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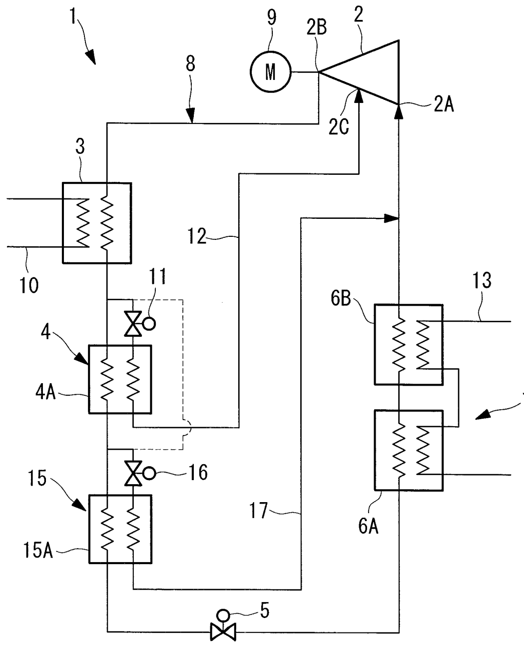
(54) REFRIGERATION DEVICE

(57) A refrigerator is provided in which a refrigerant supplied to an evaporator is precooled to a dryness of nearly zero and is supplied in a single liquid phase to increase the amount of heat exchanged by the evaporator, thereby improving cooling performance or reducing the size of the evaporator. A refrigerator (1) has a refrigeration cycle (8) formed by sequentially connecting a compressor (2) that compresses a refrigerant, a condenser (3) that condenses the high-pressure gas refrigerant, an economizer (4) that evaporates some of the condensed liquid refrigerant to cool the liquid refrigerant by means of the latent heat of evaporation thereof and that has a circuit for injecting the evaporated medium-pressure refrigerant into an intermediate inlet of the compressor, an expansion valve (5) that adiabatically expands the liquid refrigerant, and an evaporator (7) that evaporates the adiabatically expanded refrigerant, and a refrigerant precooler (15) that precools the refrigerant supplied to the evaporator (7) is disposed between the economizer (4) and the evaporator (7).

FIG. 1



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Description

Technical Field

[0001] The present invention relates to refrigerators, and particularly to a refrigerator suitable for application to a turbo refrigerator using a plate heat exchanger as an evaporator.

Background Art

[0002] Turbo refrigerators, conventionally used as high-capacity heat source systems, use shell-and-tube heat exchangers suitable for exchange of large amounts of heat as condensers and evaporators. Recently, however, dramatic advances in manufacturing technology have enabled the manufacture of turbo refrigerators with relatively low capacities, namely, less than 100 tons of refrigeration. Such low-capacity turbo refrigerators use plate heat exchangers in place of shell-and-tube heat exchangers. On the other hand, turbo refrigerators have high-efficiency performance characteristics and accordingly require the plate heat exchangers that are used to have large-size, high-performance specifications.

[0003] A typical plate heat exchanger has a structure in which a plurality of plates are stacked in parallel such that a plurality of refrigerant channels and a plurality of cooled-medium channels are alternately arranged therebetween; therefore, a major challenge that is faced when used as an evaporator is how to evenly distribute a refrigerant in a vapor-liquid two-phase state among the plurality of refrigerant channels at the entrance of the evaporator. Specifically, because the vapor-liquid two-phase refrigerant contains a large volume of vapor-phase refrigerant, an unbalanced flow due to the difference in pressure loss between the individual channels causes the liquid-phase refrigerant to be distributed in an unbalanced manner among the plurality of refrigerant channels and therefore results in an uneven distribution of the liquid-phase refrigerant, thus posing a problem in that its heat exchange performance (cooling performance) is decreased because of ineffective utilization of the heat transfer area.

[0004] Patent Document 1 proposes a refrigerator in which a nozzle and orifices are provided at a refrigerant entrance of a plate heat exchanger to evenly distribute a refrigerant among a plurality of refrigerant channels by alleviating the difference in pressure loss, thus effectively utilizing the entire heat transfer surface of the heat exchanger for improved cooling capacity. Also, to prevent a drop in efficiency due to pressure losses at the orifices in the case where plate heat exchangers are arranged in series as multiple stages to increase the amount of heat exchanged, Patent Document 2 proposes a refrigerator having an orifice mechanism, namely, through-holes, only at the front-end plate heat exchanger and a vapor-liquid separator disposed in piping connecting the plurality of plate heat exchangers so that a gas refrigerant separated by the vapor-liquid separator is returned to the downstream side of the back-end plate heat exchanger.

[0005]

Patent Document 1: Japanese Unexamined Patent Application, Publication No. 2001-165590
Patent Document 2: Japanese Unexamined Patent Application, Publication No. 2005-337688

Disclosure of Invention

[0006] The refrigerators disclosed in Patent Documents 1 and 2 above, however, are the same in that both include a refrigerant distributor having an orifice mechanism at a refrigerant entrance of a plate heat exchanger to evenly distribute a refrigerant in a vapor-liquid two-phase state among a plurality of refrigerant channels. Hence, both share a problem in that a drop in efficiency due to a pressure loss at the orifice mechanism is unavoidable and that the plate heat exchanger has a complicated structure and is expensive.

[0007] In a refrigeration cycle, a refrigerant at an entrance of an evaporator is normally in a vapor-liquid two-phase state and has a relatively low dryness, namely, about 0.1. Nevertheless, the vapor-phase refrigerant accounts for a predominantly large volume and, as described above, makes it difficult to evenly distribute the liquid-phase refrigerant among a plurality of refrigerant channels, thus constituting the underlying cause of the above problem. Accordingly, to improve the heat exchange efficiency of an evaporator for size reduction and improved performance, the challenge, which is not limited to the case where a plate heat exchanger is used, lies in how to bring the state of the refrigerant at the entrance of the evaporator closer to a single liquid phase.

[0008] An object of the present invention, which has been made in light of such circumstances, is to provide a refrigerator in which a refrigerant supplied to an evaporator can be precooled to a dryness of nearly zero and be supplied in a single liquid phase to increase the amount of heat exchanged by the evaporator, thereby improving cooling performance or reducing the size of the evaporator.

[0009] To solve the above problem, a refrigerator of the present invention employs the following solutions.
That is, a first aspect of a refrigerator according to the present invention is a refrigerator having a refrigeration cycle formed by sequentially connecting a compressor that compresses a refrigerant, a condenser that condenses the high-pressure gas refrigerant, an economizer that evaporates some of the condensed liquid refrigerant to cool the liquid refrigerant by means of the latent heat of evaporation thereof and that has a circuit for injecting the evaporated medium-pressure refrigerant into an intermediate inlet of the compressor, an expansion valve that adiabatically expands the liquid refrigerant, and an evaporator that evaporates the adiabatically expanded refrigerant, and a refrigerant precooler that precools the re-

frigerant supplied to the evaporator is disposed between the economizer and the evaporator.

[0010] According to the first aspect, the refrigerant precooler disposed between the economizer and the evaporator can precool the refrigerant supplied to the evaporator to a dryness of nearly zero to supply the refrigerant in a liquid phase to the evaporator. As a result, the temperature of the liquid refrigerant can be decreased at the same pressure to achieve a larger temperature difference between the liquid refrigerant and a cooled medium cooled by the evaporator. This ensures improvement in the refrigeration capacity and COP (coefficient of performance) by the economizer effect and allows a larger amount of heat to be exchanged at the same heat transfer coefficient, thus improving cooling performance or reducing the size of the evaporator.

