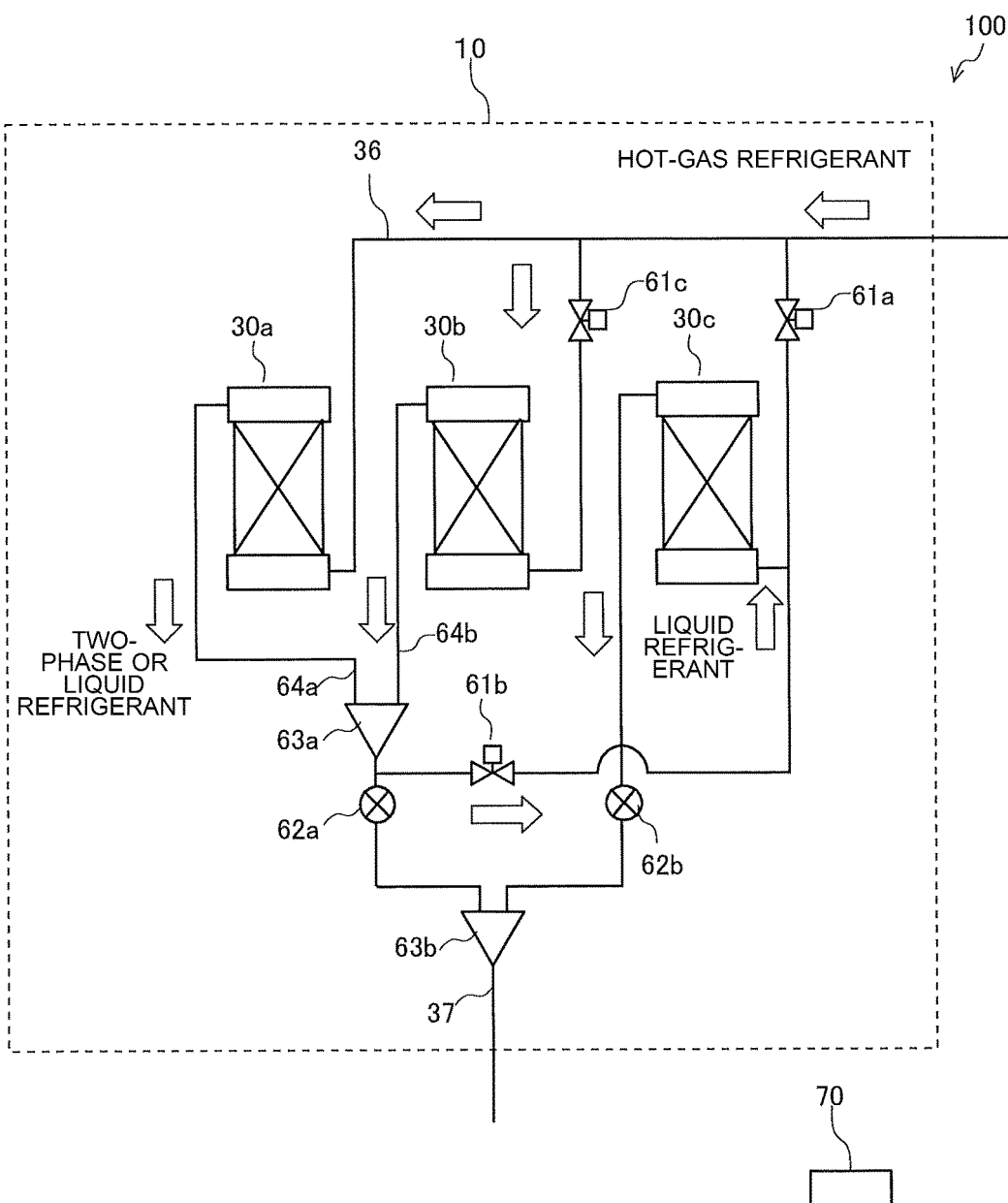


FIG. 22

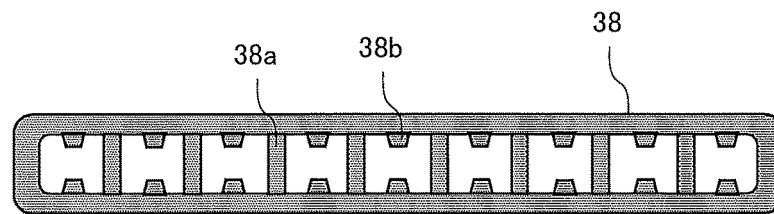


FIG. 23

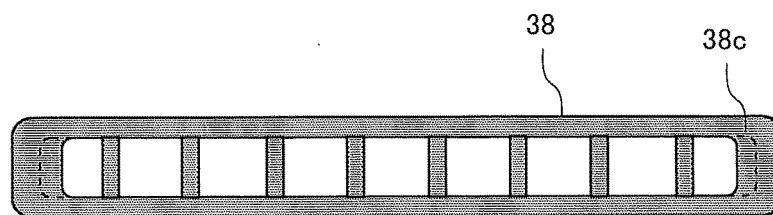


FIG. 24

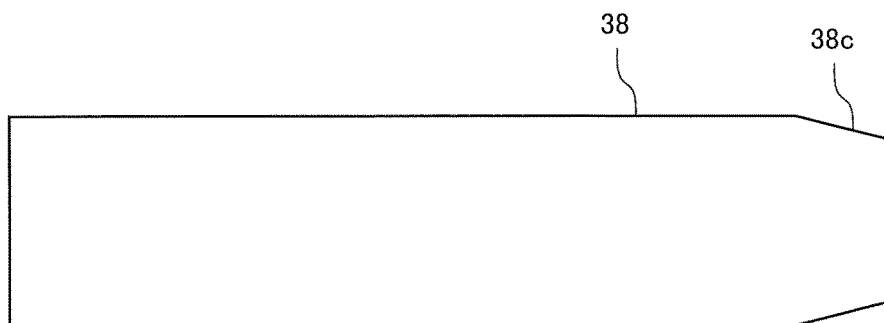


FIG. 25

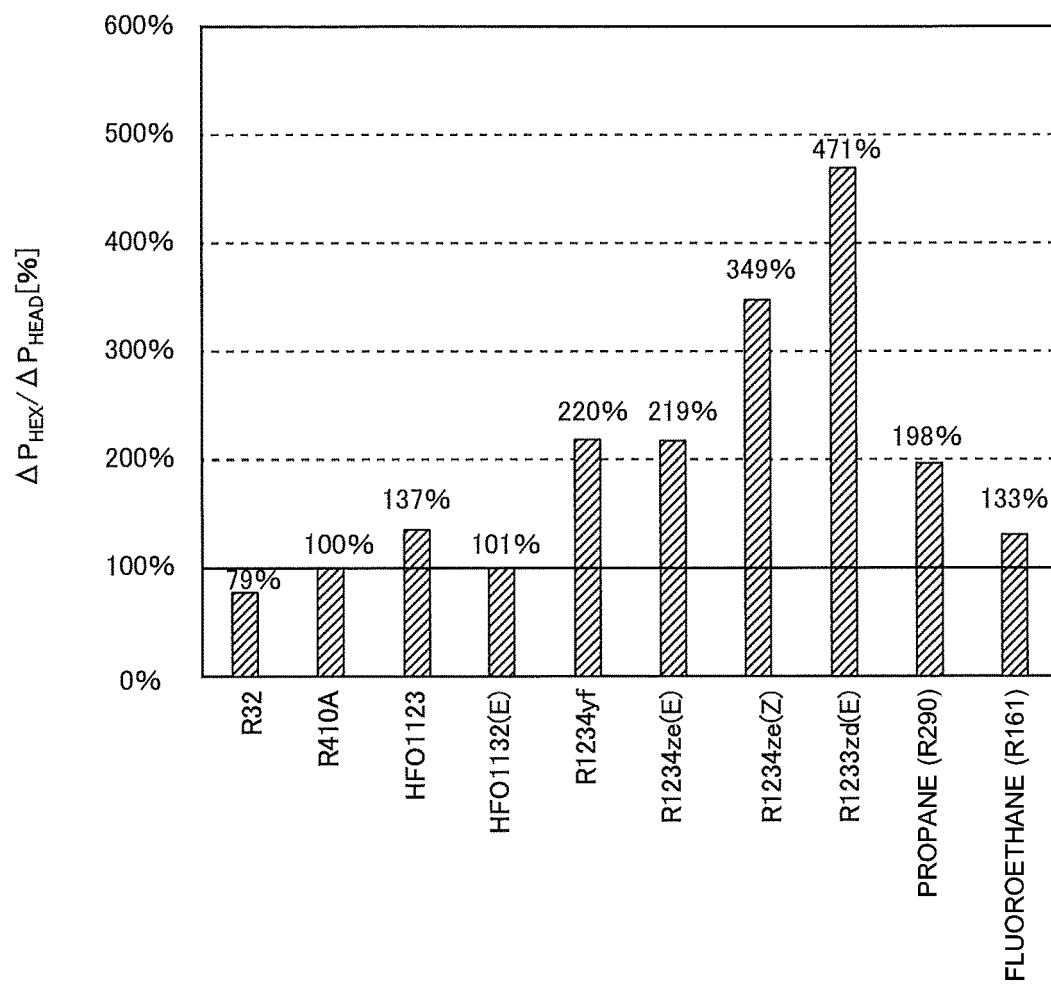


FIG. 26

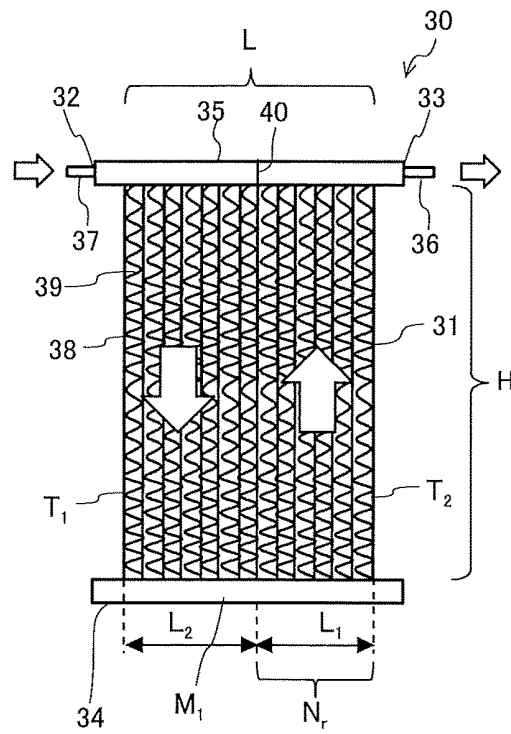
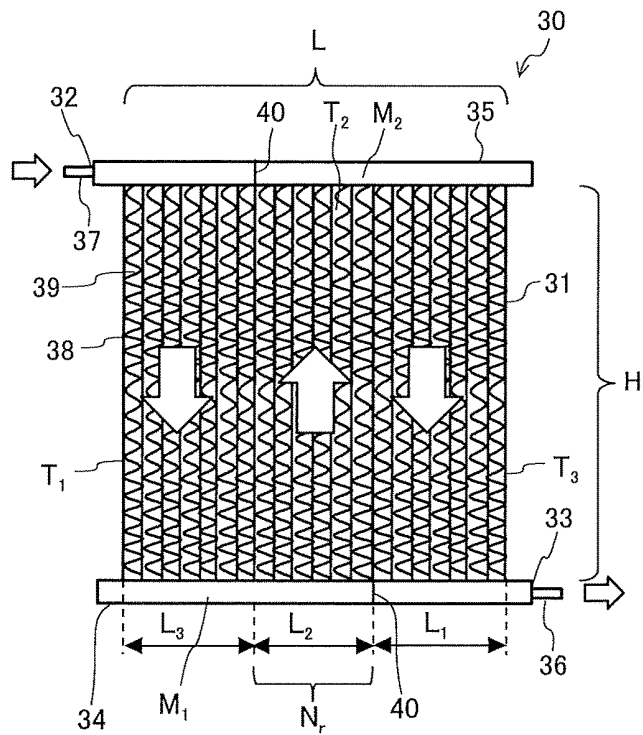


FIG. 27



## INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2021/020313

## A. CLASSIFICATION OF SUBJECT MATTER

**F25B 39/00**(2006.01)i; **F25B 39/04**(2006.01)i; **F28D 1/053**(2006.01)i; **F28F 1/02**(2006.01)i; **F28F 17/00**(2006.01)i;  
**F25B 47/02**(2006.01)i

FI: F28F17/00 501A; F28D1/053 A; F28F1/02 A; F25B47/02; F25B39/00; F25B39/04

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)

F25B39/00; F25B39/04; F28D1/053; F28F1/02; F28F17/00; F25B47/02

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 1971-2021  
 Registered utility model specifications of Japan 1996-2021  
 Published registered utility model applications of Japan 1994-2021

