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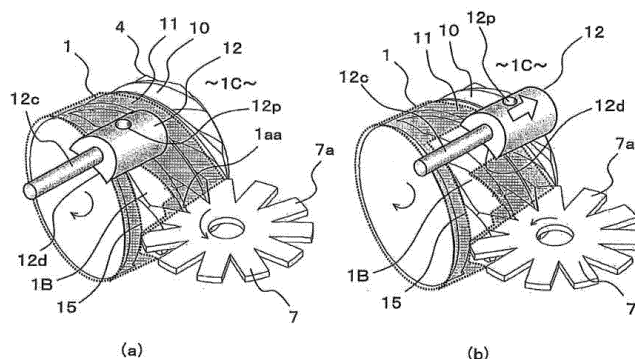
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(54) **SCREW COMPRESSOR AND REFRIGERATION CYCLE DEVICE**

(57) A screw compressor includes an economizer passage (50) provided in a casing (1) and allowing an outside of the casing (1) and a slide groove (14) in which a slide valve (12) is provided to communicate with each other, and an economizer port (12p) provided in the slide valve (12) and allowing the economizer passage (50) to communicate with one of compression chambers (11) in accordance with a position of the slide valve (12). The

slide valve (12) gradually advances a timing of starting discharge by moving from a discharge side toward a suction side. The economizer port (12p) is provided at a position where the economizer port (12p) communicates with a suction-pressure chamber (1C) when the slide valve (12) is positioned at an extreme end on the suction side.

FIG. 4



Description

Technical Field

[0001] The present invention relates to a screw compressor included in a refrigeration cycle for refrigeration, air-conditioning and the like, and having an economizer function, and to a refrigeration cycle apparatus.

Background Art

[0002] There is a hitherto known refrigeration cycle apparatus that performs an economizer operation for the purpose of improving the coefficient of performance (a ratio of the refrigeration capacity to the input to a compressor) of a refrigeration cycle. In the economizer operation, an intermediate cooling device is provided in the refrigeration cycle, and economizer gas from the intermediate cooling device is fed into a compression chamber (see Patent Literature 1, for example).

[0003] Regarding this refrigeration cycle apparatus, an exemplary economizer cycle has been disclosed in which an economizer pipe is connected to an economizer port provided in a casing of a screw compressor, whereby the economizer gas from the intermediate cooling device is fed into the compression chamber.

[0004] Furthermore, there is a hitherto known technique as a screw compressor in which a column-like slide valve that is slidable in a direction of a rotating shaft of a screw rotor is provided on the outer periphery of the screw rotor so as to control the capacity (see Patent Literature 2, for example). Such capacity control using a slide valve is referred to as mechanical capacity control, which is distinguished from rotation-speed capacity control.

[0005] In the capacity control performed by using the slide valve, a suction bypass port that connects one of compression chambers to a suction-pressure space is opened by moving the slide valve toward a discharge side, whereby the timing of completion of suction is delayed, and a small-capacity operation is implemented.

[0006] The screw compressor according to Patent Literature 2 includes an economizer passage through which refrigerant gas from an intermediate cooling device is injected into the compression chamber. Specifically, an economizer port is provided in an inner-surface side of a casing of the screw compressor, and an intermediate suction passage is provided in an outer-surface side of the casing.

[0007] The intermediate suction passage allows the outer surface of the casing and a slide groove in which the slide valve is fitted to communicate with each other. Furthermore, the slide valve has a large-diameter passage and a small-diameter passage provided therein.

[0008] According to Patent Literature 2 employing such a configuration, the intermediate suction passage and the economizer port are allowed to communicate with each other via the small-diameter passage in a small-capacity operation, whereas the intermediate suc-

tion passage and the economizer port are allowed to communicate with each other via the large-diameter passage in a large-capacity operation.

[0009] Specifically, passages having different diameters are provided in the slide valve, and the resistance in the economizer passage is changed by utilizing the movement of the slide valve. Thus, an intermediate pressure is raised in the small-capacity operation in which the pressure difference between the intermediate cooling device and the compression chamber is small, whereby a stable economizer operation is realized.

List of Citations

Patent Literature

[0010]

Patent Literature 1: Japanese Unexamined Patent Application Publication JP 5-010 614 A (page 2 and FIG. 1)

Patent Literature 2: Japanese Unexamined Patent Application Publication JP 4-136 663 A (page 3 and FIG. 1)

Summary of the Invention

Technical Problem

[0011] A hitherto mainstream energy-saving index for refrigerating machines including screw compressors is the coefficient of performance (capacity/power consumption) obtained under rated conditions (conditions at full load: 100 % load). Recently, however, an index obtained under conditions close to the actual operating conditions, for example, IPLV (integrated part load value) specified in the United States, has been attracting attention.

[0012] The total time period per year for which a typical refrigerating machine is operated under rated conditions is very short. Such a refrigerating machine is operated at part load for 90 % or more of the year-round operating time. In most cases of operation at part load, the load ranges particularly from 75 % to 50 % of full load. The full-load operation and the part-load operation are different in the amount of refrigerant that is in circulation and the operating compression ratio and therefore have different coefficients of performance.

[0013] Under such circumstances of the actual operation, the integrated part load value has attracted attention. That is, the integrated part load value is an index that focuses on the coefficient of performance under conditions at part load.

[0014] In the full-load operation, the difference between the high pressure and the low pressure in the refrigeration cycle is large, resulting in a large-capacity operation. In the part-load operation, the difference between the high pressure and the low pressure in the refrigeration cycle is small, resulting in a small-capacity op-

eration. In the full-load operation, performing the economizer operation improves the coefficient of performance.

[0015] In the part-load operation, as the difference between the high pressure and the low pressure becomes smaller, the effect of the economizer operation becomes smaller. Moreover, depending on conditions, the coefficient of performance may be deteriorated. Therefore, if whether to start or stop the economizer operation is determined in accordance with whether the operation is at full load or at part load, the integrated part load value can be improved.

[0016] According to Patent Literature 2, the large-diameter passage and the small-diameter passage that are provided in the slide valve communicate with the compression chamber via the economizer port provided in the casing. Therefore, in the state where the economizer operation is stopped, even if the slide valve is moved so that the passage in the slide valve is separated from the compression chamber, the economizer port provided in the casing remains communicating with the compression chamber.

[0017] Hence, the economizer port forms a volume part (dead volume) that is uselessly compressed from a state under the suction pressure to a state under the discharge pressure, which is a factor that generates a re-expansion loss.

[0018] Furthermore, in a region where the economizer port is provided, the length of a refrigerant leakage passage formed between adjacent ones of the compression chambers is short. Therefore, in a very-small-capacity operation such as an operation at 25 % load, the influence of leakage is not negligible.

[0019] Hence, the diameter of the economizer port needs to be made smaller than the land width between adjacent ones of screw grooves (the width of a raised part between adjacent ones of the screw grooves) so that the economizer port does not form a refrigerant leakage passage between adjacent ones of the compression chambers when the economizer operation is stopped. However, if the diameter of the economizer port is reduced, a satisfactory flow rate cannot be provided in the economizer operation.

[0020] Accordingly, in view of such requirements imposed on the economizer port, there has been a demand for an effective improvement in the design of the economizer port for realization of a high coefficient of performance obtained in a wide range of operating conditions.

[0021] The present invention is to solve the above problems and provides a screw compressor and a refrigeration cycle apparatus that each achieves a high coefficient of performance over a wide range of operating conditions with an economizer port provided at an optimized position.