[0011] In the refrigerator of the first aspect, additionally, the refrigerant precooler may evaporate some of the liquid refrigerant to cool the liquid refrigerant by means of the latent heat of evaporation thereof and may have a circuit for returning the evaporated refrigerant to a refrigerant intake circuit between the evaporator and the compressor.

[0012] According to the first aspect, because the refrigerant precooler uses some of the liquid refrigerant circulated through the refrigeration cycle as a heat sink to precool the refrigerant by means of the latent heat of evaporation thereof, it is possible to efficiently precool the liquid refrigerant and also to simplify the structure of the refrigerant precooler for ease of installation without the need to supply an external heat sink.

[0013] In the refrigerator of the first aspect, additionally, the refrigerant precooler may be constituted of a refrigerant-refrigerant heat exchanger that precools the liquid refrigerant by heat exchange with a refrigerant shunted from the liquid refrigerant and depressurized and that has a circuit for returning the evaporated refrigerant to a refrigerant intake circuit between the evaporator and the compressor.

[0014] In the above structure, because the refrigerant precooler is constituted of the refrigerant-refrigerant heat exchanger that performs refrigerant-refrigerant heat exchange and that has the circuit for returning the evaporated refrigerant to the refrigerant intake circuit between the evaporator and the compressor, the refrigerant precooler used needs no special structure, and an existing refrigerant-refrigerant heat exchanger can be directly applied. Accordingly, the refrigerant precooler can be provided at low cost.

[0015] In the refrigerator having the above structure, additionally, the economizer may be constituted of an intermediate cooler that evaporates some of the condensed liquid refrigerant to cool the liquid refrigerant by means of the latent heat of evaporation thereof, and the refrigerant may be a mixed refrigerant such as R410A.

[0016] In the above structure, because the economizer is constituted of the intermediate cooler that performs refrigerant-refrigerant heat exchange and the refrigerant precooler is constituted of the refrigerant-refrigerant heat exchanger, the economizer and the refrigerant precooler do not change the composition of the refrigerant even if the refrigeration cycle uses a mixed refrigerant, such as R410A, whose composition changes as a result of self-expansion. Accordingly, the rated capacity can be delivered without the possibility of unstable capacity due to changes in the composition of the refrigerant.

[0017] In the refrigerator of the first aspect, additionally, the refrigerant precooler may be constituted of a vapor-liquid separator that separates the liquid refrigerant into a liquid-phase refrigerant and a vapor-phase refrigerant and that has a circuit for returning the vapor-phase refrigerant having precooled the liquid-phase refrigerant by evaporation and separation to a refrigerant intake circuit between the evaporator and the compressor.

[0018] According to the first aspect, because the refrigerant precooler is constituted of the vapor-liquid separator that separates the liquid refrigerant into a liquid-phase refrigerant and a vapor-phase refrigerant and that has the circuit for returning the vapor-phase refrigerant having precooled the liquid-phase refrigerant by evaporation and separation to the refrigerant intake circuit between the evaporator and the compressor, the refrigerant precooler used needs no special structure, and an existing vapor-liquid separator can be directly employed. Accordingly, the refrigerant precooler can be provided at low cost.

[0019] In the refrigerator of the first aspect, additionally, the evaporator may be constituted of a plate heat exchanger including a plurality of plates stacked in parallel such that a plurality of refrigerant channels and a plurality of cooled-medium channels are alternately arranged.

[0020] In the above structure, because the refrigerant can be precooled to a dryness of nearly zero and be supplied to the evaporator in a liquid phase, even if the plate heat exchanger having the plurality of refrigerant channels is used for the evaporator, the liquid refrigerant can be evenly distributed among the plurality of refrigerant channels without using a distributor. As a result, a uniform liquid refrigerant distribution can be formed in the individual refrigerant channels to increase the effective heat transfer area, thus improving heat exchange performance (cooling performance). This simplifies the structure of the plate heat exchanger without the need for a refrigerant distributor and also reduces the size of the plate heat exchanger and improves the performance of the plate heat exchanger.

[0021] In the refrigerator having the above structure, additionally, the evaporator may be constituted of a plurality of the plate heat exchangers connected in series as multiple stages.

[0022] In the above structure, because the plurality of plate heat exchangers are connected in series as multiple stages, the amount of heat exchanged by the evaporator (cooling capacity) can be increased. This improves the cooling performance.

[0023] In the refrigerator having the above structure, additionally, the refrigerant precoolers constituted of the vapor-liquid separators may be arranged in series as multiple stages at individual entrances of the plurality of plate heat exchangers.

[0024] In the above structure, because the refrigerant precoolers constituted of the vapor-liquid separators are arranged in series as multiple stages at the individual entrances of the plurality of plate heat exchangers connected in series as multiple stages, only a liquid-phase refrigerant can be supplied from the refrigerant precoolers to the respective plate heat exchangers. This allows the liquid refrigerant to be evenly distributed among the individual refrigerant channels of the plurality of plate heat exchangers to improve the heat exchange performance (cooling performance) and also reduces the size of the plate heat exchangers to a compact size.

[0025] In addition, a second aspect of the refrigerator according to the present invention is a refrigerator having a heat pump cycle formed by sequentially connecting a compressor that compresses a refrigerant, a switching valve that switches a refrigerant cycle, a heat-source-side heat exchanger, an expansion valve that adiabatically expands the refrigerant, and a utilization-side heat exchanger. An economizer through which a high-pressure liquid refrigerant always flows in one direction via a refrigerant-flow-direction switching valve, which evaporates some of the high-pressure liquid refrigerant to supercool the refrigerant, and which has a circuit for injecting the evaporated medium-pressure refrigerant into an intermediate inlet of the compressor is disposed between the heat-source-side heat exchanger and the utilization-side heat exchanger, and a refrigerant precooler that precools the refrigerant supplied to the utilization-side heat exchanger or the heat-source-side heat exchanger functioning as an evaporator is disposed downstream of the economizer.

[0026] According to the second aspect, in switching between cooling and heating, the liquid refrigerant supercooled by the economizer can be supplied via the refrigerant-flow-direction switching valve to the utilization-side heat exchanger functioning as an evaporator in cooling or to the heat-source-side heat exchanger functioning as an evaporator in heating, and the medium-pressure refrigerant evaporated by the economizer can be injected into the intermediate inlet of the compressor. This improves the cooling/heating capacity and COP (coefficient of performance). At the same time, because the refrigerant precooler disposed downstream of the economizer precools the refrigerant supplied to the utilization-side heat exchanger or the heat-source-side heat exchanger functioning as an evaporator in cooling or heating so that the refrigerant can be supplied in a liquid phase with a dryness of nearly zero, the temperature of the liquid refrigerant can be decreased at the same pressure to achieve a larger temperature difference between the liquid refrigerant and a heat exchange medium subjected to heat exchange on the evaporator side. This allows a larger amount of heat to be exchanged at the same heat transfer coefficient, thus improving the heat exchange performance or reducing the size of the heat exchangers themselves.

[0027] In one of the above aspects, additionally, the refrigerant precooler may decrease the dryness of the refrigerant to nearly zero at an entrance of the evaporator.

[0028] According to the above aspect, because the refrigerant precooler decreases the dryness of the refrigerant to nearly zero at the entrance of the evaporator, only a single-phase liquid refrigerant can be reliably supplied to the evaporator. As a result, the temperature of the liquid refrigerant can be decreased at the same pressure to achieve a larger temperature difference between the liquid refrigerant and the cooled medium cooled by the evaporator. This allows a larger amount of heat to be exchanged at the same heat transfer coefficient, thus improving the cooling performance or reducing the size of the evaporator.

