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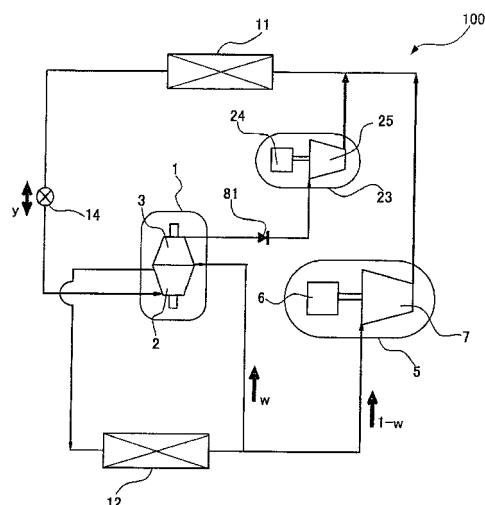
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(54) **REFRIGERATION CYCLING DEVICE AND EXPANDER INSTALLED IN SAME**

(57) Provided are a refrigeration cycle apparatus (100) and an expander (1) installed in the refrigeration cycle apparatus (100), the apparatus being capable of achieving flow rate matching without pre-expansion or bypassing of an expansion mechanism (2).

The refrigeration cycle apparatus (100) includes an expander (1) including an orbiting scroll (52) which includes a baseplate and spirals arranged on both surfaces of the baseplate, respectively, the baseplate having a high-pressure introduction hole (52e) guiding the pressure of a refrigerant sucked in an expansion mechanism (2) to a sub-compression mechanism (3).

FIG. 1



Description

Technical Field

[0001] The present invention relates to a refrigeration cycle apparatus that recovers expansion power generated upon expansion of a refrigerant to compress the refrigerant using the expansion power, and an expander installed therein.

Background Art

[0002] In a refrigeration cycle used for refrigeration and air conditioning, when expansion power is recovered in a positive displacement fluid machinery during a depressurizing process of a refrigerant and when the recovered expansion power is used for a compression process of the refrigerant in the positive displacement fluid machinery, it is necessary to consider volumetric flow rate matching of the refrigerant, a so-called "constraint of constant density ratio".

In such a positive displacement fluid machinery, the ratio of the refrigerant suction volume of an expansion mechanism to that of a compression mechanism that is operatively associated with power recovered from the expansion mechanism is fixed. When the flow rates of the refrigerant passing through these mechanisms are equal, the ratio of the specific volume of the refrigerant at the inlets of the mechanisms has to match the ratio of the suction volumes of the mechanisms.

[0003] A refrigeration cycle apparatus has been disclosed which includes an expander designed so that the ratio of the specific volume of the refrigerant at an inlet of an expansion mechanism to that of a compression mechanism matches the ratio of the suction volume of the expansion mechanism to that of the compression mechanism and in which, in order to correct the deviation from the design point caused by condition change during actual operation and in order to achieve matching, when (expansion mechanism inlet refrigerant specific volume / compression mechanism inlet refrigerant specific volume) > (expansion mechanism suction volume / compression mechanism suction volume), a predetermined flow rate of refrigerant is allowed to bypass the expansion mechanism, and when (expansion mechanism inlet refrigerant specific volume / compression mechanism inlet refrigerant specific volume) < (expansion mechanism suction volume / compression mechanism suction volume), the refrigerant is depressurized and pre-expanded by a predetermined amount of pressure before the inlet of the expansion mechanism (refer to Patent Literature 1, for example).

Citation List

Patent Literature

[0004]

Patent Literature 1: Japanese Unexamined Patent Application Publication JP A-2004-150 750 (pages 5 - 6, FIG. 3, for example)

Summary of Invention

Technical Problem

[0005] In the refrigeration cycle apparatus disclosed in Patent Literature 1, matching is intended to be performed under an operation condition deviating from the constraint of constant density ratio. For this purpose, a bypass including a control valve (control valve 7) is disposed in parallel to an expander (expander 6). The opening degree of the control valve is controlled on the basis of the bypass flow rate ratio that is set upon determination of an optimum high pressure at which COP is the highest.

[0006] In the refrigeration cycle apparatus disclosed in Patent Literature 1, however, pre-expansion for flow rate matching when (expansion mechanism inlet refrigerant specific volume / compression mechanism inlet refrigerant specific volume) < (expansion mechanism suction volume / compression mechanism suction volume) is often performed in a liquid-phase or a supercritical region on the liquid-phase side.

Accordingly, change in specific volume is small relative to the degree of reduction in pressure. Accordingly, most of the high-low pressure difference is pre-expanded or pre-expansion is performed until no recovery power can be obtained. Disadvantageously, matching may not be achieved.

[0007] In the refrigeration cycle apparatus disclosed in Patent Literature 1, although the amount of bypass is determined so that COP becomes the highest, the flow of refrigerant that has been bypassed is isenthalpically expanded by the expansion device (control valve 7). Disadvantageously, amount of energy proportionate to the above cannot be recovered, thus leading to loss of expansion energy as compared to that without the bypass.

[0008] The present invention has been made to solve the above-described disadvantages and an object of the invention is to provide a refrigeration cycle apparatus capable of enabling flow rate matching without pre-expansion or bypassing an expansion mechanism and to provide an expander installed in the refrigeration cycle apparatus.

Solution to the Problem

[0009] The present invention provides a refrigeration cycle apparatus that includes a main compressor, a radiator cooling a high-pressure refrigerant, an expander including an expansion mechanism recovering expansion power generated upon depressurization of the refrigerant and a sub-compression mechanism compressing the refrigerant using the expansion power, an evaporator heating the low-pressure refrigerant, and an additional compression mechanism further compressing the

refrigerant compressed by the sub-compression mechanism, the sub-compression mechanism being positioned downstream of the evaporator, the expansion mechanism being disposed downstream of the radiator and upstream of the evaporator.

The expander includes an orbiting scroll including a baseplate and spirals arranged on both surfaces of the baseplate, respectively, the baseplate having a high-pressure introduction hole that guides the pressure of the refrigerant sucked in the expansion mechanism to the sub-compression mechanism, an expansion side fixed scroll facing the orbiting scroll, the expansion side fixed scroll and the orbiting scroll constituting the expansion mechanism, and a sub-compression side fixed scroll facing the orbiting scroll such that the sub-compression side fixed scroll is positioned on the opposite side of the orbiting scroll from the expansion side fixed scroll, the sub-compression side fixed scroll and the orbiting scroll constituting the sub-compression mechanism.

[0010] The present invention provides an expander that includes an expansion mechanism recovering expansion power generated upon depressurization of a refrigerant and a sub-compression mechanism compressing the refrigerant using the expansion power, the expander including:

- an orbiting scroll including a baseplate and spirals arranged on both surfaces of the baseplate, respectively, the orbiting scroll having a high-pressure introduction hole that guides the pressure of the refrigerant sucked in the expansion mechanism to the sub-compression mechanism;
- an expansion side fixed scroll facing the orbiting scroll, the expansion side fixed scroll and the orbiting scroll constituting the expansion mechanism;
- a sub-compression side fixed scroll facing the orbiting scroll such that the sub-compression side fixed scroll is positioned on the opposite side of the orbiting scroll from the expansion side fixed scroll, the sub-compression side fixed scroll and the orbiting scroll constituting the sub-compression mechanism;
- an eccentric seal is disposed in a sliding portion between the orbiting scroll and the sub-compression side fixed scroll; and
- a concentric seal is disposed in a sliding portion between the orbiting scroll and the sub-compression side fixed scroll, the concentric seal being positioned on the axis side relative to the eccentric seal, in which the high-pressure introduction hole is opened between the concentric seal and the eccentric seal.

Advantageous Effects of the Invention

[0011] In the refrigeration cycle apparatus according to the invention, flow rate matching can be performed without pre-expansion or bypassing the expansion mechanism. Advantageously, flow rate matching can be performed with higher efficiency than flow rate matching

performed by bypassing or pre-expansion. Further, flow rate matching can be performed under conditions in which flow rate matching could not be achieved by pre-expansion, thus achieving a wider operating range.

[0012] In the expander according to the invention, forming of the high-pressure introduction hole can reduce heat leakage through a base plate of a central portion of an orbiting scroll. By introducing pressure before expansion to the sub-compression spiral side, balance of gas loads acting on the orbiting scroll in the axial direction can be improved, thus achieving satisfactory operation stability.

Brief Description of Drawings

[0013]

- FIG. 1 is a circuit configuration diagram schematically illustrating the configuration of a refrigerant circuit in a refrigeration cycle apparatus according to Embodiment 1 of the invention.
- FIG. 2 is a Mollier diagram illustrating operation states of the refrigeration cycle apparatus according to Embodiment 1 of the invention.
- FIG. 3 is a longitudinal sectional view schematically illustrating a sectional configuration of an expander installed in the refrigeration cycle apparatus according to Embodiment 1 of the invention.
- FIG. 4 is a plan view of an orbiting scroll of the expander installed in the refrigeration cycle apparatus according to Embodiment 1 of the invention when viewed from a sub-compression spiral side.
- FIG. 5 is a plan view of the orbiting scroll of the expander installed in the refrigeration cycle apparatus according to Embodiment 1 of the invention when viewed from an expansion spiral side.
- FIG. 6 is a fragmentary sectional view schematically illustrating states of thrusts acting on the orbiting scroll of the expander installed in the refrigeration cycle apparatus according to Embodiment 1 of the invention.
- FIG. 7 is a table illustrating four typical operation conditions of the refrigeration cycle apparatus.
- FIG. 8 is a circuit configuration diagram schematically illustrating the configuration of a refrigerant circuit of a refrigeration cycle apparatus of the related art.
- FIG. 9 includes tables illustrating results of cycle calculations when performing flow rate matching with a method of the related art.
- FIG. 10 includes tables illustrating results of cycle calculations when performing flow rate matching by flow division in the refrigeration cycle apparatus according to Embodiment 1.

FIG. 11 is a circuit configuration diagram schematically illustrating the configuration of a refrigerant circuit in a refrigeration cycle apparatus according to Embodiment 2 of the invention.

FIG. 11 is a circuit configuration diagram schematically illustrating the configuration of a refrigerant circuit in a refrigeration cycle apparatus according to Embodiment 3 of the invention.

Description of Embodiments

[0014] Embodiments of the invention will be described below with reference to the drawings.

Embodiment 1

[0015] FIG. 1 is a circuit configuration diagram schematically illustrating the configuration of a refrigerant circuit in a refrigeration cycle apparatus 100 according to Embodiment 1 of the invention. The circuit configuration and operation of the refrigeration cycle apparatus 100 will be described with reference to FIG. 1. It should be noted that the dimensional relationships of components in FIG. 1 and other subsequent drawings may be different from the actual ones.

In addition, in FIG. 1 and other subsequent drawings, components applied with the same reference signs correspond to the same or equivalent components. This is common through the full text of the description. Further, forms of components described in the full text of the description are mere examples, and the components are not limited to the described forms of components.