Solution to the Problem

[0022] A screw compressor according to the present invention includes

a casing having a discharge port and including a cylindrical inner cylinder-surface portion;

a screw rotor rotatably housed in the inner cylinder-surface portion of the casing and having a plurality of screw grooves in an outer periphery;

a gate rotor having tooth portions that are provided on an outer periphery and come into mesh with the screw grooves, the gate rotor defining compression chambers in combination with the screw grooves and the inner cylinder-surface portion;

a suction-pressure chamber provided in the casing and having a suction-pressure atmosphere;

a slide groove provided in the inner cylinder-surface portion of the casing and extending in a direction of a rotating shaft of the screw rotor;

a slide valve provided in the slide groove in such a manner as to be slidable in the direction of the rotating shaft of the screw rotor, the slide valve adjusting a timing of starting discharge;

an economizer passage provided in the casing and allowing an outside of the casing and the slide groove to communicate with each other; and

an economizer port provided in the slide valve and allowing the economizer passage to communicate with one of the compression chambers in accordance with a position of the slide valve.

[0023] The slide valve gradually advances the timing of starting discharge by proceeding from a discharge side toward a suction side. The economizer port is provided at a position where the economizer port communicates with the suction-pressure chamber when the slide valve is positioned at an extreme end on the suction side.

Advantageous Effects of the Invention

[0024] According to the present invention, when the slide valve is positioned at the extreme end on the suction side, a state that is exactly the same as that of a compressor not including the economizer port can be produced. Therefore, no re-expansion loss due to the presence of the economizer port is generated, and there is no increase in the amount of refrigerant leakage through the economizer port. Hence, a screw compressor that realizes a high coefficient of performance over a wide range of operating conditions can be provided.

Brief Description of the Drawings

[0025]

FIG. 1 is a refrigerant circuit diagram of a refrigeration cycle apparatus 200 including a screw compressor 100 according to Embodiment 1 of the present invention.

FIG. 2 is a schematic cross-sectional view of the screw compressor 100 according to Embodiment 1 of the present invention.

FIG. 3 is a cross-sectional view taken along line A-

- A illustrated in FIG. 2.
- FIG. 4 includes perspective views illustrating a part around a discharge port 15 (a housing portion) of the screw compressor 100 according to Embodiment 1 of the present invention.
- FIG. 5 illustrates a part around the discharge port 15 of the screw compressor 100 according to Embodiment 1 of the present invention.
- FIG. 6 illustrates a part around one of economizer ports 12p according to Embodiment 1 of the present invention.
- FIG. 7 illustrates a refrigeration cycle established when an economizer operation is performed in the refrigeration cycle apparatus 200 according to Embodiment 1 of the present invention.
- FIG. 8 is a pressure-specific enthalpy diagram obtained when the refrigeration cycle apparatus 200 according to Embodiment 1 of the present invention is in full-load operation.
- FIG. 9 is a pressure-specific enthalpy diagram obtained when the refrigeration cycle apparatus 200 according to Embodiment 1 of the present invention is in part-load operation with a small difference between the high pressure and the low pressure.
- FIG. 10 includes diagrams illustrating the principle of compression performed by the screw compressor 100 according to Embodiment 1 of the present invention.
- FIG. 11 includes diagrams illustrating the relationship between the angle of screw rotation and the economizer port 12p in the screw compressor 100 according to Embodiment 1 of the present invention.
- FIG. 12 includes diagrams illustrating the relationship between the angle of screw rotation and the economizer port 12p in the screw compressor 100 according to Embodiment 1 of the present invention.
- FIG. 13 illustrates a part around a slide valve 120 included in a screw compressor 100 according to Embodiment 2 of the present invention.
- FIG. 14 includes diagrams illustrating the slide valve 120 according to Embodiment 2 of the present invention that is positioned on a discharge side.
- FIG. 15 includes diagrams illustrating the slide valve 120 according to Embodiment 2 of the present invention that is positioned on a suction side.
- FIG. 15 A includes diagrams illustrating a modification of the positional relationship between the economizer port 12p and an economizer passage 50 in the screw compressor 100 according to Embodiment 2 of the present invention.
- FIG. 16 includes diagrams illustrating a modification of the economizer port 12p or economizer

ports 120p in terms of the diameter thereof.

Description of Embodiments

5 Embodiment 1

[0026] FIG. 1 is a refrigerant circuit diagram of a refrigeration cycle apparatus 200 including a screw compressor 100 according to Embodiment 1 of the present invention. In FIG. 1 and in each of the drawings to be referred to hereinafter, the same reference numerals denote the same or corresponding elements, which applies throughout this specification. Furthermore, embodiments of individual elements described throughout this specification are only exemplary and are not limited thereto.

[0027] The refrigeration cycle apparatus 200 includes a refrigerant circuit in which the screw compressor 100 that is driven by an inverter 101, a condenser 102, a high-pressure section of an intermediate cooling device 103, an expansion valve 104 as a decompression device, and an evaporator 105 are connected to one another in that order with refrigerant pipes.

[0028] The refrigeration cycle apparatus 200 further includes an economizer pipe 107 that branches off from a point between the intermediate cooling device 103 and the expansion valve 104. The economizer pipe 107 is connected to the screw compressor 100 via an intermediate-cooling-device expansion valve 106 and a low-pressure section of the intermediate cooling device 103.

[0029] The condenser 102 cools and condenses gas discharged from the screw compressor 100. The expansion valve 104 throttles and expands liquid obtained through separation performed in the condenser 102. The evaporator 105 evaporates refrigerant discharged from the expansion valve 104. The intermediate cooling device 103 causes high-pressure-side refrigerant in a portion between the condenser 102 and the expansion valve 104 and low-pressure-side refrigerant obtained through the decompression of some of the high-pressure-side refrigerant by the intermediate-cooling-device expansion valve 106 to exchange heat with each other, whereby the intermediate cooling device 103 cools the high-pressure-side refrigerant.

[0030] The refrigeration cycle apparatus 200 further includes a controller 201 that controls the entirety of the refrigeration cycle apparatus, that is, the controller 201 controls the inverter 101, the expansion valve 104, and the intermediate-cooling-device expansion valve 106; the positions of slide valves 12 of the screw compressor 100; the driving and stopping of the below-described economizer operation; and so forth.

Screw Compressor

[0031] Now, the screw compressor 100 according to Embodiment 1 of the present invention will be described with reference to FIGS. 2 to 6.

[0032] FIG. 2 is a schematic cross-sectional view (sec-

tional plan view) of the screw compressor 100 according to Embodiment 1 of the present invention. FIG. 3 is a cross-sectional view taken along line A-A illustrated in FIG. 2.

[0033] The screw compressor 100 includes a casing 1, a screw rotor 4, gate rotors 7, a motor 8 that drives the screw rotor 4 to rotate, the slide valves 12, and so forth. The casing 1 houses the screw rotor 4, the gate rotors 7, the motor 8, the slide valves 12, and so forth. The casing 1 has discharge ports 15 (see FIG. 4 to be referred to below) provided in a housing portion (inner cylinder-surface portion) 1A. Details of the discharge ports 15 will be described later.

[0034] The housing portion 1A, which is a substantially round columnar space, is provided in the casing 1. The screw rotor 4, which has a substantially round columnar shape, is housed in the housing portion 1A. The screw rotor 4 has a fluid suction side at one end thereof and a fluid discharge side at the other end thereof. The screw rotor 4 has a plurality of helical screw grooves 10 in the outer peripheral surface thereof.

[0035] Furthermore, a rotating shaft 9 as a driving shaft is provided in the center of the screw rotor 4 in such a manner as to rotate together with the screw rotor 4. The rotating shaft 9 is rotatably supported by a high-pressure-side bearing 2 and a low-pressure-side bearing 3 that are provided in the casing 1. The motor 8 is provided at an end of the rotating shaft 9 that is on the side of the low-pressure-side bearing 3. The frequency of the motor 8 is controlled by, for example, an inverter (not illustrated).