[0029] In one of the above aspects, additionally, the refrigerator may be a turbo refrigerator using a turbo compressor as the compressor.

[0030] According to the above aspect, it is possible to improve the performance of a turbo refrigerator that has high-efficiency, high-performance characteristics and to reduce the size thereof.

[0031] According to the present invention, because the refrigerant supplied to the evaporator can be precooled to a dryness of nearly zero and be supplied in a liquid phase to the evaporator, the temperature of the liquid refrigerant can be decreased at the same pressure to achieve a larger temperature difference between the liquid refrigerant and the cooled medium cooled by the evaporator. This ensures the economizer effect and allows a larger amount of heat to be exchanged at the same heat transfer coefficient, thus improving the cooling performance or reducing the size of the evaporator.

Brief Description of Drawings

[0032]

[FIG. 1] Fig. 1 is a refrigeration cycle diagram of a turbo refrigerator according to a first embodiment of the present invention.

[FIG. 2] Fig. 2 is a P-h graph of the turbo refrigerator shown in Fig. 1.

[FIG. 3] Fig. 3 is a graph showing the relationship between the refrigerant dryness and the overall heat transfer U of the turbo refrigerator shown in Fig. 1.

[FIG. 4] Fig. 4 is a refrigeration cycle diagram of a turbo refrigerator according to a second embodiment of the present invention.

[FIG. 5] Fig. 5 is a refrigeration cycle diagram of a turbo refrigerator according to a third embodiment of the present invention.

[FIG. 6] Fig. 6 is a refrigeration cycle diagram of a turbo refrigerator according to a fourth embodiment of the present invention.

Explanation of Reference Signs:

[0033]

1:	turbo refrigerator
2:	two-stage turbo compressor
3:	condenser
3A:	heat-source-side air heat exchanger
4:	economizer
4A:	intermediate heat exchanger (intermediate cooler)
5:	main expansion valve
6A, 6B:	plate heat exchanger
7:	evaporator
7A:	utilization-side heat exchanger
8, 8A:	refrigeration cycle (heat pump cycle)
15, 25, 35, 36:	refrigerant precooler
15A:	refrigerant-refrigerant heat exchanger
16:	refrigerant-precooling expansion valve
17, 26, 37, 39:	gas circuit
20A, 20B:	four-way switching valve
25A, 35A, 36A:	vapor-liquid separator

Best Mode for Carrying Out the Invention

[0034] Embodiments of the present invention will be described below with reference to the drawings.

First Embodiment

[0035] A first embodiment of the present invention will be described below using Figs. 1 to 3.
Fig. 1 shows a refrigeration cycle diagram of a turbo refrigerator according to the first embodiment of the present invention. A turbo refrigerator 1 has a refrigeration cycle 8 formed as a closed circuit by sequentially connecting a two-stage turbo compressor 2, a condenser 3, an economizer 4, a main expansion valve 5, and an evaporator 7 including two plate heat exchangers 6A and 6B connected in series as multiple stages.

[0036] The two-stage turbo compressor 2, a multistage compressor driven by an inverter motor 9, has an intermediate inlet 2C disposed between first and second impellers (not shown) in addition to an inlet 2A and an outlet 2B and is configured to sequentially compress a low-pressure refrigerant gas taken in from the inlet 2A by centrifugation through rotation of the first and second impellers and to discharge the compressed high-pressure refrigerant gas from the outlet 2B. The condenser 3 condenses the high-pressure refrigerant gas supplied from the two-stage turbo compressor 2 by heat exchange with cooling water circulated via a cooling-water circuit 10.

[0037] The economizer 4 is constituted of an intermediate cooler 4A formed of a refrigerant-refrigerant heat exchanger, such as a double-pipe heat exchanger, that performs heat exchange between the liquid refrigerant flowing through the main circuit of the refrigeration cycle 8 and a refrigerant shunted from the main circuit and depressurized by an economizer expansion valve 11 to supercool the liquid refrigerant flowing through the main circuit by means of the latent heat of evaporation of the refrigerant. In addition, the intermediate cooler 4A has a gas circuit 12 for injecting the refrigerant gas evaporated when supercooling the liquid refrigerant through the intermediate inlet 2C of the two-stage turbo compressor 2 into a medium-pressure compressed refrigerant, thus constituting an intermediate-cooler economizer cycle.

[0038] The main expansion valve 5 adiabatically expands the refrigerant supercooled through the economizer 4 and supplies it to the evaporator 7. The evaporator 7 is constituted of the plate heat exchangers 6A and 6B connected in series as multiple stages, each constituted of a plurality of plates stacked in parallel such that a plurality of refrigerant channels and a plurality of cooled-medium channels (cold water channels) are alternately arranged, and the evaporator 7 evaporates the refrigerant by heat exchange with cold water circulated through the cooled-medium channels (cold water channels) via a cold-water circuit 13 to cool the cold water to a preset temperature, for example, 7°C, by means of the latent heat of evaporation thereof. The refrigerant and the cold water preferably flow in counterflow.

[0039] In addition to the above structure, in this embodiment, a refrigerant precooler 15 is further disposed downstream of the economizer 4 to precool the refrigerant supplied to the evaporator 7 to a dryness of nearly zero. This refrigerant precooler 15 is constituted of a refrigerant-refrigerant heat exchanger 15A, such as a double-pipe heat exchanger, having nearly the same structure as the above intermediate cooler 4A for the economizer 4 and performs heat exchange between the liquid refrigerant flowing through the main circuit of the refrigeration cycle 8 and a refrigerant shunted from the main circuit downstream of the economizer 4 and depressurized by a refrigerant-precooling expansion valve 16 to cool the liquid refrigerant flowing through the main circuit by means of the latent heat of evaporation of the refrigerant. In addition, the refrigerant precooler 15 has a gas circuit 17 for returning the refrigerant gas evaporated when cooling the liquid refrigerant to a refrigerant intake circuit between the evaporator 7 and the two-stage turbo compressor 2.

[0040] Next, the operation of this embodiment will be described with reference to a P-h graph shown in Fig. 2. A low-temperature, low-pressure refrigerant gas A taken in from the inlet 2A of the two-stage turbo compressor 2 is compressed from point A to point B by the first impeller, is mixed with the medium-pressure refrigerant gas injected from the intermediate inlet 2C to reach point C, and is taken in through and compressed to point D by the second impeller.
The refrigerant discharged in this state from the two-stage turbo compressor 2 is cooled and condensed into a high-pressure liquid refrigerant at point E by the con-

denser 3. Some of the liquid refrigerant at point E is shunted and depressurized to point F by the economizer expansion valve 11 to flow into the intermediate cooler 4A. This medium-pressure refrigerant is subjected, in the intermediate cooler 4A, to heat exchange with the liquid refrigerant E flowing through the main circuit of the refrigeration cycle 8 to absorb heat from the liquid refrigerant E, thus evaporating, and is then injected via the gas circuit 12 through the intermediate inlet 2C of the two-stage turbo compressor 2 into the medium-pressure refrigerant gas being compressed.

[0041] On the other hand, the liquid refrigerant E in the main circuit subjected to heat exchange with the refrigerant at point F in the intermediate cooler 4A for the economizer 4 is supercooled to point G and reaches the refrigerant precooler 15. Some of the liquid refrigerant exiting the intermediate cooler 4A is shunted and depressurized to point H by the refrigerant-precooling expansion valve 16 to flow into the refrigerant precooler 15 for heat exchange with the liquid refrigerant G in the main circuit. This refrigerant at point H is subjected, in the refrigerant precooler 15, to heat exchange with the liquid refrigerant G in the main circuit, thus evaporating, and is then returned via the gas circuit 17 to the refrigerant intake circuit between the evaporator 7 and the two-stage turbo compressor 2 to meet the refrigerant A exiting the evaporator 7 through point I.