[0016] The refrigeration cycle apparatus 100 according to Embodiment 1 is used in devices equipped with a refrigeration cycle that circulates a refrigerant and is used, for example, a refrigerator, a freezer, a vending machine, an air-conditioning apparatus, a refrigeration apparatus, or a water heater. The refrigeration cycle apparatus 100 includes a main compressor 5, a radiator 11, a pre-expansion valve 14, an expander 1, an evaporator 12, a second compressor 23, and a check valve 81. The main compressor 5, the radiator 11, the pre-expansion valve 14, an expansion mechanism 2 of the expander 1, and the evaporator 12 are connected in series. A sub-compression mechanism 3 of the expander 1, the check valve 81, and the second compressor 23 are connected in series.

[0017] Specifically, in the refrigeration cycle apparatus 100, a main compression mechanism 7 of the main compressor 5 and the sub-compression mechanism 3 of the expander 1 are arranged in parallel on the outlet side of the evaporator 12. Furthermore, the discharge side of the sub-compression mechanism 3 of the expander 1 is connected through the check valve 81 to the suction side of a second compression mechanism 25 of the second compressor 23.

The discharge side of the second compression mechanism 25 of the second compressor 23 is connected to

the discharge side of the main compression mechanism 7 of the main compressor 5 at a point before the inlet of the radiator 11. The outlet side of the radiator 11 is connected through the pre-expansion valve 14 to the inlet side of the expansion mechanism 2 of the expander 1. The outlet side of the expansion mechanism 2 of the expander 1 is connected to the inlet side of the evaporator 12.

[0018] The main compressor 5 includes a motor 6 and the main compression mechanism 7 driven by the motor 6, and is configured to suck a refrigerant flowing from the evaporator 12 and compress the refrigerant into a high-temperature high-pressure state. This main compressor 5 may be constituted by, for example, a capacity-controllable inverter compressor. The radiator 11 functions as a condenser or a gas cooler depending on the used refrigerant, and is configured to exchange heat between air supplied from a fan (not illustrated) and the refrigerant. The pre-expansion valve 14 depressurizes and expands the refrigerant and may be constituted by a component having a variably controllable opening degree, for example, an electronic expansion valve. The evaporator 12 is configured to exchange heat between air supplied from a fan (not illustrated) and the refrigerant.

[0019] The expander 1 integrates the expansion mechanism 2 of a scroll type and the sub-compression mechanism 3 together and has functions of recovering expansion power generated when the refrigerant in the expansion mechanism 2 is expanded and of compressing the refrigerant in the sub-compression mechanism 3 using the recovered expansion power.

The second compressor 23 includes a motor 24 and the second compression mechanism 25 driven by the motor 24, and is configured to suck the refrigerant discharged from the sub-compression mechanism 3 of the expander 1 and turn the refrigerant into a high-temperature high-pressure state.

In other words, the second compressor 23 functions as an additional compression mechanism. The check valve 81, which is disposed between the sub-compression mechanism 3 of the expander 1 and the second compressor 23, is configured to permit the refrigerant to flow only in one direction.

[0020] The operation of the refrigeration cycle apparatus 100 will now be described.

When electric power is supplied to the motor 6 of the main compressor 5, the main compression mechanism 7 is driven. In the main compression mechanism 7, the sucked refrigerant is compressed from a low pressure P1 to a high pressure Ph and is then discharged therefrom. The high-temperature high-pressure refrigerant discharged from the main compressor 5 is cooled by transferring heat in the radiator 11.

The refrigerant cooled in the radiator 11 flows into the expansion mechanism 2 of the expander 1. In the expander 1, the expansion mechanism 2 recovers power generated when depressurizing the refrigerant flowing into the expansion mechanism 2 and the sub-compres-

sion mechanism 3 is driven by the recovered expansion power.

[0021] The activation of the sub-compression mechanism 3 of the expander 1 allows the refrigerant circulating through the refrigeration cycle to be diverged into a portion flowing into the sub-compression mechanism 3 of the expander 1 and a portion flowing into the main compression mechanism 7 of the main compressor 5 at a ratio of $w:(1-w)$.

Here, let v_{exi} denote the specific volume of the refrigerant at an inlet of the expansion mechanism 2, let v_s denote the specific volume of the refrigerant at an inlet of the sub-compression mechanism 3, and let σv_{EC}^* denote the ratio of the suction volume of the expansion mechanism to the suction volume of the sub-compression mechanism, then, the amount of suction (rotation speed) of the main compressor 5 is controlled such that w is $1/\sigma v_{EC}^* \times (v_{exi}/v_s)$, thus flow rate matching between expansion and sub-compression is achieved.

[0022] Furthermore, the refrigerant corresponding to the diversion ratio w is compressed by the sub-compression mechanism 3 from P_1 to an intermediate pressure P_m corresponding to the recovered power and is additionally compressed from P_m to P_h by the second compression mechanism 25 driven by the motor 24, thus achieving power balance between expansion and sub-compression.

Specifically, while the suction volume ratio σv_{EC}^* of the expander 1 is fixed and recovered power is dependent on condition, flow rate matching is devised by the diversion ratio in association with the main compressor 5 and by absorbing the balance of power by additional compression with the second compressor 23.

[0023] The pre-expansion valve 14 illustrated in FIG. 1 is configured to control pressure on the expansion mechanism side during a transient state, for example, upon activation, and is fully opened during a steady state such that it is not involved in the flow rate matching.

[0024] An operating condition of the cycle at this time plotted onto a Mollier diagram is illustrated in FIG. 2. In FIG. 2, the axis of ordinates indicates the refrigerant pressure P and the axis of abscissas indicates the specific enthalpy h . Point $b \rightarrow$ point c shown in FIG. 2 indicates a cooling process in the radiator 11 in FIG. 1. Since it is assumed that the refrigerant is CO_2 , the pressure P_h exceeds its critical pressure.

[0025] If the refrigerant is depressurized after the outlet of the radiator 11 by an expansion device which does not recover power such as an expansion valve, the pressure will decrease from point c to point d' with a constant specific enthalpy. Whereas, when the pressure is reduced by the expander 1 while expansion power is generated in the expansion mechanism 2, a process of point $c \rightarrow$ point d is experienced.

The difference $d'-d$ in specific enthalpy upon depressurization corresponds to energy recovered as power and is used as power to compress the flow corresponding to w with the sub-compression mechanism 3 as in point a

$\rightarrow$ point e . Additional compression by the second compressor 23 is indicated by point $e \rightarrow$ point b and compression by the main compressor 5 is indicated by point $a \rightarrow$ point b .

[0026] At this time, (enthalpy difference h_a-h_d) $\times$ (flow rate 1) denotes the refrigeration capacity and input electric power of (enthalpy difference h_e-h_a) $\times$ (flow rate $1-w$) + (enthalpy difference h_b-h_e) $\times$ (flow rate 1) is consumed by the motor 6 of the main compressor 5 and the motor 24 of the second compressor 23. Accordingly, the ratio of them is the COP of the cycle.

As compared to the cycle of point $a \rightarrow$ point $b \rightarrow$ point $c \rightarrow$ point $d' \rightarrow$ point a without power recovery, (enthalpy difference h_e-h_a) $\times$ (flow rate w) of the input and (enthalpy difference $h_d'-h_d$) $\times$ (flow rate 1) of the refrigeration capacity contribute to improvement of the COP.

[0027] As described above, the diversion ratio w is determined depending on the suction volume ratio σv_{EC}^* of the expander 1, and the ratio of the enthalpy difference (h_c-h_d) of the expansion mechanism 2 to the enthalpy difference (h_e-h_a) of the sub-compression mechanism 3 is equal to w . Accordingly, the level of P_m also depends on σv_{EC}^* .

In the expander 1, the refrigerant flows into the expansion mechanism 2 and the refrigerant flows into the sub-compression mechanism 3 in the same container such that power generated by gas pressure is transferred between the mechanisms. Accordingly, pressure and temperature at the sub-compression discharge of corresponding to point e can be controlled by setting of σv_{EC}^* while considering the loads on the expansion mechanism 2 and the sub-compression mechanism 3 and heat transfer therebetween.

[0028] FIG. 3 is a longitudinal sectional view schematically illustrating a sectional configuration of the expander 1 installed in the refrigeration cycle apparatus 100 according to Embodiment 1. The structure of the expander 1 will be described with reference to FIG. 3.

As described above, the expander 1 integrates the expansion mechanism 2 of the scroll type and the sub-compression mechanism 3 together and has functions of recovering expansion power through the expansion mechanism 2 generated when the refrigerant is expanded and has a function of compressing the refrigerant in the sub-compression mechanism 3 using the recovered expansion power.

The expansion mechanism 2 and the sub-compression mechanism 3 are housed in a hermetic vessel 4, serving as a pressure vessel. Referring to FIG. 3, the expansion mechanism 2 is positioned in a lower portion of the hermetic vessel 4 and the sub-compression mechanism 3 is positioned above the expansion mechanism 2.

[0029] In the bottom of the hermetic vessel 4, lubricant oil 9, such as refrigerating machine oil, is retained. The hermetic vessel 4 is connected with an expansion suction pipe 15 through which the refrigerant is sucked into the expansion mechanism 2, an expansion discharge pipe 16 through which the refrigerant expanded by the expan-

sion mechanism 2 is discharged, a sub-compression suction pipe 19 through which the refrigerant is sucked into the sub-compression mechanism 3, and a sub-compression discharge pipe 20 through which the refrigerant compressed in the sub-compression mechanism 3 is discharged.

The expansion suction pipe 15, the sub-compression discharge pipe 20, and the expansion discharge pipe 16 are communicatively connected at side surfaces of the hermetic vessel 4 to the inside thereof. The sub-compression suction pipe 19 is disposed in the upper surface of the hermetic vessel 4 so as to communicate with the inside thereof.

[0030] The expansion mechanism 2 is configured to depressurize the refrigerant sucked through the expansion suction pipe 15 such that it expands and discharges the refrigerant through the expansion discharge pipe 16. This expansion mechanism 2 includes an expansion side fixed scroll 51, which includes a baseplate and an expansion side spiral 51a disposed thereon, and an orbiting scroll 52, which includes a baseplate and an expansion side spiral 52a disposed thereon.

As illustrated in FIG. 3, the expansion side fixed scroll 51 is positioned on the lower side and the orbiting scroll 52 is positioned on the upper side. The expansion side spiral 51a, serving as an involute warp, extends from one surface of the baseplate of the expansion side fixed scroll 51. Furthermore, the expansion side spiral 52a, serving as an involute warp, extends from one surface of the baseplate of the orbiting scroll 52.

[0031] The expansion side spiral 51a of the expansion side fixed scroll 51 and the expansion side spiral 52a of the orbiting scroll 52 are arranged so as to mesh with each other. Expansion chambers 51d are formed whose volume changes by oscillatory movement of the expansion side spiral 51a and the expansion side spiral 52a. In addition, an eccentric seal 72b is disposed on an end surface of the orbiting scroll 52 on the expansion side fixed scroll 51 such that the seal surrounds a shaft 78.

[0032] The sub-compression mechanism 3 is configured to compress the refrigerant sucked through the sub-compression suction pipe 19 and discharge the refrigerant through the sub-compression discharge pipe 20. The sub-compression mechanism 3 includes a sub-compression side fixed scroll 61, which includes a baseplate and a sub-compression side spiral 61a disposed thereon, and the orbiting scroll 52 which includes a sub-compression side spiral 62a on the baseplate.