[0036] The casing 1 has a pair of gate-rotor-supporting chambers 6 provided across the housing portion 1A (i.e., the screw rotor 4) from each other. The gate-rotor-supporting chambers 6 house the respective gate rotors 7, which each have a substantially disc-like shape. The gate rotors 7 are provided on respective gate-rotor supports 5 housed in the respective gate-rotor-supporting chambers 6.

[0037] The gate-rotor supports 5 are each positioned such that a center shaft (rotating shaft) 5b thereof is substantially perpendicular to the rotating shaft 9 of the screw rotor 4. The gate-rotor supports 5 are each rotatably supported by respective bearings 5a that are space apart from each other and opposite each other in the direction of the center shaft 5b.

[0038] In FIG. 3, the arrangement of the gate rotor 7 and the gate-rotor support 5 that are housed in the gate-rotor-supporting chamber 6 provided on the left side of the housing portion 1A and the arrangement of the gate rotor 7 and the gate-rotor support 5 that are housed in the gate-rotor-supporting chamber 6 provided on the right side of the housing portion 1A are in respective orientations that are different from each other by 180° with respect to the rotating shaft 9 of the screw rotor 4.

[0039] The gate rotors 7 each define compression chambers 11 in combination with the housing portion 1A and the screw rotor 4. The gate rotors 7 each have a

plurality of gate-rotor teeth 7a on the outer periphery thereof. The gate-rotor teeth 7a are made to mesh with the screw grooves 10. More specifically, the casing 1 has gate-rotor opening ports 1a extending in the direction of the rotating shaft 9 (see FIG. 2).

[0040] The gate-rotor opening ports 1a each extend along and beyond the inclination of one of the screw grooves 10 that is positioned at the back thereof, and is continuous with a suction wall 1c of the housing portion 1A that defines one of the compression chambers that is positioned at the back thereof.

[0041] The outer periphery of each of the gate rotors 7 is positioned in a corresponding one of the gate-rotor opening ports 1a provided in the casing 1. That is, some of the gate-rotor teeth 7a of the gate rotor 7 are positioned in the housing portion 1A via the gate-rotor opening port 1a and are in mesh with corresponding ones of the screw grooves 10.

[0042] Thus, spaces individually enclosed by the gate rotor 7, the inner wall of the housing portion 1A, and the screw rotor 4 (in other words, the screw grooves 10 that are defined by the gate-rotor teeth 7a of the gate rotor 7 and the housing portion 1A) are provided as the compression chambers 11.

[0043] The casing 1 has two slide grooves 14 in the inner wall thereof. The slide grooves 14 extend in the direction of the rotating shaft 9 of the screw rotor 4. The slide valves 12 are slidably fitted in the respective slide grooves 14. More specifically, the two slide grooves 14 each have a substantially round columnar shape, with a portion of the inner peripheral surface thereof being connected to the housing portion 1A. The two slide grooves 14 are at respective positions that are at 180° from each other with respect to the rotating shaft 9 of the screw rotor 4.

[0044] The slide valves 12 fitted in the respective slide grooves 14 each have a substantially round columnar shape, as with the slide grooves 14, with a part thereof cut off such that a counter surface 1e thereof conforms to the outer peripheral wall of the housing portion 1A. A direct-acting actuator (not illustrated) is connected to the slide valve 12 via a connecting portion 12c. When the direct-acting actuator is driven, the slide valve 12 moves within the slide groove 14 in the direction of the rotating shaft 9 of the screw rotor 4.

Detailed Configuration Around Discharge Port 15

[0045] A detailed configuration around each of the discharge ports 15 of the screw compressor 100 according to Embodiment 1 will now be described.

[0046] FIG. 4 includes perspective views illustrating a part around one of the discharge ports 15 (the housing portion) of the screw compressor 100 according to Embodiment 1 of the present invention. The perspective views in FIG. 4 are seen in the direction of white arrow B illustrated in FIG. 3.

[0047] FIG. 4(a) illustrates a state where the slide valve

12 is positioned on the discharge side. FIG. 4(b) illustrates a state where the slide valve 12 is positioned on the suction side. In FIG. 4, for simple illustration of the configuration around the discharge port 15, elements such as guide portions that are connected to the connecting portion 12c are not illustrated.

[0048] FIG. 5 illustrates a configuration around the discharge port 15 of the screw compressor 100 according to Embodiment 1 of the present invention in a state where the slide valve 12 is positioned at the extreme end on the suction side. The expression "extreme end on the suction side" used herein refers to "extreme end on the suction side" of the range within which the slide valve 12 moves for adjusting the timing of discharge, and does not necessarily coincide with "extreme end on the suction side" of the full slidable range of the slide valve 12.

[0049] That is, the two kinds of "extreme end on the suction side" coincide with each other if the range within which the slide valve 12 moves for adjusting the timing of discharge and the slidable range of the slide valve 12 are the same. Otherwise, "extreme end on the suction side" used herein may be nearer to the discharge side than "extreme end on the suction side" of the slidable range of the slide valve 12. The same applies to an expression "extreme end on the discharge side" to be used hereinafter.

[0050] As illustrated in FIG. 4, the slide valve 12 is fitted in the slide groove 14 (see FIG. 5) in such a manner as to be movable parallel to the rotating shaft 9 (see FIG. 2). The timing of starting discharge is adjusted by changing the position of a discharge-side end face 12d of the slide valve 12. That is, in a part-load operation and when the compression ratio is relatively low, the slide valve 12 slides toward the suction side, thereby advancing the timing of starting discharge.

[0051] In contrast, in a full-load operation or in a part-load operation and when the compression ratio is relatively high, the slide valve 12 slides toward the discharge side, thereby delaying the timing of starting discharge.

[0052] That is, the discharge port 15 is defined by the inner wall of an opening port 1B provided in the casing 1 (more specifically, an opening port provided in the housing portion 1A of the casing 1) and the discharge-side end face 12d of the slide valve 12.

[0053] Here, the discharge port 15 is defined as illustrated in FIG. 5, for facilitating the understanding of the following description. The discharge port 15 has a variable port 16 (the coarsely hatched part in FIG. 5) and a fixed port 17 (the densely hatched part in FIG. 5).

[0054] The variable port 16 is an area of the discharge port 15 that is defined by a screw-rotor central angle $\phi 1$, which also defines the area where the slide valve 12 is positioned. In other words, the variable port 16 is an area of the discharge port 15 that overlaps an area provided by extending the counter surface 1e of the slide valve 12 in the sliding direction.

[0055] The variable port 16 changes the timing of starting discharge in accordance with the position of the dis-

charge-side end face 12d of the slide valve 12. The variable port 16 has an opening area that is variable in accordance with the position of the discharge-side end face 12d of the slide valve 12.

[0056] The fixed port 17 is an area of the discharge port 15 excluding the variable port 16 and is provided between the variable port 16 and the gate rotor 7 (see FIG. 4).

[0057] Now, the position where the slide valve 12 is attached will be described. Herein, as illustrated in FIG. 5, the angle between an end face 1aa of the gate-rotor opening port 1a that is on the side of the slide valve 12 (hereinafter the end face is referred to as gate-rotor-opening surface) and the center of the slide valve 12 is denoted as $\phi 3$, and the position where the slide valve 12 is attached is defined by the angle $\phi 3$.

[0058] The lower limit of $\phi 3$ is set to be larger than 30° , which is the value employed in the known art, so that a wide discharge area can be provided. The upper limit of $\phi 3$ is set to be an angle at which the slide valve 12 does not interfere with components that support the gate rotor provided on the opposite side.

[0059] The upper limit changes with the size of the slide valve 12. For example, if the slide valve 12 has approximately the same size as in the known art (if its width defined by the central angle $\phi 1$ of the screw rotor 4 is about 40°), the upper limit of $\phi 3$ is 100° .