[0042] The liquid refrigerant at point G is cooled to point J by precooling in the refrigerant precooler 15, is depressurized to point K by the main expansion valve 5, and reaches the entrance of the evaporator 7. The low-pressure refrigerant at point K, as shown in Fig. 2, is a single-phase liquid refrigerant with a dryness of nearly zero. Thus, the refrigerant precooler 15 disposed between the economizer 4 and the evaporator 7 can further precool the refrigerant supercooled by the economizer 4 to supply a single-phase liquid refrigerant with a dryness of nearly zero to the evaporator 7.

[0043] The refrigerant supplied to the evaporator 7 in a single liquid phase is first evenly distributed among the plurality of refrigerant channels of the front-end plate heat exchanger 6A and flows therethrough while being subjected to heat exchange with the cold water circulated through the cooled-medium channels (cold water channels) via the cold-water circuit 13 so that some refrigerant evaporates. The refrigerant flowing out of the front-end plate heat exchanger 6A then flows into the back-end plate heat exchanger 6B and is similarly subjected to heat exchange with the cold water so that the remaining refrigerant evaporates. Thus, the cold water circulated via the cold-water circuit 13 is cooled to a preset temperature and is supplied to the load side. The refrigerant flowing through the plate heat exchangers 6A and 6B, which turns into a slightly superheated low-pressure gas refrigerant A at the exit thereof, meets the gas refrigerant from the gas circuit 17 and is taken into the two-stage turbo compressor 2 again, with the subsequent operation being the same as above.

[0044] Thus, this embodiment provides the following advantages.

Because the refrigerant can be supplied to the evaporator 7 in a single liquid phase with a dryness of nearly zero, the temperature of the liquid refrigerant can be decreased at the same pressure to achieve a larger temperature difference between the liquid refrigerant and the cooled medium (cold water) cooled by the evaporator 7. This ensures improvement in the refrigeration capacity and COP (coefficient of performance) by the economizer 4 and allows a larger amount of heat to be exchanged at the same heat transfer coefficient, thus improving the cooling performance or reducing the size of the evaporator 7.

[0045] Specifically, as shown in Fig. 3, the refrigerant supplied to the evaporator 7 (plate heat exchanger 6A) is normally in a vapor-liquid two-phase state and has a dryness of about 0.1 and an overall heat transfer U of A1 at the entrance thereof and an overall heat transfer U of B1 at the exit thereof. As in Patent Document 2 above, therefore, a vapor-liquid separator can be disposed between the front-end plate heat exchanger 6A and the back-end plate heat exchanger 6B to separate the vapor-phase refrigerant at the exit of the front-end plate heat exchanger 6A, thereby improving the overall heat transfer U at the exit to B2. Because the amount of heat Q exchanged by the evaporator 7 is represented by $Q = A * U * \Delta Tm$, where A is the heat transfer area and ΔTm is the volume-change temperature difference, the heat transfer area A can be reduced to reduce the size of the evaporator 7 if the overall heat transfer U is increased to increase the amount of heat Q exchanged. As in this embodiment, if the refrigerant precooler 15 is provided to precool the refrigerant supplied to the evaporator 7 so that the refrigerant dryness at the evaporator entrance is decreased to nearly zero and accordingly the overall heat transfer U is increased to A2, it is possible to improve the cooling performance or to reduce the size of the evaporator 7 more effectively than in the case of the refrigerator disclosed in Patent Document 2.

[0046] In addition, because the refrigerant precooler 15 uses some of the liquid refrigerant circulated through the refrigeration cycle 8 as a heat sink to precool the liquid refrigerant by means of the latent heat of evaporation thereof, it is possible to efficiently precool the liquid refrigerant and also to simplify the structure of the refrigerant precooler 15 for ease of installation without the need to supply an external heat sink.

In addition, because the refrigerant precooler 15 is constituted of the refrigerant-refrigerant heat exchanger 15A, such as a double-pipe heat exchanger, that performs refrigerant-refrigerant heat exchange and that has the gas circuit 17 for returning the evaporated refrigerant to the refrigerant intake circuit between the evaporator 7 and the two-stage turbo compressor 2, the refrigerant precooler 15 needs no special structure, and an existing refrigerant-refrigerant heat exchanger can be directly applied. Accordingly, the refrigerant precooler 15 can be provided at low cost.

[0047] In addition, because the economizer 4 and the refrigerant precooler 15 are constituted of refrigerant-refrigerant heat exchangers, such as double-pipe heat exchangers, that perform refrigerant-refrigerant heat exchange, the economizer 4 and the refrigerant precooler 15 do not change the composition of the refrigerant even if the refrigeration cycle 8 uses a mixed refrigerant, such as R410A, whose composition changes as a result of self-expansion, so that the rated capacity can be delivered without the possibility of unstable capacity due to changes in the composition of the refrigerant.

[0048] In addition, because the refrigerant precooler 15 can precool the refrigerant to a dryness of nearly zero and supply it to the evaporator 7 in a single liquid phase, even if the plate heat exchangers 6A and 6B having the plurality of refrigerant channels are used for the evaporator 7, the liquid refrigerant can be evenly distributed among the plurality of refrigerant channels without using a distributor. This allows formation of a uniform liquid refrigerant distribution in the individual refrigerant channels to increase the effective heat transfer area, thus improving the heat exchange performance (cooling performance), and also simplifies the structure of the plate heat exchangers 6A and 6B. In particular, the heat exchange efficiency can be increased because an orifice mechanism can be omitted for reduced pressure loss. In addition, because the evaporator 7 can be constituted by connecting the plurality of plate heat exchangers 6A and 6B in series as multiple stages, the amount of heat exchanged by the evaporator 7 can be increased to improve the cooling performance.

[0049] In addition, because the superheated refrigerant gas evaporated by the refrigerant precooler 15 is returned to the refrigerant intake circuit between the evaporator 7 and the two-stage turbo compressor 2 via the gas circuit 17, even if some refrigerant droplets are carried over from the evaporator 7, they can be reliably evaporated. Thus, carry-over of refrigerant droplets to the two-stage turbo compressor 2 can be prevented.

In this embodiment, the circuit for supplying some of the liquid refrigerant to the refrigerant precooler 15 may be constituted of a circuit branched from the circuit for shunting some of the liquid refrigerant from the upstream side of the economizer 4 to the intermediate cooler 4A, as indicated by the broken line in Fig. 1.

Second Embodiment

[0050] Next, a second embodiment of the present invention will be described using Fig. 4.

This embodiment differs from the first embodiment described above in the structure of a refrigerant precooler 25. The other points are similar to those of the first embodiment, and a description thereof will therefore be omitted.

In this embodiment, the refrigerant precooler 25 is constituted of a vapor-liquid separator 25A disposed on the entrance side of the evaporator 7 (plate heat exchanger 6A). A vapor-phase refrigerant separated by the vapor-liquid separator 25A is returned to the refrigerant intake circuit between the evaporator 7 and the two-stage turbo compressor 2 via a gas circuit 26 having an on/off valve 27.

[0051] As described above, because the refrigerant precooler 25 constituted of the vapor-liquid separator 25A disposed on the entrance side of the evaporator 7 (plate heat exchanger 6A) can supply a single liquid phase with a dryness of nearly zero to the evaporator 7 (plate heat exchanger 6A), the same effects and advantages as the first embodiment described above can be provided. In addition, the vapor-liquid separator 25A needs no special structure, and existing vapor-liquid separators widely used for refrigerators can be directly applied, so that the refrigerant precooler 25 can be provided at low cost.