As illustrated in FIG. 3, the sub-compression side fixed scroll 61 is positioned on the upper side and the orbiting scroll 52 is positioned on the lower side. The sub-compression side spiral 61a, serving as an involute warp, extends from one surface of the baseplate of the sub-compression side fixed scroll 61. Furthermore, the sub-compression side spiral 62a, serving as an involute warp, extends from the other surface of the baseplate of the orbiting scroll 52.

[0033] The sub-compression side spiral 61a of the sub-

compression side fixed scroll 61 and the sub-compression side spiral 62a of the orbiting scroll 52 are arranged so as to mesh with each other. Sub-compression chambers 61d are formed whose volumes change by oscillatory movement of the sub-compression side spiral 61a and the sub-compression side spiral 62a. Furthermore, an oil return hole 31 for returning the lubricant oil 9 to the bottom of the hermetic vessel 4 extends in the axial direction through outer regions of the sub-compression side fixed scroll 61 and the expansion side fixed scroll 51. In addition, an eccentric seal 72a and a concentric seal 73 are arranged on an end surface of the orbiting scroll 52 facing the sub-compression side fixed scroll 61 such that the seals surround the shaft 78. Furthermore, a discharge valve 32 is disposed in a refrigerant discharge portion of the sub-compression side fixed scroll 61. This discharge valve 32 is opened to connect the sub-compression chamber 61d to the sub-compression discharge pipe 20 and is closed to isolate the sub-compression chamber 61d from the sub-compression discharge pipe 20.

[0034] The expansion side fixed scroll 51, the orbiting scroll 52, and the sub-compression side fixed scroll 61 each have a through-hole in substantially central portions thereof. The shaft 78 is inserted through the through-holes. The orbiting scroll 52 of the expansion mechanism 2 and the orbiting scroll 52 of the sub-compression mechanism 3 share the baseplate and is integrally configured. The baseplate has a high-pressure introduction hole 52e penetrating therethrough in the axial direction. The high-pressure introduction hole 52e connects the expansion chamber 51d to the space between the eccentric seal 72a and the concentric seal 73.

[0035] The shaft 78 is rotatably supported at both ends by a lower bearing 51b disposed at the center of the expansion side fixed scroll 51 and an upper bearing 61b disposed at the center of the sub-compression side fixed scroll 61. The orbiting scroll 52 has an orbiting bearing 52b formed in a thick portion at each central portion of the expansion side spiral 52a and the sub-compression side spiral 61a such that the orbiting bearing 52b is supported by a crank 78a of the shaft 78 penetrating therethrough. Accordingly, the orbiting scroll 52 can orbit in association with the rotation of the shaft 78.

[0036] An oil pump 76 for pumping the lubricant oil 9 is attached to the lower end of the shaft 78. In addition, the shaft 78 has therein an oil supply hole (not illustrated) through which the lubricant oil 9 pumped up by the oil pump 76 is allowed to pass. The lubricant oil 9 pumped up by the oil pump 76 passes through the oil supply hole in the shaft 78 and is supplied to the lower bearing 51b and the upper bearing 61b.

Furthermore, the lubricant oil 9 used in the bearings is returned to the bottom of the hermetic vessel 4 through the oil return hole 31. Furthermore, an orbiting scroll movement space having a predetermined size is provided on the periphery of the orbiting scroll 52 such that the orbiting scroll 52 can perform oscillatory movement.

[0037] An Oldham groove 52d is formed on the outer region of the orbiting scroll 52 on the side of the expansion side fixed scroll 51. An Oldham ring 77 that restricts rotation movement of the orbiting scroll 52 and enables orbital motion thereof is disposed in the Oldham groove 52d. In addition, a balancer 79a is attached to an upper end side of the shaft 78 and a balancer 79b is attached to a lower end portion thereof.

The balancers 79a and 79b are configured to cancel out centrifugal forces generated by oscillatory movement of the orbiting scroll 52. The material, size, shape, and other characteristics of the balancers are not particularly limited.

[0038] Specifically, as illustrated in FIG. 3, in the expander 1, the orbiting scroll 52 including the expansion side spiral 52a and the sub-compression side spiral 62a on each respective surfaces of the baseplate that are positioned back to back is combined with the expansion side fixed scroll 51 and the sub-compression side fixed scroll 61, thus forming the expansion mechanism 2 and the sub-compression mechanism 3, respectively.

[0039] In the expander 1, therefore, with the power generated when the high-pressure refrigerant flowing through the expansion suction pipe 15 in the expansion mechanism 2 is expanded, the orbiting scroll 52 performs oscillatory movement while being restricted by the Oldham ring 77 and the shaft 78 and the sub-compression mechanism 3 increases the pressure of the low-pressure refrigerant sucked through the sub-compression suction pipe 19 via a suction port (not illustrated because it is positioned in a section of different phase).

The refrigerant, which has been increased to an intermediate pressure, push opens the discharge valve 32 in a discharge port (in another section), so that the refrigerant is discharged to the sub-compression discharge pipe 20. The expanded refrigerant is discharged through the expansion discharge pipe 16 (in another section).

[0040] In the use of a dual-sided scroll mechanism expander like the expander 1 in which the expansion side spiral 52a and the sub-compression side spiral 62a are integrated back to back, the refrigerant on the expansion side passes by the refrigerant on the sub-compression side, with the baseplate of the orbiting scroll therebetween.

Accordingly, if the difference in temperature between the refrigerant on the expansion side and that on the sub-compression side is too large, heat leakage through the baseplate of the orbiting scroll may reach a level that cannot be disregarded. In particular, heat leakage from the sub-compression side to the expansion side acts such that the efficiency of cycle is reduced. It is therefore preferable to prevent heat leakage to the extent possible.

[0041] In the Mollier diagram of FIG. 2, the expansion process is indicated by point **c** → point **d** and the sub-compression process is indicated by point **a** → point **e**. Accordingly, the central portion of the expansion side spiral is in contact with the refrigerant in a state at point **c**, the outer region of the expansion side spiral is in contact

with the refrigerant in a state at point **d**, the outer region of the sub-compression side spiral is in contact with the refrigerant in a state at point **a**, and the central portion of the sub-compression side spiral is in contact with the refrigerant in a state at point **e**.

In other words, the refrigerant in the state at point **e** and the refrigerant in the state at point **c** are positioned back to back in the central portion of the baseplate and the refrigerant in the state at point **a** and the refrigerant in the state at point **d** are positioned back to back in the outer region of the baseplate.

As regards points **c**, **d**, and **a** of these points, pressures are determined depending on the operation condition. As regards point **e**, the quantity of state can be controlled while changing the diversion ratio **w** by selecting the ratio σvEC^* of the expansion mechanism suction volume to the sub-compression mechanism suction volume when designing the expander 1, because the level of P_m depends on σvEC^* as described above.

[0042] Furthermore, since the expansion process from the high pressure P_h to the low pressure P_1 and the sub-compression process from the low pressure P_1 to the intermediate pressure P_m proceed on each corresponding surface of the orbiting scroll, gas loads (thrusts) acting on the orbiting scroll in the axial direction are not equal to each other.

When the thrust difference excessively increases depending on the intermediate pressure P_m , an increase in friction loss caused by pressing of the edges of the spirals against each other or a reduction in operation stability may be caused. The suction volume ratio σvEC^* , therefore, has to be selected in consideration of both heat leakage and thrusts.

If the suction volume ratio of the expander 1 were to be selected so that the difference in temperature between point **e** and point **c** in the central portion of the baseplate is at a level at which a reduction in cycle efficiency due to internal heat leakage is permissible under predetermined conditions, P_m will be approximately less than or equal to $(P_h + P_1)/2$.

[0043] In this situation, although depending on the shape and size of the spirals, occurrence of excessive pressing of the edges of the sub-compression side spirals against each other is inevitable due to the thrust difference. Accordingly, no operation can be performed in such a situation. However, by introducing the high pressure on the expansion suction side to the central portion on the sub-compression side and increasing the thrust acting on the sub-compression side, it will be possible to counterbalance the thrust from the expansion side.

Accordingly, in the expander 1, the high-pressure introduction hole 52e penetrating through the orbiting scroll in the axial direction is provided so that the high pressure before expansion on the expansion inlet side acts on the space between the concentric seal 73 and the eccentric seal 72a in the central portion of the end surface on the sub-compression side.

[0044] FIG. 4 is a plan view of the orbiting scroll when

viewed from the sub-compression spiral side. FIG. 5 is a plan view of the orbiting scroll when viewed from the expansion spiral side. Features of the expander 1 will be described in more detail with reference to Figs. 4 and 5. The orbiting scroll has the orbiting bearing 52b at its center through which the shaft 78 penetrates. On each of the expansion side and the sub-compression side, a portion surrounding the orbiting bearing 52b is bulb-shaped (such that start points of involutes are connected by an arc).

[0045] In the orbiting scroll of the expander 1, the size of the expansion side spiral 52a operating in the range from the low pressure P1 to the high pressure Ph is smaller than that of the sub-compression side spiral 62a operating in the range from the low pressure P1 to the intermediate pressure Pm such that the area subject to pressure of the thrust acting on the expansion side is reduced.

In the outer region of the expander 1 on the expansion side, the Oldham groove 52d, in which a key of the Oldham ring 77 is fitted, is disposed. As illustrated in FIG. 3, the Oldham ring 77, positioned between the orbiting scroll and the expansion side fixed scroll 51, restricts the position of the orbiting scroll at the outer region of the baseplate of the orbiting scroll on the expansion side.

[0046] Referring to FIG. 5, a portion of the orbiting bearing 52b in communication with a low-pressure atmosphere in the container is separated from the central portion at high pressure before expansion by the eccentric seal 72b on an end surface of the bulb-shaped portion, surrounding the orbiting bearing 52b, of the expansion side spiral 52a. The high-pressure introduction hole 52e is formed in the innermost part of the expansion chamber 51d so as to guide pressure, immediately after being sucked into the expansion mechanism, to the sub-compression side.

Referring to FIG. 4, the high-pressure introduction hole 52e on the sub-compression side is opened between the concentric seal 73 and the eccentric seal 72a on an end surface of the bulb-shaped portion. The concentric seal 73 partitions the orbiting bearing with low pressure and the eccentric seal 72a partitions the central portion on the sub-compression side with intermediate pressure, such that the high pressure acts on a portion which is inside the eccentric seal 72a and outside the concentric seal 73.

[0047] FIG. 6 is a fragmentary sectional view schematically illustrating states of thrusts acting on the orbiting scroll. Thrusts acting on the orbiting scroll of the expander 1 will be described in detail with reference to FIG. 6. Note that arrows illustrated in FIG. 6 indicate thrust loads acting on each surface of the orbiting scroll.

[0048] Referring to FIG. 6, pressures ranging from the high pressure Ph to the low pressure P1 act on a portion between the eccentric seal 72b and the outer portion of the spiral on the expansion side and pressures ranging from the intermediate pressure Pm to the low pressure P1 act on a portion between the eccentric seal 72a and

the outer portion of the spiral on the sub-compression side.

In addition, the high pressure Ph guided through the high-pressure introduction hole 52e from the expansion side acts on the portion that is inside the eccentric seal 72a and outside the concentric seal 73. Thus, overall, the thrust on the expansion side is approximately counter-balanced against that on the sub-compression side.