[0060] Here, a rotation-side slide surface of the variable port 16 is denoted by 161, and a non-rotation-side slide surface of the variable port 16 is denoted by 16r. Furthermore, the fixed port 17 has a step in a suction-side end face thereof. Hereinafter, portions of the suction-side end face that are divided by the step are denoted as, in order from the side of the variable port 16, inclined surface 17a and perpendicular surface 17b.

[0061] Furthermore, two portions of the fixed port 17 that are sectioned in the peripheral direction with respect to the step are hereinafter occasionally distinguished from each other as follows: one of the two portions having the inclined surface 17a is denoted as sectioned fixed port 17ax, and the other portion having the perpendicular surface 17b is denoted as sectioned fixed port 17bx. The sectioned fixed port 17bx spreads over an area defined by a screw-rotor central angle $\phi 2$, which is about 10° , for example.

[0062] The screw compressor 100 further includes, in the casing 1, economizer passages 50 (see FIG. 3) for guiding refrigerant gas from the intermediate cooling device 103 to one of the compression chambers 11 (one of the screw grooves 10 that is in a compression step). The economizer passages 50 are provided in the casing 1 in such a manner as to allow the outside of the casing 1 and the respective slide grooves 14 to communicate with each other.

[0063] The economizer pipe 107 is connected to each of the economizer passages 50, whereby the intermediate cooling device 103 and each of the economizer passages 50 are connected to each other. Furthermore, the

screw compressor 100 has economizer ports 12p each provided in the round columnar portion of a corresponding one of the slide valves 12.

[0064] As illustrated in the right part of FIG. 5, the economizer port 12p extends through the slide valve 12 from the outer peripheral surface, which is a slide-contact surface with respect to the slide groove 14, to the inner peripheral surface, which is a slide-contact surface with respect to the screw rotor 4.

Detailed Configuration Around Economizer Port 12p

[0065] A detailed configuration around the economizer port 12p according to Embodiment 1 will now be described.

[0066] FIG. 6 illustrates a configuration around one of the economizer port 12p according to Embodiment 1 of the present invention.

[0067] The economizer passage 50 includes a tubular passage 50a connected to the economizer pipe 107, and a long groove 50b connected to the slide groove 14. The long groove 50b extends along the slide surface of the slide valve 12. The long groove 50b has a length l that corresponds to a range within which the position of the slide valve is controlled in an economizer operation.

[0068] The economizer operation refers to an operation in which the intermediate-cooling-device expansion valve 106 is opened so as to allow the economizer pipe 107 and the screw compressor 100 to communicate with each other, and economizer gas obtained on the downstream side of the low-pressure section of the intermediate cooling device 103 is injected into one of the compression chambers 11 of the screw compressor 100.

[0069] The groove width of the long groove 50b (the length in the peripheral direction of the screw rotor) is larger than a diameter d of the economizer port 12p, as illustrated in the right part of FIG. 6. The economizer-port diameter d is set to be the largest diameter (smaller than or equal to the smallest tooth thickness) that does not allow adjacent ones of the compression chambers 11 of the screw rotor 4 to communicate with each other.

Description of Operations

[0070] Operations according to Embodiment 1 will now be described.

[0071] First, an operation of the refrigerant circuit at full load will be described.

[0072] FIG. 7 illustrates a refrigeration cycle established when the economizer operation is performed in the refrigeration cycle apparatus 200 according to Embodiment 1 of the present invention. FIG. 8 is a pressure-specific enthalpy diagram obtained when the refrigeration cycle apparatus 200 according to Embodiment 1 of the present invention is in full-load operation. The arrows illustrated in FIG. 7 represent flows of the refrigerant.

[0073] The solid lines represent refrigerant liquid, and the broken lines represent refrigerant gas. The state of

the refrigerant at each of the points in FIG. 8 that are denoted by respective reference numerals with parentheses is the state of the refrigerant at a corresponding one of the points of the pipes illustrated in FIG. 7 that are denoted by the same reference numerals.

[0074] Referring to FIGS. 7 and 8, the refrigerant gas (1) discharged from the evaporator 105 and being at a pressure P_s is taken into the screw compressor 100, where the refrigerant gas is compressed to be at a pressure P_d and is discharged. The thus discharged refrigerant gas (5) is subcooled to be in the state (6) by the condenser 102. The high-pressure subcooled liquid (6) flows into the high-pressure section of the intermediate cooling device 103, where the subcooled liquid is further cooled to be in the state (8).

[0075] Some of the high-pressure liquid (8) discharged from the intermediate cooling device 103 branches off into the intermediate-cooling-device expansion valve 106, where the high-pressure liquid is throttled and expanded to be in the state (7) at an intermediate pressure P_m and flows into the low-pressure section of the intermediate cooling device 103 again.

[0076] The other portion of the high-pressure liquid (high-pressure-side refrigerant) (6) that has directly flowed into the high-pressure section of the intermediate cooling device 103 after having been discharged from the condenser 102 exchanges heat with the refrigerant liquid (low-pressure-side refrigerant) that has flowed into the low-pressure section of the intermediate cooling device 103 again via the intermediate-cooling-device expansion valve 106, whereby the high-pressure liquid is further subcooled to be in the state (8). That is, the further subcooling increases the refrigeration effect to be produced by the evaporator 105.

[0077] Meanwhile, the refrigerant liquid (low-pressure-side refrigerant) (7) that has flowed into the low-pressure section of the intermediate cooling device 103 again evaporates by exchanging heat with the high-pressure-side refrigerant, thereby turning into refrigerant gas (7a).

[0078] The refrigerant gas (7a) flows through the economizer pipe 107 and the economizer passages 50 and is injected into screw grooves 10 that are under compression from the economizer ports 12p of the slide valves 12, thereby being mixed with the compressed gas ((2) to (3)).

[0079] In this process, the compressing power changes with the amount and the timing of the gas flowing into the screw compressor 100. Therefore, in terms of improving the coefficient of performance, it is important to increase the refrigeration capacity while suppressing the increase in compressing power as much as possible. Accordingly, there is an optimum intermediate pressure P_m .

[0080] Now, an operation of the refrigerant circuit that is in part-load operation with a small difference between the high pressure and the low pressure will be described.

[0081] FIG. 9 is a pressure-specific enthalpy diagram obtained when the refrigeration cycle apparatus 200 according to Embodiment 1 of the present invention is in

part-load operation with a small difference between the high pressure and the low pressure.

[0082] In the part-load operation and when the difference between the high pressure and the low pressure is small, the pressure difference between the intermediate pressure (at the exit of the intermediate cooling device) and the compression chamber is small as graphed in FIG. 9. Therefore, a relationship of intermediate pressure < compression chamber 11 is transitionally established during the economizer operation, making the operation unstable.

[0083] Moreover, the effect of increasing the refrigeration capacity becomes smaller than the increase in the power generated by the injection of the economizer gas during the compression. Consequently, the coefficient of performance is lowered. Hence, when the difference between the high pressure and the low pressure is small, the intermediate-cooling-device expansion valve 106 illustrated in FIG. 7 is closed so that the economizer operation is not performed.

Description of Operation of Compressor

[0084] An operation of the screw compressor 100 configured as described above will now be described.

[0085] FIG. 10 includes diagrams illustrating the principle of compression performed by the screw compressor 100 according to Embodiment 1 of the present invention.

[0086] As illustrated in FIG. 10, when the screw rotor 4 is rotated on the rotating shaft 9 (see FIG. 2) by the motor 8 (see FIG. 2), the gate-rotor teeth 7a of each of the gate rotors 7 relatively move in corresponding ones of the screw grooves 10. Thus, a series of a suction step, a compression step, and a discharge step that is regarded as one cycle is repeated in each of the compression chambers 11. Herein, the above steps will be described, focusing on one of the compression chambers 11 that is shaded in gray in FIG. 10.