This embodiment illustrates the case where the single plate heat exchanger 6A is provided as the evaporator 7; naturally, a plurality of plate heat exchangers may be connected in series in multiple stages, as in the first embodiment.

Third Embodiment

[0052] Next, a third embodiment of the present invention will be described using Fig. 5.

This embodiment differs from the first embodiment described above in the structure of refrigerant precoolers 35 and 36. The other points are similar to those of the first embodiment, and a description thereof will therefore be omitted.

In this embodiment, the evaporator 7 constituted of the plurality of plate heat exchangers 6A and 6B connected in series as multiple stages is provided with refrigerant precoolers 35 and 36 constituted of vapor-liquid separators 35A and 36A, respectively, arranged in series as multiple stages at the entrances of the respective plate heat exchangers 6A and 6B. In addition, vapor-phase refrigerants separated by the vapor-liquid separators 35A and 36A are returned to the refrigerant intake circuit between the evaporator 7 and the two-stage turbo compressor 2 via gas circuits 37 and 39 having on/off valves 38 and 40, respectively.

[0053] As described above, if the evaporator 7 is constituted of the plurality of plate heat exchangers 6A and 6B connected in series as multiple stages, the refrigerant precoolers 35 and 36 constituted of the vapor-liquid separators 35A and 36A can be arranged in series as multiple stages at the entrances of the respective plate heat exchangers 6A and 6B to supply only a single-phase liquid refrigerant with a dryness of nearly zero from the refrigerant precoolers 35 and 36 to the respective plate heat exchangers 6A and 6B. Thus, the same effects and advantages as the first embodiment described above can be provided. In addition, because the liquid refrigerant can be evenly distributed among the individual refrigerant channels of the plurality of plate heat exchangers 6A and 6B, it is possible to improve the heat exchange performance (cooling performance) and to reduce the size of the plate heat exchangers 6A and 6B to a compact size.

Fourth Embodiment

[0054] Next, a fourth embodiment of the present invention will be described using Fig. 6.
This embodiment differs from the first embodiment described above in that a four-way switching valve 20A for switching the refrigeration cycle and a four-way switching valve 20B for switching the refrigerant flow direction are provided to form a heat pump cycle so that the turbo refrigerator 1 can perform heating and cooling. The other points are similar to those of the first embodiment, and a description thereof will therefore be omitted.
The turbo refrigerator 1 of this embodiment includes the four-way switching valve 20A capable of reversing the refrigeration cycle between the discharge pipe and the intake pipe of the two-stage turbo compressor 2 to form a heat pump cycle 8A that can be switched between a cooling cycle and a heating cycle and also includes, instead of the water-cooled condenser 3, an air heat exchanger 3A equipped with a fin-and-tube refrigerant distributor 21 and capable of using air 10A as a heat source.
[0055] In addition, the four-way switching valve 20B capable of switching the refrigerant flow direction is disposed between the heat-source-side air heat exchanger 3A and a utilization-side heat exchanger 7A constituted of the plate heat exchangers 6A and 6B connected in series as multiple stages so that a high-pressure liquid refrigerant always flows in one direction through the economizer 4 and the refrigerant precooler 15 to achieve an economizer effect and a refrigerant-precooling effect in either of cooling and heating.
[0056] In the above structure, the four-way switching valves 20A and 20B can be switched to the direction indicated by the solid arrows so that the heat-source-side air heat exchanger 3A functions as a condenser and the utilization-side heat exchanger 7A functions as an evaporator, thereby supplying cold water from the utilization-side heat exchanger 7A to achieve cooling. On the other hand, the four-way switching valves 20A and 20B can be switched to the direction indicated by the dashed arrows so that the utilization-side heat exchanger 7A functions as a condenser and the heat-source-side air heat exchanger 3A functions as an evaporator, thereby supplying hot water from the utilization-side heat exchanger 7A to achieve heating. During the operation, the refrigerant flows in one direction through the economizer 4 and the refrigerant precooler 15 to provide an economizer effect and a refrigerant-precooling effect in either of cooling and heating, as in the above embodiments.
[0057] According to this embodiment, therefore, the liquid refrigerant supercooled by the economizer 4 can be supplied to the heat exchanger functioning as an evaporator in either of cooling and heating (the utilization-side heat exchanger 7A in cooling and the heat-source-side air heat exchanger 3A in heating), and the medium-pressure refrigerant evaporated by the economizer 4 can be injected into the intermediate inlet 2C of the two-stage turbo compressor 2. This improves the cooling/heating capacity and COP (coefficient of performance).
At the same time, because the refrigerant precooler 15 disposed downstream of the economizer 4 precools the refrigerant supplied to the utilization-side heat exchanger 7A or the heat-source-side air heat exchanger 3A functioning as an evaporator in cooling or heating so that the refrigerant can be supplied in a single liquid phase with a dryness of nearly zero, the temperature of the liquid refrigerant can be decreased at the same pressure to achieve a larger temperature difference between the liquid refrigerant and the heat exchange medium subjected to heat exchange on the evaporator side. This allows a larger amount of heat to be exchanged at the same heat transfer coefficient, thus improving the heat exchange performance or reducing the size of the heat exchangers themselves.
[0058] In this embodiment, the switching valves 20A and 20B for switching the refrigeration cycle and the refrigerant flow direction do not necessarily have to be four-way switching valves; for example, they can be replaced with bridge circuits composed of four electromagnetic on/off valves. In addition, the refrigerant precooler 15 can be constituted of the vapor-liquid separator 25A or 35A and 36A as in the second and third embodiments shown in Figs. 4 and 5.
[0059] In addition, the present invention is not limited to the invention according to the above embodiments; modifications are permitted where appropriate without departing from the spirit thereof. Naturally, the present invention can be similarly applied to, for example, a multistage-economizer turbo refrigerator constituted of a multistage turbo compressor including three or more stages. In addition, although an intermediate-cooler economizer cycle has been described as an example of an economizer cycle, the present invention can be similarly applied to a vapor-liquid-separator economizer cycle using a vapor-liquid separator. In addition, the evaporator used is not limited to a plate heat exchanger; naturally, another type of evaporator, such as a shell-and-tube heat exchanger or a fin-and-tube heat exchanger, can be used instead.

Claims

1. A refrigerator having a refrigeration cycle formed by sequentially connecting a compressor that compresses a refrigerant, a condenser that condenses the high-pressure gas refrigerant, an economizer that evaporates some of the condensed liquid refrigerant to cool the liquid refrigerant by means of the latent heat of evaporation thereof and that has a circuit for injecting the evaporated medium-pressure

refrigerant into an intermediate inlet of the compressor, an expansion valve that adiabatically expands the liquid refrigerant, and an evaporator that evaporates the adiabatically expanded refrigerant,
wherein a refrigerant precooler that precools the refrigerant supplied to the evaporator is disposed between the economizer and the evaporator.

2. The refrigerator according to Claim 1, wherein the refrigerant precooler evaporates some of the liquid refrigerant to cool the liquid refrigerant by means of the latent heat of evaporation thereof and has a circuit for returning the evaporated refrigerant to a refrigerant intake circuit between the evaporator and the compressor.

3. The refrigerator according to Claim 1 or 2, wherein the refrigerant precooler is constituted of a refrigerant-refrigerant heat exchanger that precools the liquid refrigerant by heat exchange with a refrigerant shunted from the liquid refrigerant and depressurized and that has a circuit for returning the evaporated refrigerant to a refrigerant intake circuit between the evaporator and the compressor.