At this time, although the placement of the communicating path (high-pressure introduction hole 52e) between the expansion side and the sub-compression side causes an increase in heat leakage when the difference in temperature between the central portion on the expansion side and that on the sub-compression side is large, there is no problem so long as the difference in temperature therebetween is restrained by selection of the suction volume ratio.

[0049] Furthermore, the high-pressure introduction hole 52e guiding the high pressure before expansion to the end surface of the bulb-shaped portion of the spiral on the sub-compression side is illustrated as a narrow hole penetrating through the orbiting scroll as illustrated in the sectional view of FIG. 3.

Even if high-pressure is guided from a piping before suction of the expansion mechanism, for example, a piping before the expansion suction pipe 15, through the sub-compression side fixed scroll 61 to the end surface of the bulb-shaped portion at the center of the spiral on the sub-compression side, the function will not change.

In this case, it is needless to say that the high-pressure introduction hole 52e opened on the bottom surface of the warp in the central portion of the sub-compression side fixed scroll has to be positioned outside the concentric seal 73 and inside the eccentric seal 72a of the orbiting scroll at any time even when the orbiting scroll is performing an oscillatory movement.

[0050] FIG. 7 is a table illustrating four typical operation conditions of the refrigeration cycle apparatus. FIG. 8 is a circuit configuration diagram schematically illustrating a refrigerant circuit configuration of a refrigeration cycle apparatus of the related art (hereinafter, referred to as the "refrigeration cycle apparatus 100' "). FIG. 9 includes tables illustrating results of cycle calculations when performing flow rate matching with a method of the related art.

The four typical operation conditions in the refrigeration cycle apparatus 100' will be described with reference to Figs. 7 to 9. The refrigeration cycle apparatus 100' illustrated in FIG. 8 includes a refrigeration cycle in which flow rate matching is performed by pre-expanding (expansion valve 13') and bypassing the expansion mechanism (bypass pipe 40').

[0051] The refrigeration cycle apparatus 100' differs from the refrigeration cycle apparatus 100 according to Embodiment 1 in that the second compressor 23 is not disposed and the expansion valve 13' and the bypass pipe 40' are arranged.

When flow rate matching is performed in the refrigeration

cycle apparatus 100' in FIG. 8 under the four conditions in FIG. 7, (expansion mechanism suction volume/compression mechanism suction volume) at which both a bypass ratio x and a pre-expansion coefficient y , serving as the ratio of reduction in pressure by pre-expansion before an expansion inlet that is a ratio between the overall high and low pressures of the pre-expansion width, are 0, namely, (expansion mechanism inlet refrigerant specific volume/compression mechanism inlet refrigerant specific volume) that is determined depending on operation condition is σvEC .

Furthermore, (expansion mechanism inlet refrigerant specific volume/expansion mechanism outlet refrigerant specific volume) is σvE . The cycle COP at this time is a value COP_{th}.

[0052] σvE on the operation condition side corresponds to the expansion volume ratio (expansion start volume/expansion end volume) of an expansion mechanism 2' = σvE^* . Typically, σvE^* cannot be changed because it is specific to the expander design. If σvE^* significantly differs from σvE , therefore, recovered power will be markedly reduced due to underexpansion or overexpansion.

[0053] FIG. 9 illustrates results of cycle calculations in each of which (expansion mechanism suction volume/compression mechanism suction volume) = σvEC^* is set to be equal to (expansion mechanism inlet refrigerant specific volume/compression mechanism inlet refrigerant specific volume) and (expansion start volume/expansion end volume) = σvE^* is set to be equal to (expansion mechanism inlet refrigerant specific volume/expansion mechanism outlet refrigerant specific volume) for a certain condition and flow rate matching is performed to the other three conditions with the pre-expansion coefficient y and the bypass ratio x .

[0054] FIG. 9 illustrates the pre-expansion coefficient y , the bypass ratio x , the intermediate pressure P_m serving as pressure at the sub-compression outlet, which are necessary for flow rate matching, and the ratio of the COP at this time to the COP_{th} illustrated in FIG. 7 in the use of an expander 1' in which the expansion volume ratio = (expansion mechanism suction volume/compression mechanism suction volume) is σvEC^* and (expansion start volume/expansion end volume) is σvE^* under conditions that (expansion mechanism inlet refrigerant specific volume/compression mechanism inlet refrigerant specific volume) is σvEC and (expansion mechanism inlet refrigerant specific volume/expansion mechanism outlet refrigerant specific volume) is σvE .

[0055] When $\sigma vEC^* = \sigma vEC$, neither bypass nor pre-expansion is needed. When $\sigma vEC^* < \sigma vEC$, bypass is performed to match the flow rates. When $\sigma vEC^* > \sigma vEC$, pre-expansion is performed to match the flow rates. When σvEC^* is significantly greater than σvEC , even if maximum pre-expansion is performed, matching will not be achieved, or, matching may be achieved but the COP ratio may be below 100 %, so that the performance improvement effect due to expansion power recovery may

not be obtained.

Such a case corresponds to a cooling rated condition when σvEC^* is set to a heating condition. It will be easily understood that this method is not suitable for using the expander 1', which is designed for heating, under the cooling rated condition.

[0056] Whereas, results of cycle calculations in the case where flow rate matching is performed in the refrigeration cycle apparatus 100 according to Embodiment 1 using the diversion ratio w are illustrated in FIG. 10. FIG. 10 illustrates w necessary for matching, the intermediate pressure P_m dependent on w , and the COP ratio at this time, when $\sigma vEC^* = 0.5$ is fixed.

Since there is no power recovery loss associated with flow rate matching, the COP ratio reflects the influence of underexpansion or overexpansion and drops when σvE does not agree with the expansion volume ratio σvE^* , particularly, during cooling operation with one designed for heating condition or during heating operation with one designed for cooling condition. It is, however, understood that there is no case where the COP ratio is below 100 %, as illustrated in FIG. 9.

[0057] Operation states of the refrigeration cycle apparatus 100' can be described using the Mollier diagram of FIG. 2. A refrigerant is subjected to sub-compression as in point **a** → point **e** in a sub-compression mechanism 3' and is then compressed as in point **e** → point **b** in a main compressor 5'. The refrigerant is cooled as in point **b** → point **c** by a radiator 11' and is then subjected to isentropic expansion as in point **c** → point **d** in the expansion mechanism 2'.

In accordance with the requirement of the flow rate matching, however, the flow rate x of refrigerant is allowed to flow through the bypass and is depressurized by the expansion valve 13' such that it is subjected to isenthalpic expansion as in point **c** → point **d'** and the flow rate of refrigerant passing through the expansion mechanism 2' will become $1-x$.

Alternatively, the refrigerant is isenthalpically expanded from point **c** toward point **d'** by the amount corresponding to the pre-expansion coefficient y in the pre-expansion valve 14 and is then isentropically expanded by the expansion mechanism 2'.

[0058] Accordingly, when bypass is performed, recovered power amounts to the flow rate $1-x$ corresponding to the enthalpy difference $d'-d$, and when pre-expansion is performed, recovered power amounts to the enthalpy difference corresponding to the isentropic expansion from pressure $P_1 + (P_h - P_1) \cdot (1-y)$ to pressure P_1 . In each of these cases, recovered power decreases more than that in the case where the whole amount of refrigerant is subjected to isentropic expansion without bypassing or pre-expansion.

As compared with the refrigeration cycle apparatus 100 in which recovered power associated with flow rate matching hardly decreases and pressure is increased by the amount of the diversion ratio w in the sub-compression mechanism 3', in the refrigeration cycle apparatus

100', the full flow of refrigerant is compressed using recovered power that is lower by the amount of loss caused by flow rate matching. Accordingly, the level of P_m is lower than that in FIG. 2 and values of P_m in the tables of Figs. 9 and 10 are also lower.

[0059] According to the matching method of the related art, by bypassing the expansion mechanism or by pre-expanding, the ratio of the volumetric flow rate at the expansion inlet to that at the sub-compression inlet is made to match the ratio of the expansion mechanism suction volume to the sub-compression mechanism suction volume. In other words, the volumetric flow rate is controlled mainly on the expansion process side.

Disadvantageously, recovered power decreases, and compression work by the main compressor accordingly increases. On the other hand, according to the matching method in the refrigeration cycle apparatus 100, the volumetric flow rate is controlled on the compression process side, that is, by using the diversion ratio w , which is the ratio of the compression process from the low pressure P_1 to the intermediate pressure P_m by the sub-compression mechanism 3 of the expander 1 to that by the main compression mechanism 7 driven by a power source.

This is a factor of the difference between the COP based on the matching method in the refrigeration cycle apparatus 100' and that based on the matching method using flow division in the refrigeration cycle apparatus 100.

[0060] As described above, in the refrigeration cycle apparatus 100 according to Embodiment 1, the flow rate of refrigerant sucked into the main compressor is controlled such that the ratio w of the flow rate of refrigerant sucked into the sub-compression mechanism 3 to the full flow is (expansion mechanism inlet refrigerant specific volume/sub-compression mechanism inlet refrigerant specific volume)/(expansion mechanism suction volume/sub-compression mechanism suction volume), thus enabling flow rate matching.

Advantageously, flow rate matching can be performed with higher efficiency than that in flow rate matching by bypassing or by pre-expanding or even under conditions that flow rate matching could not be performed with pre-expansion, thus achieving a wider operating range.

[0061] Furthermore, according to the expander 1 used in the refrigeration cycle apparatus 100, controlling the ratio of the expansion mechanism suction volume to the sub-compression mechanism suction volume reduces the difference in temperature between the sub-compression discharge side and the expansion inlet side, so that heat leakage in the central portion of the orbiting scroll through the baseplate can be reduced.

Guiding pressure before expansion to the sub-compression spiral side improves balance of gas loads acting on the orbiting scroll in the axial direction, thus improving operation stability. In the refrigeration cycle apparatus 100 equipped with this expander 1, therefore, a reduction in cycle efficiency due to internal heat leakage is small in addition to the above-described advantages.

[0062] Although the explanatory description is made with the Mollier diagram of FIG. 2 based on the assumption that CO_2 refrigerant is used, any refrigerant other than CO_2 may be used in the refrigeration cycle apparatus 100. A refrigerant available in the refrigeration cycle apparatus 100 will now be described. Examples of the refrigerant that can be used in the refrigeration cycle apparatus include a non-azeotropic refrigerant mixture, a near-azeotropic refrigerant mixture, and a single refrigerant.

Examples of the non-azeotropic refrigerant mixture include R-407C (R32/R125/R134a) that are HFC (hydrofluorocarbon) refrigerants. Examples of the near-azeotropic refrigerant mixture include R410A (R32/R125) and R404A (R125/R143a/R134a), which are HFC refrigerants.

[0063] Furthermore, examples of the single refrigerant include R22 that is an HCFC (hydrochlorofluorocarbon) refrigerant and R134a that is an HFC refrigerant. In addition, any of propane, isobutene, and ammonia, serving as natural refrigerants, can be used. Furthermore, examples of a refrigerant undergoing transition to its supercritical state include a refrigerant mixture of carbon dioxide and ether (such as dimethyl ether or hydrofluoroether). A refrigerant for application or use of the refrigeration cycle apparatus 100 may therefore be used.