[0087] FIG. 10(a) illustrates a state of the compression chamber 11 that is in the suction step. When the screw rotor 4 is driven by the motor 8 and rotates in the direction of the solid-line arrow, the gate rotor 7 on the lower side in FIG. 10 rotates in the direction of the white arrow with the rotation of the screw rotor 4.

[0088] Meanwhile, the gate rotor 7 on the upper side in FIG. 10 rotates in the direction opposite to the direction of rotation of the gate rotor 7 on the lower side, as indicated by the white arrow. In the suction step, the capacity of the compression chamber 11 is largest, the compression chamber 11 communicates with a low-pressure space of the casing 1 (see FIG. 2), and the compression chamber 11 is filled with a low-pressure refrigerant gas.

[0089] When the screw rotor 4 further rotates, the gate-rotor teeth 7a of each of the two gate rotors 7 sequentially move toward a corresponding one of the discharge ports 15 with the rotation of the screw rotor 4. Thus, the capacity (volume) of the compression chamber 11 is reduced as illustrated in FIG. 10(b).

[0090] Note that the slide valves 12 are not illustrated in FIG. 10 (b). In FIG. 10(b), each of the variable ports 16 is closed by a corresponding one of the slide valves 12, and the capacity of the compression chamber 11 is smaller than in the state illustrated in FIG. 10(a). Thus, the refrigerant gas in the compression chamber 11 is compressed.

[0091] When the screw rotor 4 further rotates, the compression chamber 11 communicates with the discharge port 15 as illustrated in FIG. 10 (c). Thus, the high-pressure refrigerant gas compressed in the compression chamber 11 is discharged to the outside from the discharge port 15. The same compression process is performed on the back side of the screw rotor 4.

[0092] The screw grooves 10 (see FIG. 4) that are open without being covered with the casing 1 (i.e., the inner wall of the housing portion 1A) communicate with the gate rotor 7 and the gate-rotor-supporting chamber 6 that are on the other side (the gate rotor 7 and the gate-rotor-supporting chamber 6 that are hidden in FIG. 4), and each contain a suction-pressure atmosphere.

[0093] Hereinafter, each of spaces (including the gate-rotor-supporting chambers 6) in the casing 1 that are not covered with the inner wall of the housing portion 1A and each contain a suction-pressure atmosphere is defined as suction-pressure chamber 1C.

[0094] Now, the positional relationship between the economizer port 12p and the screw grooves 10 will be described with reference to FIGS. 11 and 12.

[0095] FIGS. 11 and 12 each illustrate the relationship between the angle of screw rotation and the economizer port 12p in the screw compressor 100 according to Embodiment 1 of the present invention. FIG. 11 includes diagrams illustrating a state where the slide valve 12 is positioned on the discharge side (an operating state in which the compression ratio is high, such as a case of the full-load operation).

[0096] FIG. 12 includes diagrams illustrating a state where the slide valve 12 is positioned on the suction side (an operating state in the part-load operation and in which the compression ratio is relatively low). FIG. 11(a) to 11(c) and FIG. 12(a) to 12(c) are each a development of the outer peripheral surface of the screw rotor 4. FIG. 11(d) and FIG. 12(d) are cross-sectional views taken along lines C-C illustrated in FIG. 11(a) and FIG. 12(a), respectively.

[0097] Reference numerals A1 to A9 in FIG. 11 and reference numerals B1 to B11 in FIG. 12 denote the screw grooves 10 at respective angles of screw rotation $\theta A(1)$ to $\theta A(9)$ and $\theta B(1)$ to $\theta B(11)$. That is, FIG. 11 illustrates a process in which the capacity of each of the screw grooves 10 is reduced while the angle of rotation changes in order of $\theta A(1) \Rightarrow \theta A(2) \Rightarrow \theta A(3) \Rightarrow \theta A(4) \Rightarrow \theta A(5) \Rightarrow \theta A(6) \Rightarrow \theta A(7) \Rightarrow \theta A(8) \Rightarrow \theta A(9)$.

[0098] FIG. 12 illustrates a process in which the capacity of each of the screw grooves 10 is reduced while the angle of rotation changes in order of $\theta B(1) \Rightarrow \theta B(2) \Rightarrow \theta B(3) \Rightarrow \theta B(4) \Rightarrow \theta B(5) \Rightarrow \theta B(6) \Rightarrow \theta B(7) \Rightarrow \theta B(8)$.

=> $\theta B(9)$ => $\theta B(10)$ => $\theta B(11)$.

[0099] In FIG. 12, screw grooves B1 and B2 that are hatched with oblique lines are each a screw groove 10 that is in the suction step. That is, the screw grooves B1 and B2 are each at a position that is not completely closed by the gate rotor 7 and the inner wall of the housing portion 1A. In FIGS. 11 and 12, screw grooves A1, A2, A3, and B3 that are shaded are each a screw groove 10 that is in the compression step.

[0100] Screw grooves A4 to A9 and B4 to B11 that are not shaded are each a screw groove 10 that is in the discharge step. The substantial discharge areas in the discharge step correspond to the areas where the discharge port 15 faces the screw grooves 10, and are each hatched with a grid pattern in FIGS. 11 and 12.

Case of Full-Load Operation

[0101] The positional relationship between the economizer port 12p and the screw grooves 10 in the full-load operation will now be described with reference to FIG. 11.

[0102] In the full-load operation, the economizer operation is performed. In the economizer operation, the slide valve 12 is moved toward the discharge side, as illustrated in FIG. 11(d), to a position where the variable port 16 is completely closed as illustrated in FIG. 11 (a) to FIG. 11(c). Furthermore, the economizer passage 50 provided in the casing 1 and the economizer port 12p communicate with each other.

[0103] As illustrated in FIG. 11 (a), the economizer port 12p starts to communicate with the screw groove A1 that is in a state immediately after the completion of suction and is at a low pressure. The economizer port 12p moves over the screw groove A2 and then the screw groove A3 that are in the compression step. While the economizer port 12p moves over the screw grooves A2 and A3, economizer gas is injected into each of these screw grooves 10 from the economizer port 12p because of the pressure difference between the intermediate pressure P_m and the screw groove 10.

[0104] When the economizer port 12p is positioned over a screw groove 10 that is at a high pressure, the intermediate pressure is raised and, accordingly, the effect of increasing the capacity by performing the economizer operation (the degree of subcooling at point (8) in FIG. 8) becomes small. Hence, in this case, the economizer gas is injected into a screw groove 10 that is at as low pressure as possible.

[0105] If a large amount of refrigerant gas is injected into a screw groove 10 that is in the suction step, the amount of refrigerant circulation is reduced, which triggers a reduction in the refrigeration capacity. Therefore, the economizer port 12p is allowed to communicate with the screw groove 10 when suction is almost complete.

[0106] That is, the economizer port 12p first communicates with the screw groove A1 that is at the beginning of compression as illustrated in FIG. 11 (a), passes over the screw grooves A2 and A3 that are in the midst of the

compression step, and is completely closed to the screw groove at the screw groove A4. This cycle is repeated.

[0107] Even in the part-load operation, if the difference between the high pressure and the low pressure is relatively large and an economizer effect can be produced, the economizer operation is performed by allowing the economizer passage 50 provided in the casing 1 and the economizer port 12p to communicate with each other. In the economizer operation performed in the part-load operation, the slide valve 12 is moved further toward the suction side than in the full-load operation or to the same slide position as in the full-load operation.

Case of Part-Load Operation With Small Difference Between High Pressure

and Low Pressure

[0108] The positional relationship between the economizer port 12p and the screw grooves 10 in the part-load operation with a small difference between the high pressure and the low pressure will now be described with reference to FIG. 12.