4. The refrigerator according to Claim 3, wherein the economizer is constituted of an intermediate cooler that evaporates some of the condensed liquid refrigerant to cool the liquid refrigerant by means of the latent heat of evaporation thereof, and the refrigerant is a mixed refrigerant such as R410A.

5. The refrigerator according to Claim 1 or 2, wherein the refrigerant precooler is constituted of a vapor-liquid separator that separates the liquid refrigerant into a liquid-phase refrigerant and a vapor-phase refrigerant and that has a circuit for returning the vapor-phase refrigerant having precooled the liquid-phase refrigerant by evaporation and separation to a refrigerant intake circuit between the evaporator and the compressor.

6. The refrigerator according to one of Claims 1 to 5, wherein the evaporator is constituted of a plate heat exchanger including a plurality of plates stacked in parallel such that a plurality of refrigerant channels and a plurality of cooled-medium channels are alternately arranged.

7. The refrigerator according to Claim 6, wherein the evaporator is constituted of a plurality of the plate heat exchangers connected in series as multiple stages.

8. The refrigerator according to Claim 7, wherein the refrigerant precoolers constituted of the vapor-liquid separators are arranged in series as multiple stages at individual entrances of the plurality of plate heat exchangers.

9. A refrigerator having a heat pump cycle formed by sequentially connecting a compressor that compresses a refrigerant, a switching valve that switches a refrigerant cycle, a heat-source-side heat exchanger, an expansion valve that adiabatically expands the refrigerant, and a utilization-side heat exchanger,
wherein an economizer through which a high-pressure liquid refrigerant always flows in one direction via a refrigerant-flow-direction switching valve, which evaporates some of the high-pressure liquid refrigerant to supercool the refrigerant, and which has a circuit for injecting the evaporated medium-pressure refrigerant into an intermediate inlet of the compressor is disposed between the heat-source-side heat exchanger and the utilization-side heat exchanger, and a refrigerant precooler that precools the refrigerant supplied to the utilization-side heat exchanger or the heat-source-side heat exchanger functioning as an evaporator is disposed downstream of the economizer.

10. The refrigerator according to one of Claims 1 to 9, wherein the refrigerant precooler decreases the dryness of the refrigerant to nearly zero at an entrance of the evaporator.

11. The refrigerator according to one of Claims 1 to 10, wherein the refrigerator is a turbo refrigerator using a turbo compressor as the compressor.

FIG. 1

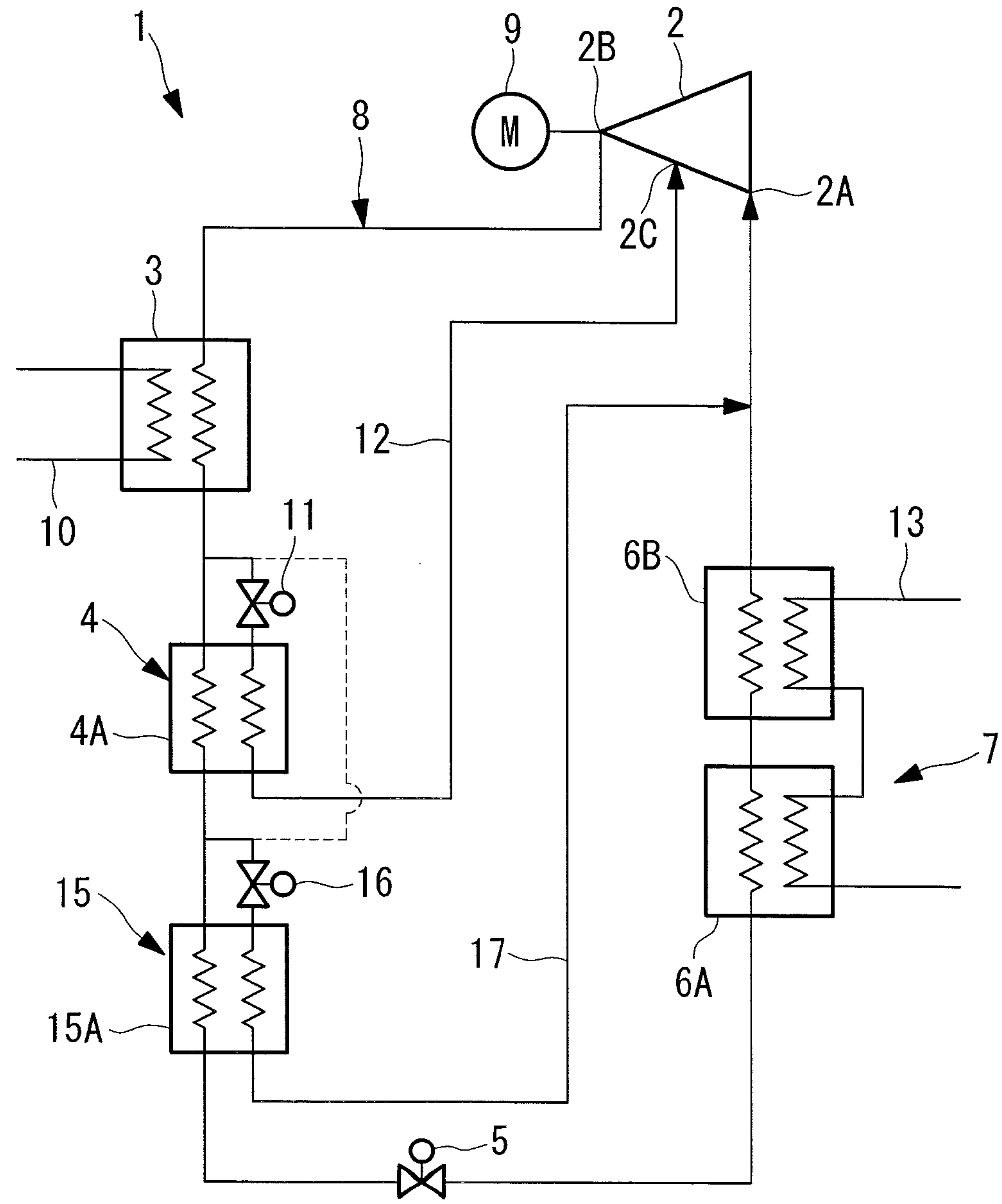
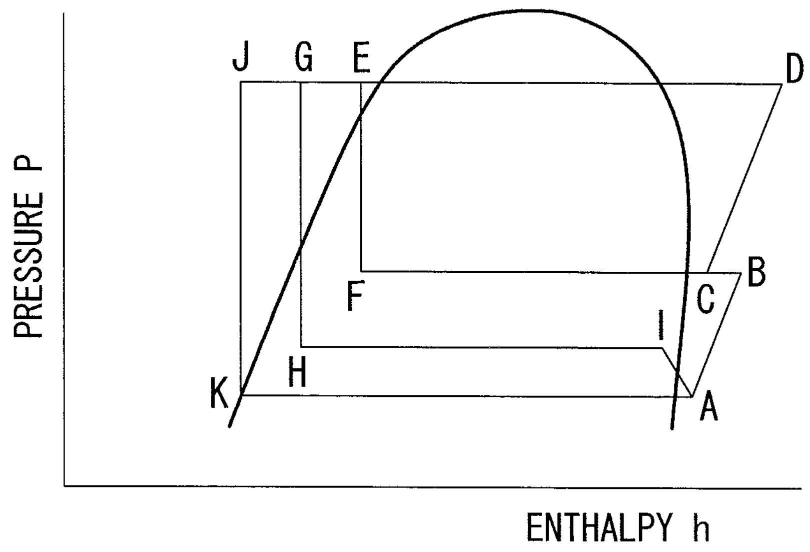