Embodiment 2

[0064] FIG. 11 is a circuit configuration diagram schematically illustrating a refrigerant circuit configuration of a refrigeration cycle apparatus 100a according to Embodiment 2 of the invention. Features of the refrigeration cycle apparatus 100a will be described with reference to FIG. 11. Note that the same components as those in Embodiment 1 are designated by the same reference numerals.

The difference between Embodiment 1 and Embodiment 2 will be mainly described. Furthermore, various refrigerants described in Embodiment 1 may be used in the refrigeration cycle apparatus 100a.

[0065] Like the refrigeration cycle apparatus 100 according to Embodiment 1, the refrigeration cycle apparatus 100a according to Embodiment 2 is used as an apparatus including a refrigeration cycle through which a refrigerant is circulated, for example, a refrigerator, a freezer, a vending machine, an air-conditioning apparatus, a refrigeration apparatus, or a water heater.

The refrigeration cycle apparatus 100a also includes a main compressor 5, a radiator 11, a pre-expansion valve 14, an expander 1, an evaporator 12, a second compressor 23, and a check valve 81. Note that a connection state of the components differs from that in the refrigeration cycle apparatus 100 according to Embodiment 1.

[0066] In the refrigeration cycle apparatus 100a, flow is divided at an outlet of the evaporator 12 such that the flow rate w of refrigerant flows to the sub-compression mechanism 3 and the flow rate $1-w$ of refrigerant flows

to the main compressor 5.

The second compressor 23 additionally compresses not only the refrigerant discharged from the sub-compression mechanism 3 but also that discharged from the main compressor 5 such that the refrigerant is compressed from an intermediate pressure to a high pressure. Specifically, in the refrigeration cycle apparatus 100 according to Embodiment 1, the main compressor 5 and the second compressor 23 are arranged in parallel.

Whereas, in the refrigeration cycle apparatus 100a, the main compressor 5 and the second compressor 23 are arranged in series such that the refrigerant discharged from the sub-compression mechanism 3 is permitted to flow through the check valve 81 to a point between the main compressor 5 and the second compressor 23.

[0067] As regards the second compressor 23 in the refrigeration cycle apparatus 100 according to Embodiment 1, since it additionally compresses only the refrigerant that has been subjected to sub-compression, a compressor having a small stroke volume may be used. On the other hand, the second compressor 23 in the refrigeration cycle apparatus 100a according to Embodiment 2 additionally compresses not only the refrigerant that has been subjected to sub-compression but also the refrigerant that has been compressed by the main compressor 5. Accordingly, a compressor having a relatively large stroke volume may be used.

[0068] For example, in the case where the design rotation speed of each of the compressors (the main compressor 5, the second compressor 23) of the refrigeration cycle apparatus 100 according to Embodiment 1 is approximately 50 [rps], the main compressor 5 has a stroke volume of approximately 29.2 [cm³/rev] and the second compressor 23 has a stroke volume of approximately 5.9 [cm³/rev].

Whereas, in the case where the design rotation speed of each of the compressors (the main compressor 5, the second compressor 23) of the refrigeration cycle apparatus 100a according to Embodiment 2 is approximately 50 [rps], the main compressor 5 has a stroke volume of approximately 29.2 [cm³/rev] and the second compressor 23 has a stroke volume of approximately 26.9 [cm³/rev].

[0069] Generally, when using a positive displacement compressor, the smaller the stroke volume of the compressor, the harder it will be to maintain its efficiency. In the refrigeration cycle apparatus 100a, the efficiency of the entire cycle is increased because a compressor having a relatively large stroke volume can be used as the second compressor 23.

[0070] As described above, in the refrigeration cycle apparatus 100a according to Embodiment 2, the flow rate of refrigerant sucked into the main compressor is controlled such that the ratio w of the flow rate of refrigerant sucked into the sub-compression mechanism 3 to the full flow is (expansion mechanism inlet refrigerant specific volume/sub-compression mechanism inlet refrigerant specific volume)/(expansion mechanism suction vol-

ume/sub-compression mechanism suction volume), thus enabling flow rate matching.

Advantageously, flow rate matching can be performed with higher efficiency than that in flow rate matching by bypassing or by pre-expanding or even under conditions that flow rate matching could not be performed with pre-expansion, thus achieving a wider operating range. Moreover, since a compressor having a relatively large stroke volume can be used as the second compressor 23, the efficiency of the entire cycle is further increased.

[0071] Furthermore, according to the expander 1 used in the refrigeration cycle apparatus 100a, controlling the ratio of the expansion mechanism suction volume to the sub-compression mechanism suction volume reduces the difference in temperature between the sub-compression discharge side and the expansion inlet side, so that heat leakage in the central portion of the orbiting scroll through the baseplate can be reduced.

Guiding pressure before expansion to the sub-compression spiral side improves balance of gas loads acting on the orbiting scroll in the axial direction, thus improving operation stability. In the refrigeration cycle apparatus 100a equipped with this expander 1, therefore, a reduction in cycle efficiency due to internal heat leakage is small in addition to the above-described advantages.

Embodiment 3

[0072] FIG. 12 is a circuit configuration diagram schematically illustrating a refrigerant circuit configuration of a refrigeration cycle apparatus 100b according to Embodiment 3 of the invention. Features of the refrigeration cycle apparatus 100b will be described with reference to FIG. 12. Note that the same components as those in Embodiments 1 and 2 are designated by the same reference numerals.

The difference between Embodiment 3 and Embodiments 1 and 2 will be mainly described. Furthermore, various refrigerants described in Embodiment 1 may be used in the refrigeration cycle apparatus 100b.

[0073] Like the refrigeration cycle apparatus 100 according to Embodiment 1, the refrigeration cycle apparatus 100b according to Embodiment 3 is used as an apparatus including a refrigeration cycle through which a refrigerant is circulated, for example, a refrigerator, a freezer, a vending machine, an air-conditioning apparatus, a refrigeration apparatus, or a water heater.

Similarly, the refrigeration cycle apparatus 100b includes a main compressor 5, a radiator 11, a pre-expansion valve 14, an expander 1, an evaporator 12, and a check valve 81. In other words, the refrigeration cycle apparatus 100b differs from the refrigeration cycle apparatus 100 according to Embodiment 1 and the refrigeration cycle apparatus 100a according to Embodiment 2 in that a second compressor is not included.

[0074] In the refrigeration cycle apparatus 100b, flow is divided at an outlet of the evaporator 12 such that the flow rate w of refrigerant flows to the sub-compression

mechanism 3 and the flow rate 1-w of refrigerant flows to the main compressor 5. The refrigerant divided at a low-pressure is compressed in the sub-compression mechanism 3 and is allowed to return to a compression chamber of the main compressor 5 that is in the course of compression without being additionally compressed by the second compressor.

Specifically, in the refrigeration cycle apparatus 100b, the full flow of refrigerant is compressed from an intermediate pressure to a high pressure in the main compressor 5. The main compressor 5, therefore, includes a path and a port (injection port) for taking in the refrigerant from the sub-compression mechanism 3 into the compression chamber.

[0075] In the refrigeration cycle apparatus 100b, the main compressor 5 has to include a path and a port for taking in the refrigerant from the sub-compression mechanism 3 into the compression chamber according to the intermediate pressure but a second compressor is not disposed, so cost can be accordingly reduced. In other words, part of the main compressor 5 serves as an additional compression mechanism in the refrigeration cycle apparatus 100b.

[0076] As described above, in the refrigeration cycle apparatus 100b according to Embodiment 3, the flow rate of refrigerant sucked into the main compressor is controlled such that the ratio w of the flow rate of refrigerant sucked into the sub-compression mechanism 3 to the full flow is $(\text{expansion mechanism inlet refrigerant specific volume}/\text{sub-compression mechanism inlet refrigerant specific volume})/(\text{expansion mechanism suction volume}/\text{sub-compression mechanism suction volume})$, thus enabling flow rate matching.

Advantageously, flow rate matching can be performed with higher efficiency than that in flow rate matching by bypassing or by pre-expanding or even under conditions that flow rate matching could not be performed with pre-expansion, thus achieving a wider operating range. Moreover, since the second compressor can be omitted, the cost can be reduced by the omission.

[0077] Furthermore, according to the expander 1 used in the refrigeration cycle apparatus 100b, controlling the ratio of the expansion mechanism suction volume to the sub-compression mechanism suction volume reduces the difference in temperature between the sub-compression discharge side and the expansion inlet side, so that heat leakage in the central portion of the orbiting scroll through the baseplate can be reduced.

Guiding pressure before expansion to the sub-compression spiral side improves balance of gas loads acting on the orbiting scroll in the axial direction, thus improving operation stability. In the refrigeration cycle apparatus 100a equipped with this expander 1, therefore, a reduction in cycle efficiency due to internal heat leakage is small in addition to the above-described advantages.

List of Reference Signs

[0078]

5	1	= expander
	1'	= expander
	2	= expansion mechanism
	2'	= expansion mechanism
	3	= sub-compression mechanism
10	3'	= sub-compression mechanism
	4	= hermetic vessel
	5	= main compressor
	5'	= main compressor;
	6	= motor
15	6'	= motor
	7	= main compression mechanism
	7'	= main compression mechanism
	9	= lubricant oil
	11	= radiator
20	11'	= radiator
	12	= evaporator
	12'	= evaporator
	13'	= expansion valve
	14	= pre-expansion valve
25	14'	= pre-expansion valve
	15	= expansion suction pipe
	16	= expansion discharge pipe
	19	= sub-compression suction pipe
	20	= sub-compression discharge pipe
30	23	= second compressor
	24	= motor
	25	= second compression mechanism
	31	= oil return hole
	32	= discharge valve
35	40'	= bypass pipe
	51	= expansion side fixed scroll
	51a	= expansion side spiral
	51b	= lower bearing
	51d	= expansion chamber
40	52	= orbiting scroll
	52a	= expansion side spiral
	52b	= orbiting bearing
	52d	= Oldham groove
	52e	= high-pressure introduction hole
45	61	= sub-compression side fixed scroll
	61a	= sub-compression side spiral
	61b	= upper bearing
	61d	= sub-compression chamber
	62a	= sub-compression side spiral
50	72a	= eccentric seal
	72b	= eccentric seal
	73	= concentric seal
	76	= oil pump
	77	= Oldham ring
55	78	= shaft
	78a	= crank
	79a	= balancer
	79b	= balancer

81 = check valve
 81' = check valve
 100 = refrigeration cycle apparatus
 100' = refrigeration cycle apparatus
 100a = refrigeration cycle apparatus
 100b = refrigeration cycle apparatus.

Claims

1. A refrigeration cycle apparatus, comprising:

- a main compressor, a radiator cooling a high-pressure refrigerant, an expander including an expansion mechanism recovering expansion power generated upon depressurization of the refrigerant and a sub-compression mechanism compressing the refrigerant using the expansion power, an evaporator heating the low-pressure refrigerant, and an additional compression mechanism further compressing the refrigerant compressed by the sub-compression mechanism, the sub-compression mechanism being positioned downstream of the evaporator, the expansion mechanism being disposed downstream of the radiator and upstream of the evaporator; and

the expander including

- an orbiting scroll including a baseplate and spirals arranged on both surfaces of the baseplate, respectively, the baseplate having a high-pressure introduction hole that guides the pressure of the refrigerant sucked in the expansion mechanism to the sub-compression mechanism,
 - an expansion side fixed scroll facing the orbiting scroll, the expansion side fixed scroll and the orbiting scroll constituting the expansion mechanism, and
 - a sub-compression side fixed scroll facing the orbiting scroll such that the sub-compression side fixed scroll is positioned on the opposite side of the orbiting scroll from the expansion side fixed scroll, the sub-compression side fixed scroll and the orbiting scroll constituting the sub-compression mechanism.