[0109] When the difference between the high pressure and the low pressure is small in the part-load operation, the economizer operation is stopped. To stop the economizer operation, the slide valve 12 is moved toward the suction side as illustrated in FIG. 12(d), and the economizer port 12p is positioned in an area where the inner wall of the housing portion 1A is not present (in the suction-pressure chamber 1C) as illustrated in FIG. 4(b). In this state, the economizer passage 50 provided in the casing 1 and the economizer port 12p do not communicate with each other.

[0110] During the economizer operation, the economizer port 12p is constantly in communication with the suction-pressure chamber 1C. Hence, when the difference between the high pressure and the low pressure is small in the part-load operation, the operation progresses from the suction step to the discharge step while the economizer port 12p is prevented from acting on the screw grooves 10.

[0111] According to Patent Literature 2, as described above, a re-expansion loss occurs when the economizer port passes over any screw grooves while the economizer operation is stopped. In contrast, according to Embodiment 1, the economizer port 12p is not involved at all in the operation in which the economizer operation is not performed. Therefore, the deterioration in the performance due to the re-expansion loss can be prevented.

[0112] Furthermore, in the part-load operation, since the capacity is small, the influence of leakage that may occur between adjacent ones of the compression chambers is prominent. In the configuration according to Embodiment 1, however, the economizer port 12p is not involved at all in the operation in which the economizer operation is not performed. Therefore, the leakage between adjacent ones of the screw grooves 10 through

the economizer port 12p can be prevented.

[0113] To summarize, Embodiment 1 produces the following advantageous effects. In Embodiment 1, the economizer port 12p is provided at a position where the economizer port 12p communicates with one of the compression chambers 11 and the economizer passage 50 when the slide valve 12 is positioned at the extreme end on the discharge side.

[0114] Furthermore, the economizer port 12p is provided at a position where the economizer port 12p communicates with the suction-pressure chamber 1C when the slide valve 12 is positioned at the extreme end on the suction side. In such a configuration, in the full-load operation or the like in which the economizer effect is significant and the difference between the high pressure and the low pressure is large, the coefficient of performance can be improved by performing the economizer operation.

[0115] On the other hand, in the part-load operation in which the pressure difference is small and the coefficient of performance is not expected to be improved by the economizer operation, the economizer operation is stopped, whereby a high coefficient of performance can be achieved without the occurrence of re-expansion loss and leakage loss that may occur with the presence of the economizer port 12p. That is, according to Embodiment 1, the screw compressor 100 and the refrigeration cycle apparatus 200 can realize a high coefficient of performance in a wide range of operating conditions.

Embodiment 2

[0116] In Embodiment 1, the angle ϕ_3 between the gate-rotor-opening surface 1aa and the center of the slide valve 12 that is at the attached position is set within a range of $30^\circ < \phi_3 < 90^\circ$. In Embodiment 2, a case where $\phi_3 = 30^\circ$ will be discussed.

[0117] FIG. 13 illustrates a part around a slide valve 120 included in a screw compressor 100 according to Embodiment 2 of the present invention.

[0118] Embodiment 2 will be described focusing on differences from Embodiment 1. Elements that are not described in Embodiment 2 are the same as those described in Embodiment 1.

[0119] In Embodiment 2, as illustrated in FIG. 13, the center of the slide valve 120 is positioned such that $\phi_3 = 30^\circ$. That is, a fixed port 170 according to Embodiment 2 includes only the sectioned fixed port 17bx described in Embodiment 1 and does not include the sectioned fixed port 17ax.

[0120] Now, the relationship between economizer ports 120p and the screw grooves 10 according to Embodiment 2 will be described.

[0121] FIG. 14 includes diagrams illustrating the slide valve 120 according to Embodiment 2 of the present invention that is positioned on the discharge side. FIG. 15 includes diagrams illustrating the slide valve 12 according to Embodiment 2 of the present invention that is po-

sitioned on the suction side. The hatched parts and the like in FIGS. 14 and 15 are in the same states, respectively, as those illustrated in FIGS. 11 and 12, and description thereof is therefore omitted.

[0122] As is obvious from the comparison between FIG. 14 illustrating Embodiment 2 and FIG. 11 illustrating Embodiment 1, the land width between adjacent ones of the screw grooves 10 within the angular range in which the variable port 16, or the slide valve 120, is provided is smaller than in Embodiment 1. Hence, in Embodiment 2, a plurality of economizer ports 120p are provided along the inclination of the screw grooves 10, whereby a satisfactory passage area is secured.

[0123] Note that the economizer ports 120p are provided at a position where the economizer ports 120p communicate with one of the compression chambers 11 and the economizer passage 50 when the slide valve 120 is positioned at the extreme end on the discharge side, as in Embodiment 1. Furthermore, the economizer ports 120p are provided at a position where the economizer ports 120p communicate with the suction-pressure chamber 1C when the slide valve 120 is positioned at the extreme end on the suction side, as in Embodiment 1.

Case of Full-Load Operation

[0124] In the full-load operation, as illustrated in FIG. 14(d), the economizer passage 50 provided in the casing 1 and the economizer ports 120p communicate with each other, and the economizer operation is performed, as in Embodiment 1.

[0125] In Embodiment 1, as illustrated in FIG. 11, the variable port 16 is closed by the slide valve 120. In Embodiment 2, the slide valve 12 opens the variable port 16 so that a satisfactory discharge area is secured. As illustrated in FIG. 14(a), the economizer ports 120p start to communicate with the screw groove A1 that is in a state a little before the completion of suction and is at a low pressure.

[0126] Subsequently, the economizer ports 120p move over the screw groove A2 and then the screw groove A3 that are in the compression step, and economizer gas is injected into each of these screw grooves 10 from the economizer ports 120p because of the pressure difference between the intermediate pressure P_m and the screw groove 10.

[0127] Here, suppose that the economizer ports 120p each need to have a diameter that does not allow adjacent ones of the compression chambers 11 to communicate with each other, the diameter becomes small. In that case, the intermediate pressure is raised, and the effect of increasing the capacity by performing the economizer operation (the degree of subcooling at point (8) in FIG. 8) is reduced. Hence, in Embodiment 2, the plurality of economizer ports 120p are provided, whereby a satisfactory amount of flow is secured.

Case of Part-Load Operation With Small Difference Between High Pressure

and Low Pressure

[0128] The relationship between the economizer ports 120p and the screw grooves 10 in the part-load operation with a small difference between the high pressure and the low pressure will now be described with reference to FIG. 15.

[0129] When the difference between the high pressure and the low pressure is small in the part-load operation, the economizer operation is stopped. To stop the economizer operation, the slide valve 120 is moved toward the suction side as illustrated in FIG. 15(d), and the economizer ports 120p are positioned at the edge of the suction wall 1c of the housing portion 1A. Furthermore, the economizer passage 50 provided in the casing 1 and the economizer ports 120p provided in the slide valve 120 do not communicate with each other.

[0130] While the economizer operation is stopped, the economizer ports 120p move over the screw groove B1 and the screw groove B2 that are in the suction step and over only a small portion of the screw groove B3 that is at the beginning of compression. The increase in the pressure in each of the screw grooves 10 during this process is small and does not significantly affect the occurrence of re-compression loss and leakage loss.

[0131] To summarize, Embodiment 2 also produces the advantageous effects produced in Embodiment 1. Furthermore, the following advantageous effects are produced. In Embodiment 2 (in which the length of slide is smaller than in Embodiment 1 and the handleable range of compression ratio is therefore smaller than in Embodiment 1), the slide valve 120 is positioned nearer to the gate rotor 7 than in Embodiment 1. Thus, a space for providing other structural components and structures can be provided near a position at a screw central angle of 90°.

[0132] Since the slide valve 120 is positioned nearer to the gate rotor 7 than in Embodiment 1, the land width between adjacent ones of the screw grooves 10 within the angular range in which the slide valve 120 is provided is smaller than in Embodiment 1 while the advantageous effects described above are produced.