FIG. 2


J
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PRESSURE P
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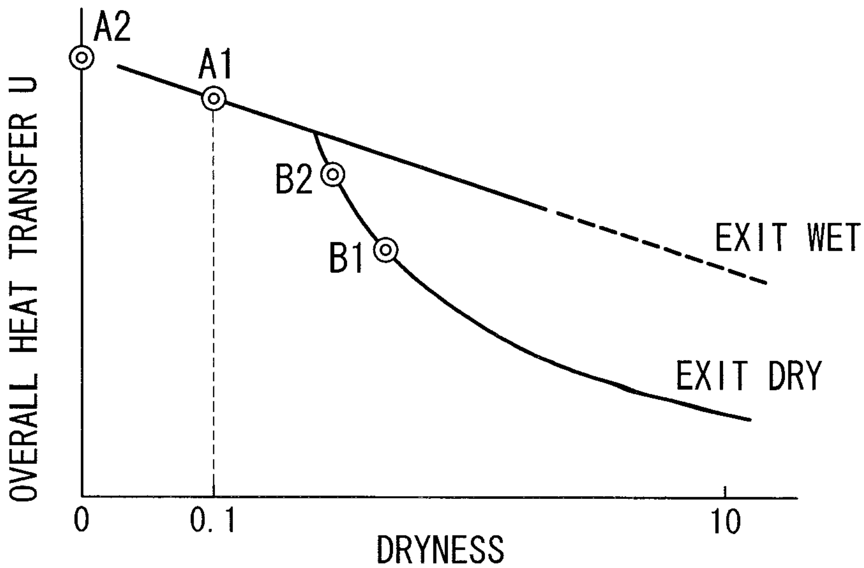
FIG. 3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A1
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B2
B1
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EXIT DRY
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0.1
10
DRYNESS