2. The refrigeration cycle apparatus of claim 1, wherein an eccentric seal and a concentric seal that is positioned on an axis side relative to the eccentric seal are arranged in a sliding portion between the orbiting scroll and the sub-compression side fixed scroll, and the high-pressure introduction hole is opened between the concentric seal and the eccentric seal.

3. The refrigeration cycle apparatus of claim 1 or 2,

wherein the additional compression mechanism is a second compressor disposed in parallel to the main compressor.

4. The refrigeration cycle apparatus of claim 1 or 2, wherein the additional compression mechanism is a second compressor disposed in series with the main compressor, the second compressor sucking and compressing both the refrigerant compressed by the sub-compression mechanism and the refrigerant compressed by the main compressor.

5. The refrigeration cycle apparatus of claim 1 or 2, wherein a portion of a compression mechanism included in the main compressor functions as the additional compression mechanism.

6. The refrigeration cycle apparatus of any one of claims 1 to 5, wherein a refrigerant that enters a supercritical state on the high-pressure side is used as the refrigerant.

7. An expander that includes an expansion mechanism recovering expansion power generated upon depressurization of a refrigerant and a sub-compression mechanism compressing the refrigerant using the expansion power, the expander comprising:

- an orbiting scroll including a baseplate and spirals arranged on both surfaces of the baseplate, respectively, the orbiting scroll having a high-pressure introduction hole that guides the pressure of the refrigerant sucked in the expansion mechanism to the sub-compression mechanism;
 - an expansion side fixed scroll facing the orbiting scroll, the expansion side fixed scroll and the orbiting scroll constituting the expansion mechanism;
 - a sub-compression side fixed scroll facing the orbiting scroll such that the sub-compression side fixed scroll is positioned on the opposite side of the orbiting scroll from the expansion side fixed scroll, the sub-compression side fixed scroll and the orbiting scroll constituting the sub-compression mechanism;
 - an eccentric seal disposed in a sliding portion between the orbiting scroll and the sub-compression side fixed scroll; and
 - a concentric seal disposed in a sliding portion between the orbiting scroll and the sub-compression side fixed scroll, the concentric seal being positioned on the axis side relative to the eccentric seal, wherein the high-pressure introduction hole is opened between the concentric seal and the eccentric seal.