[0133] In such a case, if the diameter of the economizer port is designed to be smaller than the land width, a satisfactory passage area cannot be secured with one economizer port 120p. In Embodiment 2, however, since the plurality of economizer ports 120p are provided along the inclination of the screw grooves 10, a satisfactory passage area can be secured.

[0134] In Embodiment 1 and Embodiment 2, when the economizer operation is not performed, the economizer passage 50 and the economizer port 12p or the economizer ports 120p are not allowed to communicate with each other. Alternatively, as illustrated in FIG. 15A (when the slide valve is positioned on the suction side), if the

economizer pipe 107 is closed by the intermediate-cooling-device expansion valve 106 or the like, there is no chance that the economizer gas may leak toward the suction side and prevent the suction gas from flowing into the compression chambers 11.

[0135] Hence, in terms of standardization of components and so forth, the economizer passage 50 and the economizer port 12p, 120 p may be allowed to communicate with each other. Even in that case, the same advantageous effects are produced.

[0136] FIG. 16 includes diagrams illustrating a modification of the economizer port 12p or the economizer ports 120p in terms of the diameter thereof. Part (a) is a development illustrating the inner wall of the housing portion 1A and the outer peripheral surface of the screw rotor 4. Part (b) is a cross-sectional view taken along line d-d illustrated in part (a).

[0137] In Embodiment 1 and Embodiment 2, the economizer port 12p and the economizer ports 120p each have a diameter that does not allow adjacent ones of the compression chambers 11 to communicate with each other. If the economizer port 12p or the economizer ports 120p are used only in the economizer operation and the refrigerant to be injected flows as illustrated by white arrows in FIG. 16(b), leakage of the refrigerant between adjacent ones of the compression chambers does not occur.

[0138] Hence, depending on the operating range of use, the economizer port 12p or the economizer ports 120p may be wider than the land width as illustrated in FIG. 16(a). Such a case also produces the advantageous effects produced in Embodiment 1 and Embodiment 2.

[0139] The land width of the screw rotor 4 becomes smaller as the position of the slide valve 12 becomes closer to the gate rotor 7. Therefore, in a case where the diameter of the economizer port is designed to be smaller than the land width, the diameter of the economizer port can be made larger and the flow rate can be controlled more stably by setting the position of the slide valve 12 such that $\phi 3$ falls within a range from a value larger than 30°, which is the value employed in the known art, up to about 100° at which the slide valve 12 does not interfere with components that support the gate rotor 7 on the other side.

[0140] Thus, the screw compressor 100 can exhibit a high coefficient of performance in a wide range of operating conditions from a high compression ratio to a low compression ratio and can operate at a high efficiency throughout the year.

List of Reference Signs

[0141]

1	casing
1A	housing portion (inner cylinder-surface portion)
1B	opening port
1C	suction-pressure chamber

1a	gate-rotor opening port	
1aa	gate-rotor-opening surface	
1c	suction wall	
1e	counter surface	
2	high-pressure-side bearing	5
3	low-pressure-side bearing	
4	screw rotor	
5	gate-rotor support	
5a	bearing	
5b	center shaft	10
6	gate-rotor-supporting chamber	
7	gate rotor	
7a	gate-rotor tooth	
8	motor	
9	rotating shaft	15
10	screw groove	
11	compression chamber	
12	slide valve	
12c	connecting portion	
12d	inclined surface (discharge-side end face)	20
12p	economizer port	
14	slide groove	
15	discharge port	
16	variable port	
161	slide surface (rotation-side slide surface)	25
16r	non-rotation-side slide surface	
17	fixed port	
17a	inclined surface	
17ax	sectioned fixed port	
17b	perpendicular surface	30
17bx	sectioned fixed port	
50	economizer passage	
50a	tubular passage	
50b	long groove	
100	screw compressor	35
101	inverter	
102	condenser	
103	intermediate cooling device	
104	expansion valve	
105	evaporator	40
106	intermediate-cooling-device expansion valve	
107	economizer pipe	
120	slide valve	
120p	economizer port	
170	fixed port	45
200	refrigeration cycle apparatus	
201	controller	

Claims

1. A screw compressor comprising:

a casing having a discharge port and including a cylindrical inner cylinder-surface portion;
a screw rotor housed in the inner cylinder-surface portion of the casing, being rotatable, and having a plurality of screw grooves in an outer

periphery thereof;

a gate rotor having tooth portions that are provided on an outer periphery and come into mesh with the screw grooves, the gate rotor defining compression chambers in combination with the screw grooves and the inner cylinder-surface portion;

a suction-pressure chamber provided in the casing and having a suction-pressure atmosphere;
a slide groove provided on the inner cylinder-surface portion of the casing and extending in a direction of a rotating shaft of the screw rotor;
a slide valve provided in the slide groove and being slidable in the direction of the rotating shaft of the screw rotor, and the slide valve configured to adjust a timing of starting discharge;
an economizer passage provided in the casing and allowing an outside of the casing and the slide groove to communicate with each other; and

an economizer port provided in the slide valve and allowing the economizer passage to communicate with one of the compression chambers in accordance with a position of the slide valve, wherein the slide valve gradually advances the timing of starting discharge as the slide valve proceeds toward a suction side from a discharge side, and

wherein the economizer port is provided at a position where the economizer port communicates with the suction-pressure chamber when the slide valve is positioned at an extreme end on the suction side.

2. The screw compressor of claim 1, wherein the economizer port is provided at a position where the economizer port communicates with one of the compression chambers and the economizer passage when the slide valve is positioned at an extreme end on the discharge side.

3. The screw compressor of claim 1 or 2, wherein pairs of the gate rotor and the slide valve are provided, and a position of a center of each slide valve falls within a range between 30° and 100° of a central angle of the screw rotor, the range is defined by the position of the center of each slide valve and a position of an end face of a gate-rotor opening port, that is on a side of the slide valve, the gate-rotor opening port being one of openings provided in the casing and in which some of the tooth portions of a corresponding one of the gate rotors are positioned.

4. The screw compressor of any one of claims 1 to 3, wherein the economizer port is one of a plurality of economizer ports provided along an inclination of the screw grooves.

5. The screw compressor of any one of claims 1 to 4, further comprising a motor connected to the screw rotor via a driving shaft and that rotates the screw rotor, wherein the motor is driven by an inverter. 5
6. A refrigeration cycle apparatus comprising:
- a refrigerant circuit in which, in order of appearance, the screw compressor of any one of claims 1 to 5, a condenser, a high-pressure section of an intermediate cooling device, a decompression device, and an evaporator are connected to one another with refrigerant pipes; and an economizer pipe branched from a point between the intermediate cooling device and the decompression device, and connected to the economizer passage of the screw compressor via an intermediate-cooling-device expansion valve and a low-pressure section of the intermediate cooling device. 10 15 20