FIG. 4

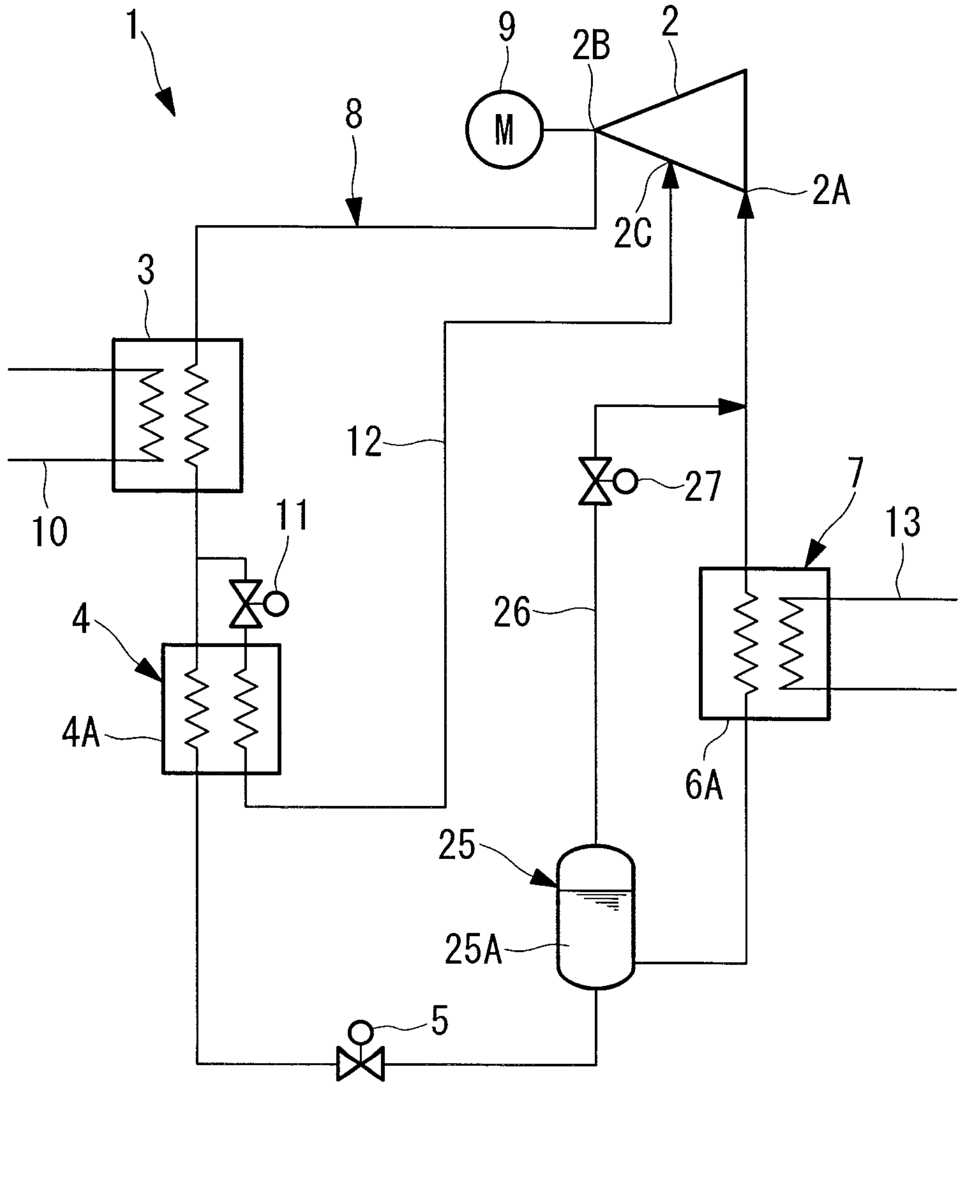


FIG. 5

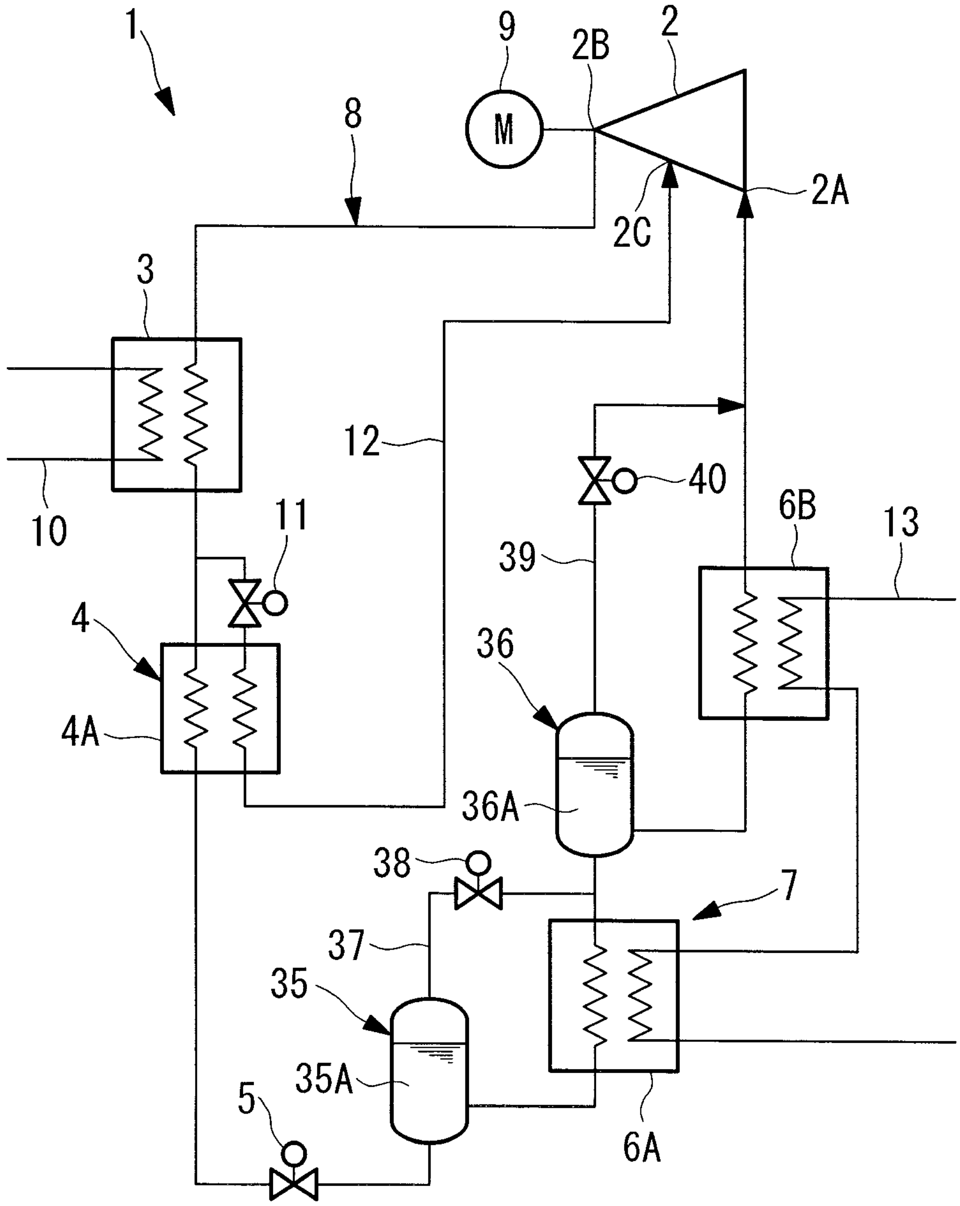
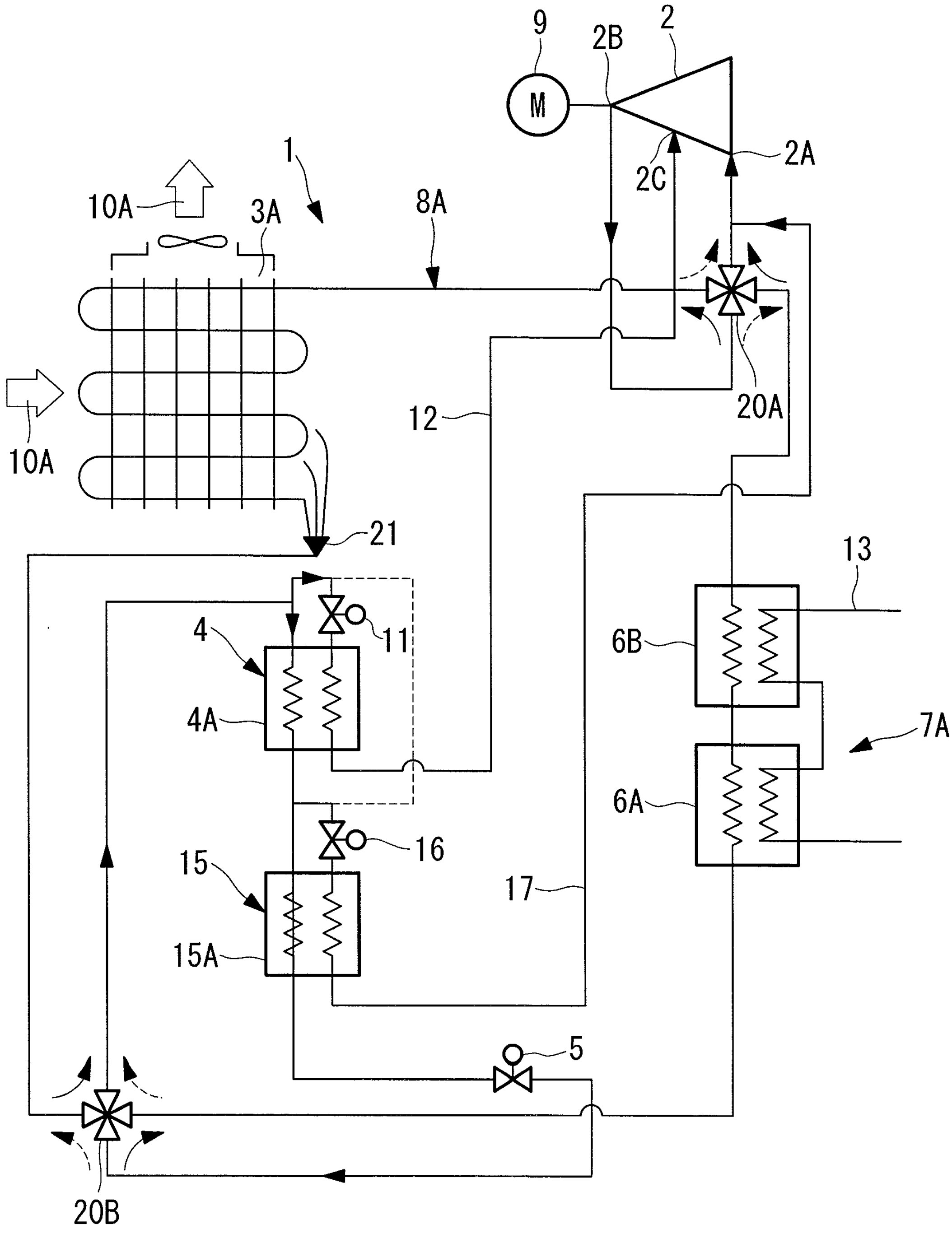


FIG. 6


9
M
2B
2
2A
2C
1
10A
3A
8A
20A
10A
12
21
13
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11
6B
4A
7A
6A
16
15
17
15A
5
20B

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2008/070473

A. CLASSIFICATION OF SUBJECT MATTER
F25B1/053(2006.01)i, *F25B1/00*(2006.01)i, *F25B1/10*(2006.01)i

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
F25B1/053, F25B1/00, F25B1/10

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

Jitsuyo Shinan Koho	1922-1996	Jitsuyo Shinan Toroku Koho	1996-2008
Kokai Jitsuyo Shinan Koho	1971-2008	Toroku Jitsuyo Shinan Koho	1994-2008

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X Y	WO 2007/105511 A1 (Daikin Industries, Ltd.), 20 September, 2007 (20.09.07), Par. Nos. [0025] to [0041], [0050] to [0058]; Figs. 1, 2 & JP 2007-240025 A	1-4,9 5-8,10,11
Y	JP 2006-118799 A (Denso Corp.), 11 May, 2006 (11.05.06), Par. Nos. [0007], [0035]; Fig. 3 (Family: none)	5-8,10,11
Y	JP 58-2563 A (Daikin Industries, Ltd.), 08 January, 1983 (08.01.83), Column 6, line 16 to column 9, line 13; Fig. 1 (Family: none)	8,10,11

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

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"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
26 December, 2008 (26.12.08)	13 January, 2009 (13.01.09)

Name and mailing address of the ISA/	Authorized officer
Japanese Patent Office	
Facsimile No.	Telephone No.

Form PCT/ISA/210 (second sheet) (April 2007)

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2008/070473

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	JP 2006-250479 A (Fujitsu General Ltd.), 21 September, 2006 (21.09.06), Par. No. [0007] (Family: none)	10,11

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REFERENCES CITED IN THE DESCRIPTION

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- JP 2001165590 A **[0005]**
- JP 2005337688 A **[0005]**