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. |
|-----------|--------------------------------------------------------------------------------------------------------------------------------------------------------|-----------------------|
| A         | JP 2013-174398 A (JAPAN CLIMATE SYSTEMS CORP.) 05 September 2013 (2013-09-05)<br>claim 1, paragraph [0009], fig. 4                                     | 1-17                  |
| A         | JP 10-220919 A (CALSONIC CORP.) 21 August 1998 (1998-08-21)<br>claim 1, paragraph [0009], fig. 1, 2                                                    | 1-7                   |
| A         | WO 2016/092943 A1 (JOHNSON CONTROLS HITACHI AIR CONDITIONING TECHNOLOGY (HONGKONG) LTD.) 16 June 2016 (2016-06-16)<br>paragraphs [0081]-[0084], fig. 2 | 1-17                  |
| A         | JP 2006-183962 A (DENSO CORP.) 13 July 2006 (2006-07-13)<br>claim 1, paragraph [0008], fig. 1                                                          | 1-17                  |
| A         | WO 2021/079422 A1 (MITSUBISHI ELECTRIC CORP.) 29 April 2021 (2021-04-29)<br>paragraph [0040], fig. 6                                                   | 1-17                  |
| A         | JP 58-37465 A (NIPPONDENSO CO., LTD.) 04 March 1983 (1983-03-04)<br>pp. 2, 3, fig. 1-3                                                                 | 1-17                  |

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

\* Special categories of cited documents:

"A" document defining the general state of the art which is not considered to be of particular relevance

"E" earlier application or patent but published on or after the international filing date

"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

"&" document member of the same patent family

Date of the actual completion of the international search

**22 June 2021**

Date of mailing of the international search report

**06 July 2021**

Name and mailing address of the ISA/JP

**Japan Patent Office (ISA/JP)  
 3-4-3 Kasumigaseki, Chiyoda-ku, Tokyo 100-8915  
 Japan**

Authorized officer

Telephone No.

INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2021/020313

C. DOCUMENTS CONSIDERED TO BE RELEVANT

| Category* | Citation of document, with indication, where appropriate, of the relevant passages        | Relevant to claim No. |
|-----------|-------------------------------------------------------------------------------------------|-----------------------|
| A         | JP 2008-267686 A (DENSO CORP.) 06 November 2008 (2008-11-06)<br>paragraphs [0011], [0012] | 1-17                  |

Form PCT/ISA/210 (second sheet) (January 2015)

**INTERNATIONAL SEARCH REPORT**  
**Information on patent family members**

International application No.

**PCT/JP2021/020313**

| Patent document<br>cited in search report | Publication date<br>(day/month/year) | Patent family member(s)                                                                    | Publication date<br>(day/month/year) |
|-------------------------------------------|--------------------------------------|--------------------------------------------------------------------------------------------|--------------------------------------|
| JP 2013-174398 A                          | 05 September 2013                    | (Family: none)                                                                             |                                      |
| JP 10-220919 A                            | 21 August 1998                       | US 6302193 B1<br>claim 1, fig. 1, 2<br>US 2002/0023736 A1<br>EP 851188 A2<br>EP 1223391 A1 |                                      |
| WO 2016/092943 A1                         | 16 June 2016                         | US 2017/0268790 A1<br>paragraphs [0095]-[0098], fig. 2<br>EP 3232139 A1<br>CN 107003048 A  |                                      |
| JP 2006-183962 A                          | 13 July 2006                         | US 2006/0185386 A1<br>claim 1, fig. 1<br>DE 102005062423 A1<br>CN 1796904 A                |                                      |
| WO 2021/079422 A1                         | 29 April 2021                        | (Family: none)                                                                             |                                      |
| JP 58-37465 A                             | 04 March 1983                        | (Family: none)                                                                             |                                      |
| JP 2008-267686 A                          | 06 November 2008                     | US 2008/0271479 A1<br>paragraphs [0005]-[0008]<br>EP 1992890 A1                            |                                      |

Form PCT/ISA/210 (patent family annex) (January 2015)

**REFERENCES CITED IN THE DESCRIPTION**

*This list of references cited by the applicant is for the reader's convenience only. It does not form part of the European patent document. Even though great care has been taken in compiling the references, errors or omissions cannot be excluded and the EPO disclaims all liability in this regard.*

**Patent documents cited in the description**

- JP 2018096638 A [0004]