FIG. 1

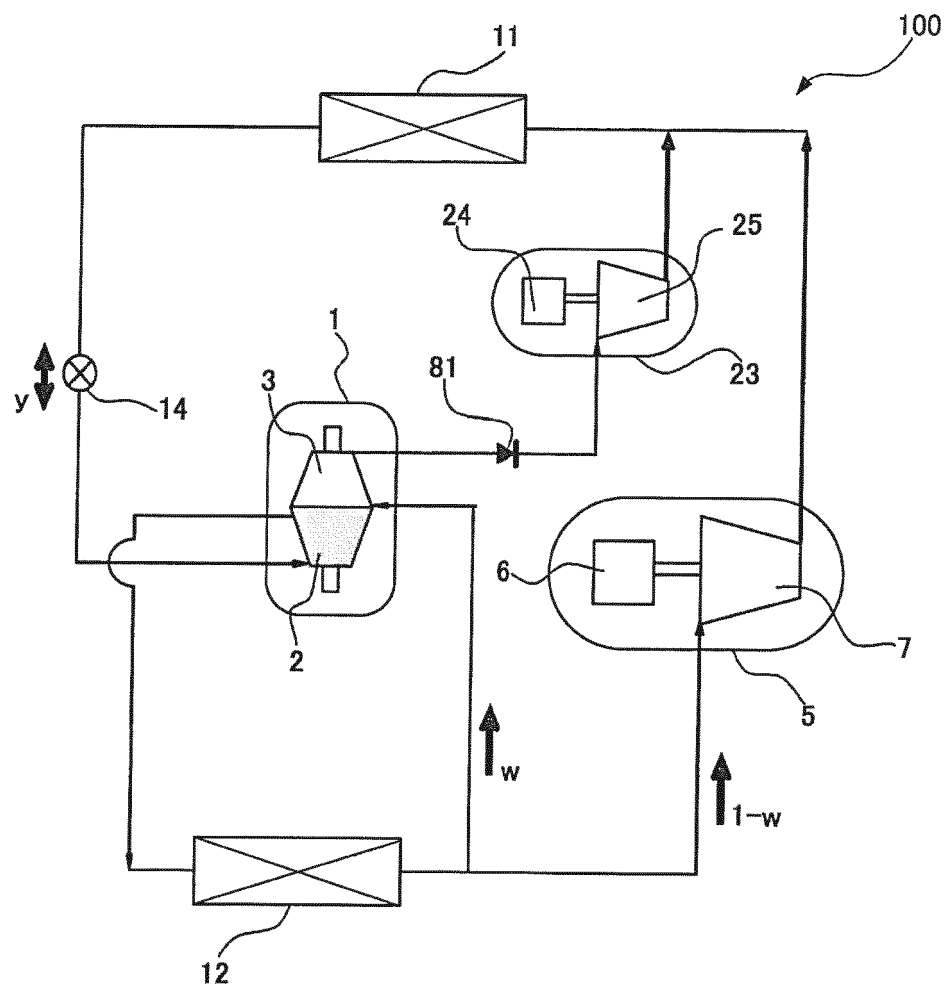


FIG. 2

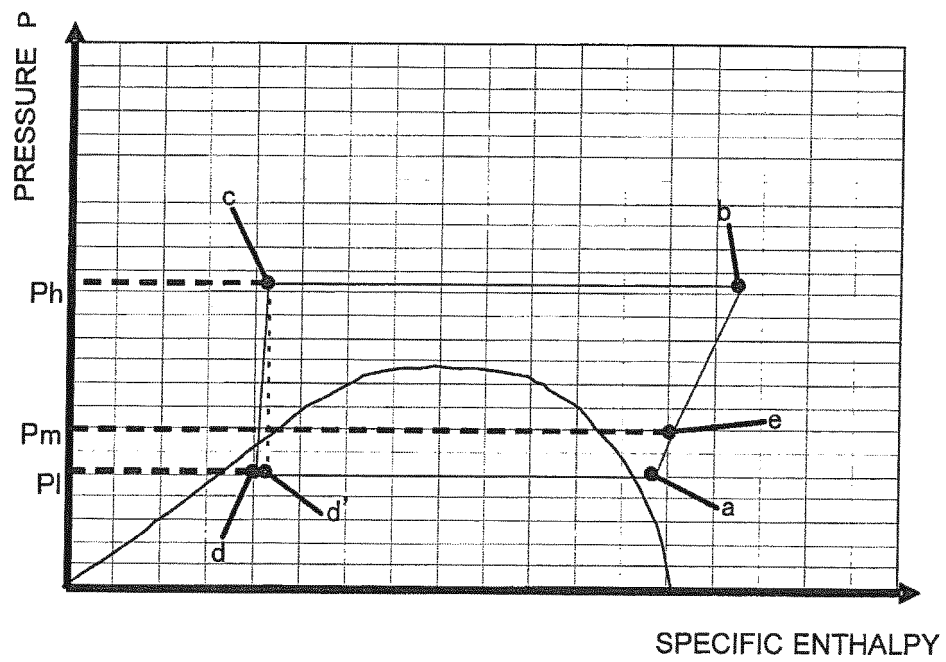


FIG. 3

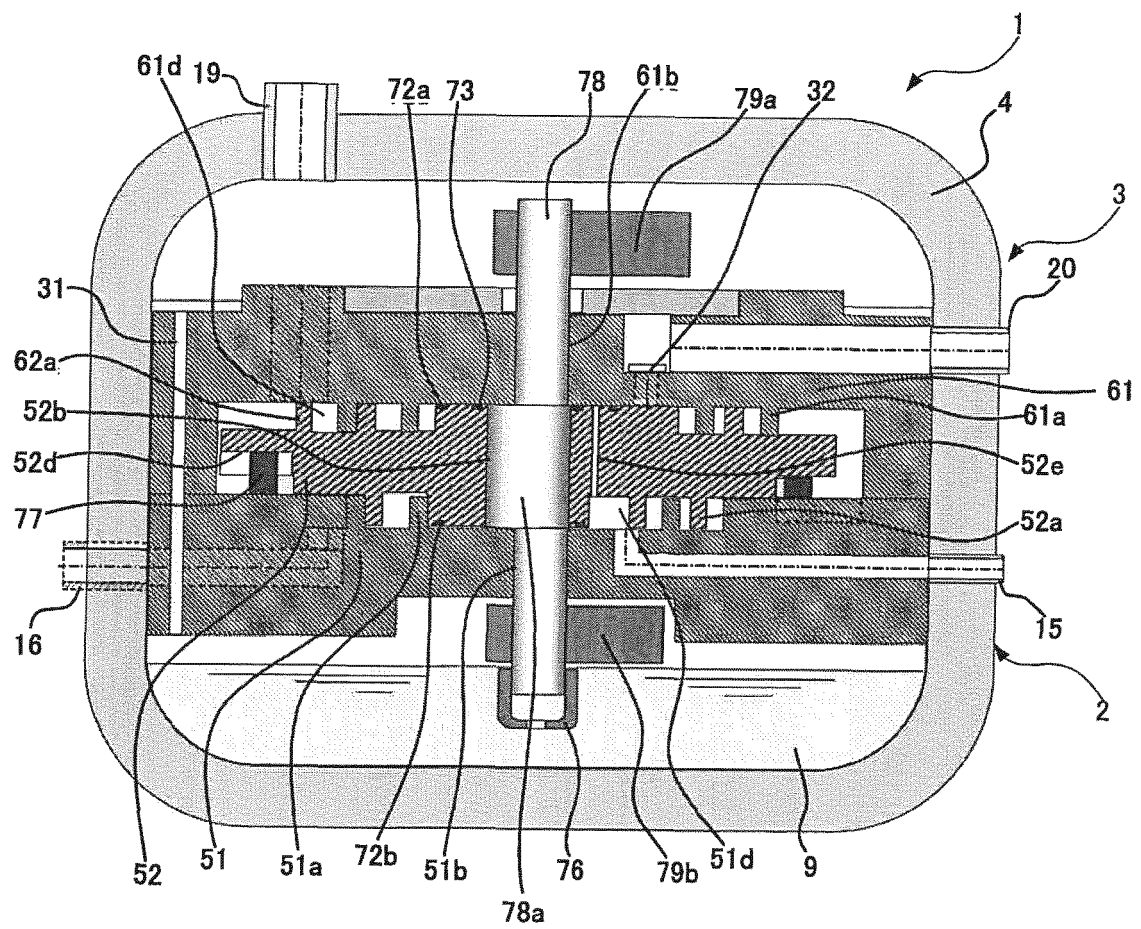


FIG. 4

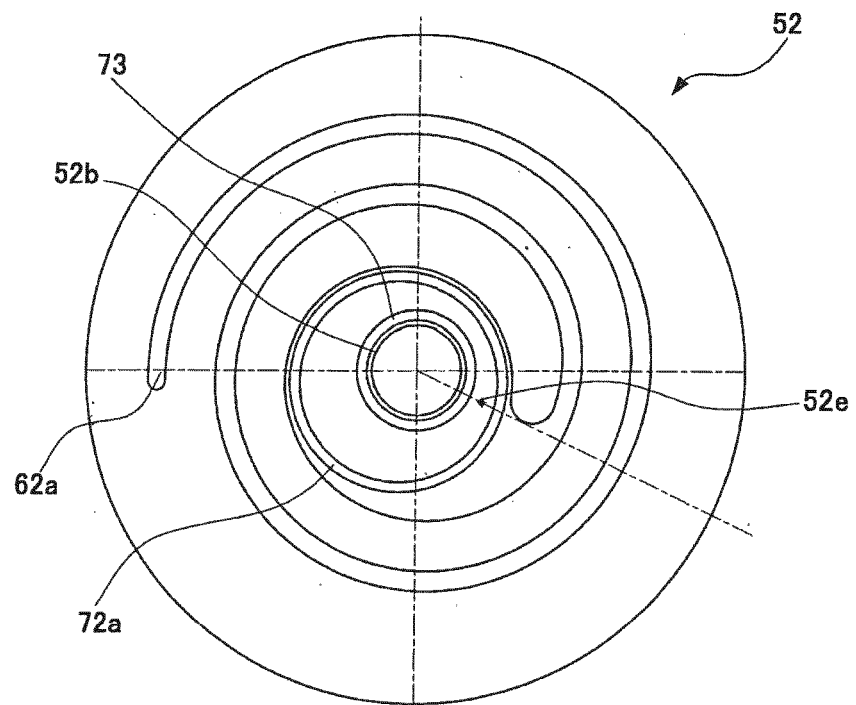


FIG. 5

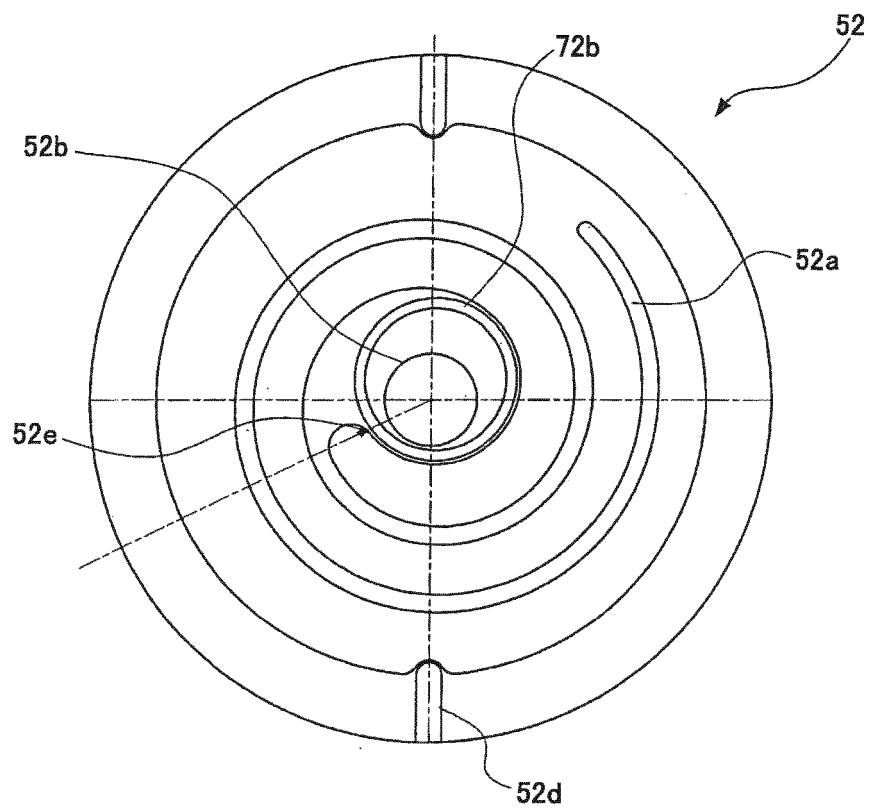


FIG. 6

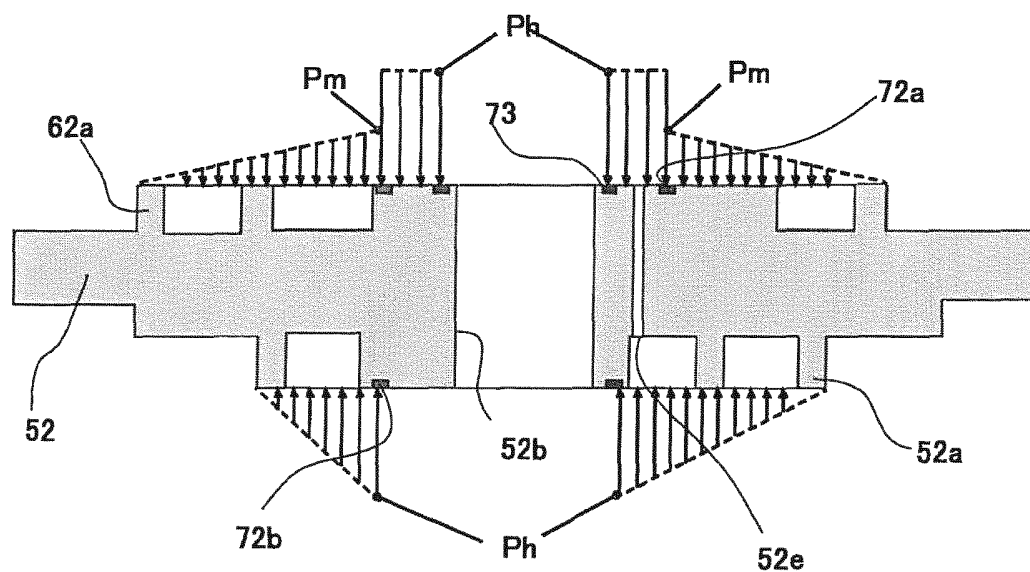


FIG. 7

	COOLING RATED	COOLING INTERMEDIATE	HEATING RATED	HEATING INTERMEDIATE	
PRESSURE BEFORE EXPANSION	10.64	9.21	9.86	7.75	[MPa]
TEMPERATURE BEFORE EXPANSION	20.2	24.9	32.5	26.1	[°C]
PRESSURE AFTER EXPANSION	4.15	5.00	4.29	3.82	[MPa]
TEMPERATURE AFTER EXPANSION	6.8	14.3	8.1	3.6	[°C]
SUCTION PRESSURE	4.09	4.98	3.46	3.61	[MPa]
SUCTION TEMPERATURE	9.6	17.4	-0.2	1.4	[°C]
DISCHARGE PRESSURE	10.85	9.28	9.90	7.76	[MPa]
DISCHARGE TEMPERATURE	96.2	72.1	91.7	69.6	[°C]
σ_{vE}	0.673	0.756	0.476	0.437	
σ_{vEC}	0.093	0.124	0.125	0.126	
C.O.P.th	3.36	4.92	4.17	5.94	
RATIO	100.0%	100.0%	100.0%	100.0%	

FIG. 8

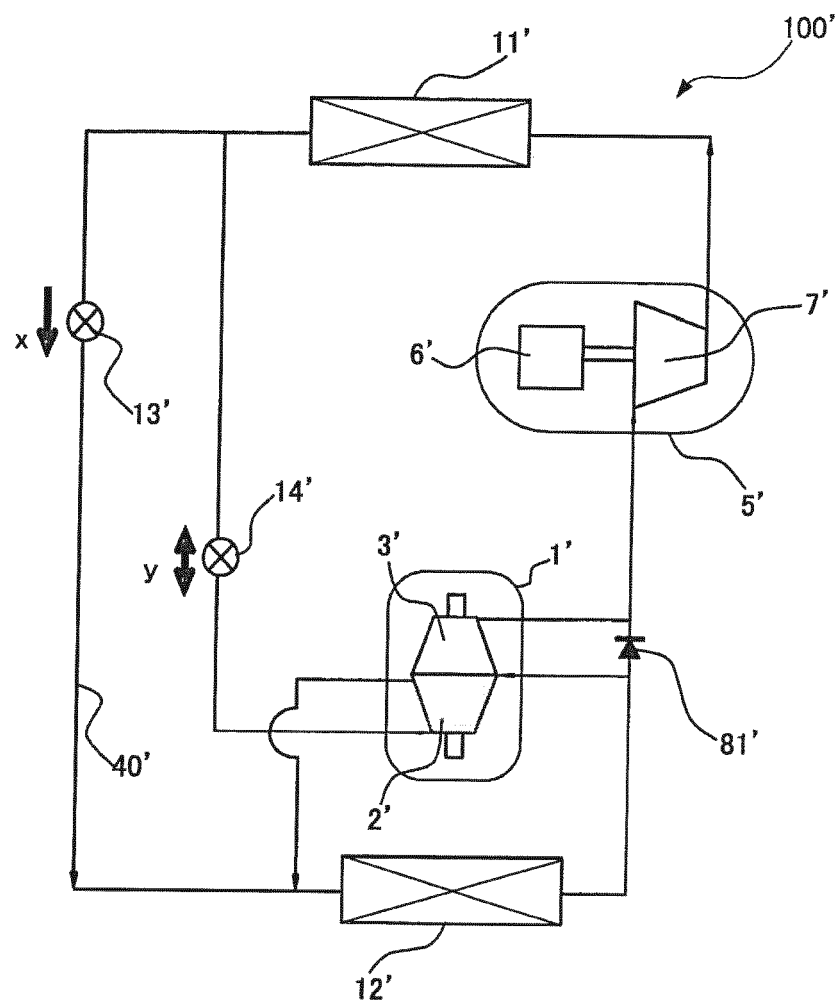


FIG. 9

		COOLING RATED	COOLING INTERMEDIATE	HEATING RATED	HEATING INTERMEDIATE
σ_{vE}		0.673	0.756	0.476	0.437
σ_{vEC}		0.093	0.124	0.125	0.126
AT COOLING RATED	PRE-EXPANSION COEFFICIENT γ	0%	0%	0%	0%
	BYPASS RATIO x	0%	25.2%	26.2%	26.8%
	MEDIUM PRESSURE P_m	4.48 [MPa]	5.20 [MPa]	3.74 [MPa]	3.78 [MPa]
	COP RATIO	113.0%	109.0%	107.1%	105.9%
$\sigma_{vE*}=0.673$ $\sigma_{vEC*}=0.170$					

		COOLING RATED	COOLING INTERMEDIATE	HEATING RATED	HEATING INTERMEDIATE
σ_{vE}		0.673	0.756	0.476	0.437
σ_{vEC}		0.093	0.124	0.125	0.126
AT COOLING INTERMEDIATE	PRE-EXPANSION COEFFICIENT γ	91.8%	0%	0%	0%
	BYPASS RATIO x	0%	0%	1.3%	2.1%
	MEDIUM PRESSURE P_m	4.12 [MPa]	5.31 [MPa]	3.80 [MPa]	3.80 [MPa]
	COP RATIO	101.1%	113.8%	108.6%	106.7%
$\sigma_{vE*}=0.756$ $\sigma_{vEC*}=0.124$					

		COOLING RATED	COOLING INTERMEDIATE	HEATING RATED	HEATING INTERMEDIATE
σ_{vE}		0.673	0.756	0.476	0.437
σ_{vEC}		0.093	0.124	0.125	0.126
AT HEATING RATED	PRE-EXPANSION COEFFICIENT γ	92.2%	26.1%	0%	0%
	BYPASS RATIO x	0%	0%	0%	0.8%
	MEDIUM PRESSURE P_m	4.00 [MPa]	5.08 [MPa]	4.00 [MPa]	3.98 [MPa]
	COP RATIO	97.1%	104.0%	114.1%	113.4%
$\sigma_{vE*}=0.476$ $\sigma_{vEC*}=0.125$					

		COOLING RATED	COOLING INTERMEDIATE	HEATING RATED	HEATING INTERMEDIATE
σ_{vE}		0.673	0.756	0.476	0.437
σ_{vEC}		0.093	0.124	0.125	0.126
AT HEATING INTERMEDIATE	PRE-EXPANSION COEFFICIENT γ	92.6%	39.8%	8.6%	0%
	BYPASS RATIO x	0%	0%	0.0%	0%
	MEDIUM PRESSURE P_m	3.97 [MPa]	5.00 [MPa]	3.92 [MPa]	4.02 [MPa]
	COP RATIO	96.2%	100.9%	111.8%	115.1%
$\sigma_{vE*}=0.437$ $\sigma_{vEC*}=0.126$					

FIG. 10

		COOLING RATED	COOLING INTERMEDIATE	HEATING RATED	HEATING INTERMEDIATE
σ_{vE}		0.673	0.756	0.476	0.437
AT COOLING RATED	FLOW DIVISION RATIO w	18.5%	24.7%	25.1%	25.3%
	MEDIUM PRESSURE P_m	6.55 [MPa]	6.28 [MPa]	5.19 [MPa]	4.61 [MPa]
	$\sigma_{vE*}=0.673$	COP RATIO	113.0%	112.4%	109.8%
$\sigma_{vEC*}=0.5$					

		COOLING RATED	COOLING INTERMEDIATE	HEATING RATED	HEATING INTERMEDIATE
σ_{vE}		0.673	0.756	0.476	0.437
AT COOLING INTERMEDIATE	FLOW DIVISION RATIO w	18.5%	24.7%	25.1%	25.3%
	MEDIUM PRESSURE P_m	6.41 [MPa]	6.43 [MPa]	4.98 [MPa]	4.44 [MPa]
	$\sigma_{vE*}=0.756$	COP RATIO	112.3%	113.8%	108.7%
$\sigma_{vEC*}=0.5$					

		COOLING RATED	COOLING INTERMEDIATE	HEATING RATED	HEATING INTERMEDIATE
σ_{vE}		0.673	0.756	0.476	0.437
AT HEATING RATED	FLOW DIVISION RATIO w	18.5%	24.7%	25.1%	25.3%
	MEDIUM PRESSURE P_m	6.03 [MPa]	5.75 [MPa]	6.01 [MPa]	5.26 [MPa]
	$\sigma_{vE*}=0.476$	COP RATIO	110.4%	107.3%	114.1%
$\sigma_{vEC*}=0.5$					

		COOLING RATED	COOLING INTERMEDIATE	HEATING RATED	HEATING INTERMEDIATE
σ_{vE}		0.673	0.756	0.476	0.437
AT HEATING INTERMEDIATE	FLOW DIVISION RATIO w	18.5%	24.7%	25.1%	25.3%
	MEDIUM PRESSURE P_m	5.88 [MPa]	5.61 [MPa]	5.77 [MPa]	5.46 [MPa]
	$\sigma_{vE*}=0.437$	COP RATIO	109.7%	106.0%	112.9%
$\sigma_{vEC*}=0.5$					

FIG. 11

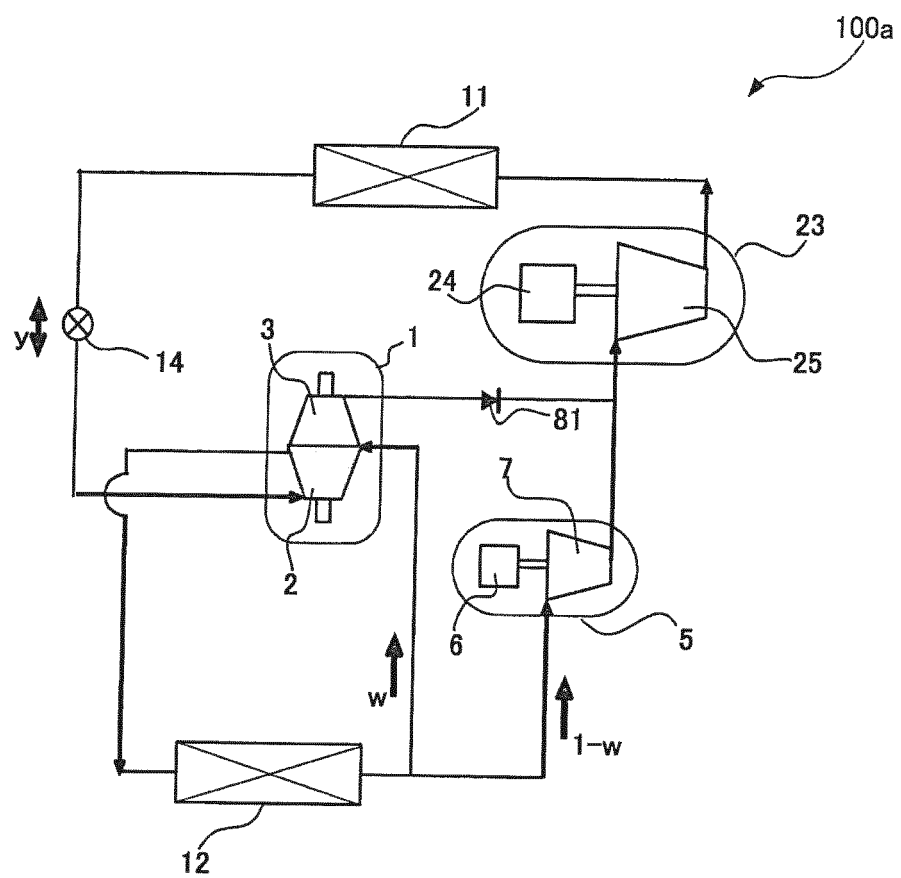
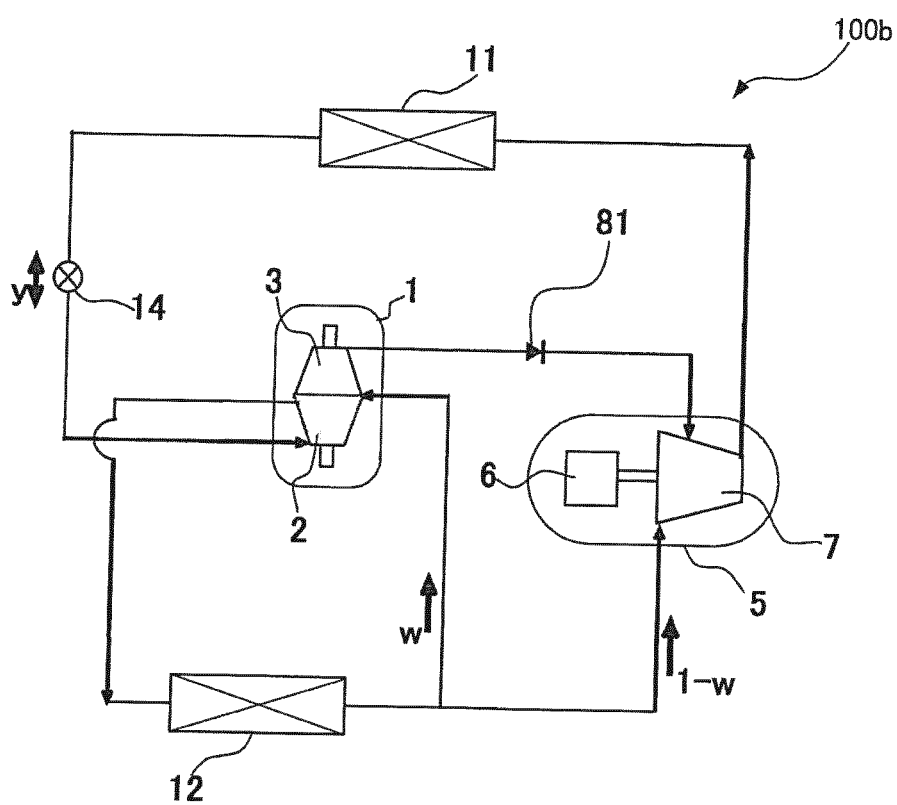


FIG. 12



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2010/000060

A. CLASSIFICATION OF SUBJECT MATTER

F25B1/10(2006.01)i, F01C13/04(2006.01)i, F04C23/02(2006.01)i, F25B1/00(2006.01)i, F25B11/02(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)
F25B1/10, F01C13/04, F04C23/02, F25B1/00, F25B11/02

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 1996-2010
Kokai Jitsuyo Shinan Koho 1971-2010 Toroku Jitsuyo Shinan Koho 1994-2010

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	WO 2009/101818 A1 (Panasonic Corp.), 20 August 2009 (20.08.2009), paragraphs [0010] to [0038]; fig. 1 (Family: none)	1-7
Y	JP 2007-132622 A (Daikin Industries, Ltd.), 31 May 2007 (31.05.2007), paragraphs [0021] to [0051]; fig. 1 (Family: none)	1-7
Y	JP 4-340062 A (Nippondenso Co., Ltd.), 26 November 1992 (26.11.1992), paragraphs [0010] to [0015]; fig. 1 to 3 (Family: none)	1-7

☒ Further documents are listed in the continuation of Box C.

☐ See patent family annex.

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Date of the actual completion of the international search
04 February, 2010 (04.02.10)

Date of mailing of the international search report
16 February, 2010 (16.02.10)

Name and mailing address of the ISA/
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INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2010/000060

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 2005-257103 A (Mitsubishi Electric Corp.), 22 September 2005 (22.09.2005), paragraphs [0002] to [0023]; fig. 1 to 2 (Family: none)	1-7
Y	WO 2007/023599 A1 (Mitsubishi Electric Corp.), 01 March 2007 (01.03.2007), paragraphs [0033] to [0054], [0065] to [0074]; fig. 2 to 5, 7 & US 2009/0064709 A1 & EP 1939547 A1 & WO 2007/023599 A1 & CN 101180505 A	1-7
Y	JP 2008-88854 A (Daikin Industries, Ltd.), 17 April 2008 (17.04.2008), paragraphs [0042] to [0068]; fig. 1 to 4 (Family: none)	1-7
Y	JP 2008-241069 A (Mitsubishi Electric Corp.), 09 October 2008 (09.10.2008), paragraphs [0009], [0010]; fig. 1 (Family: none)	4,5

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

REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 2004150750 A [0004]