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FIG. 1

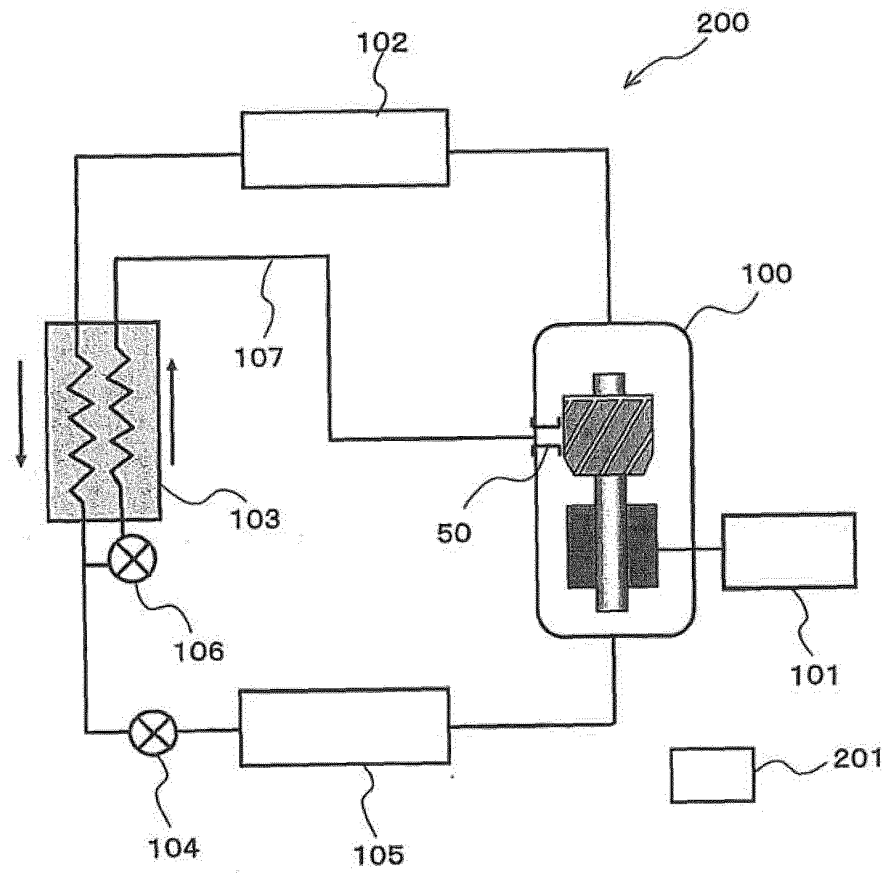


FIG. 2

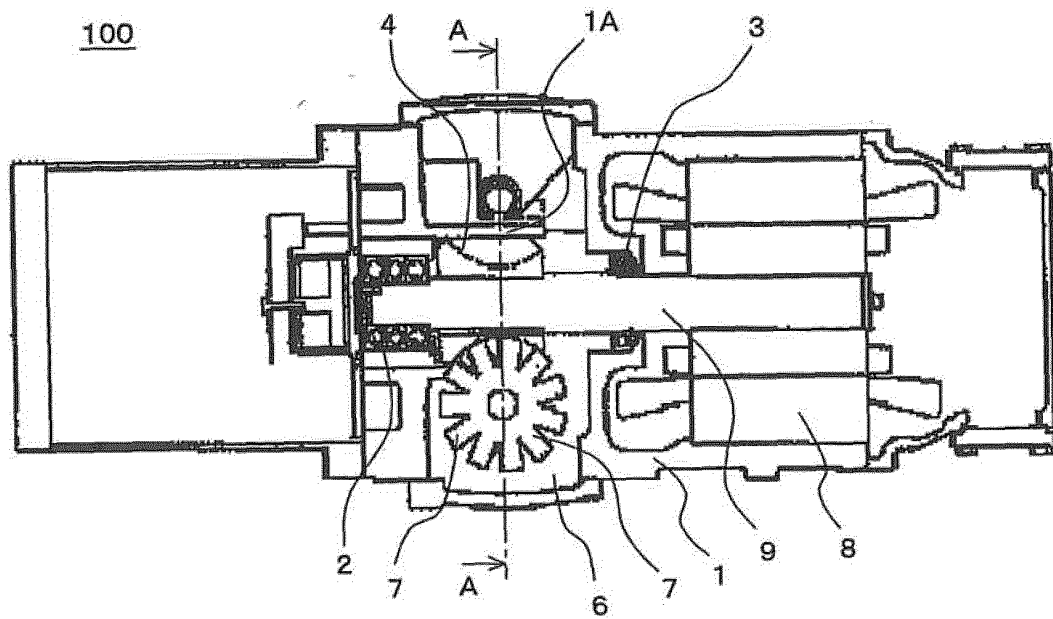


FIG. 3

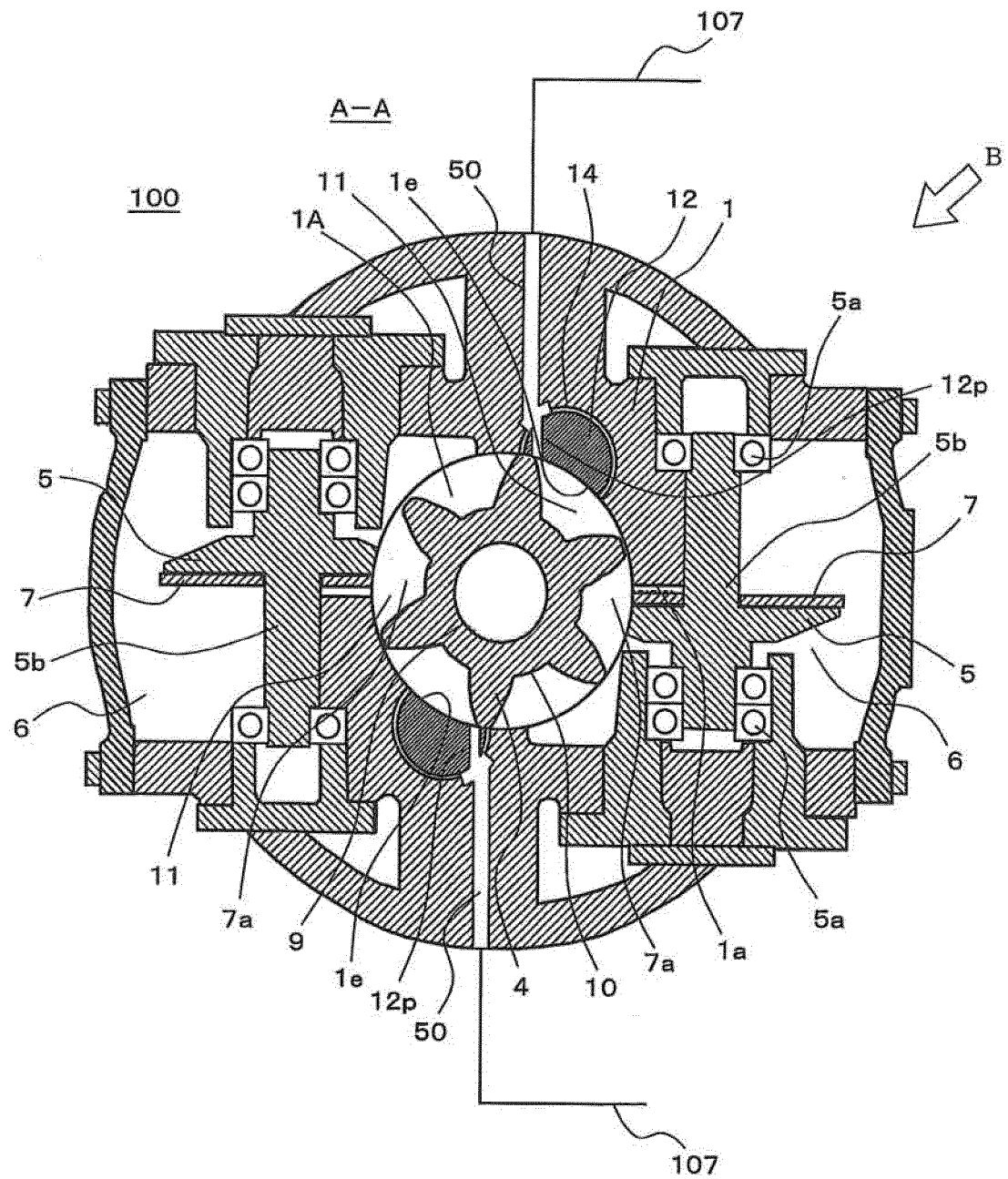


FIG. 4

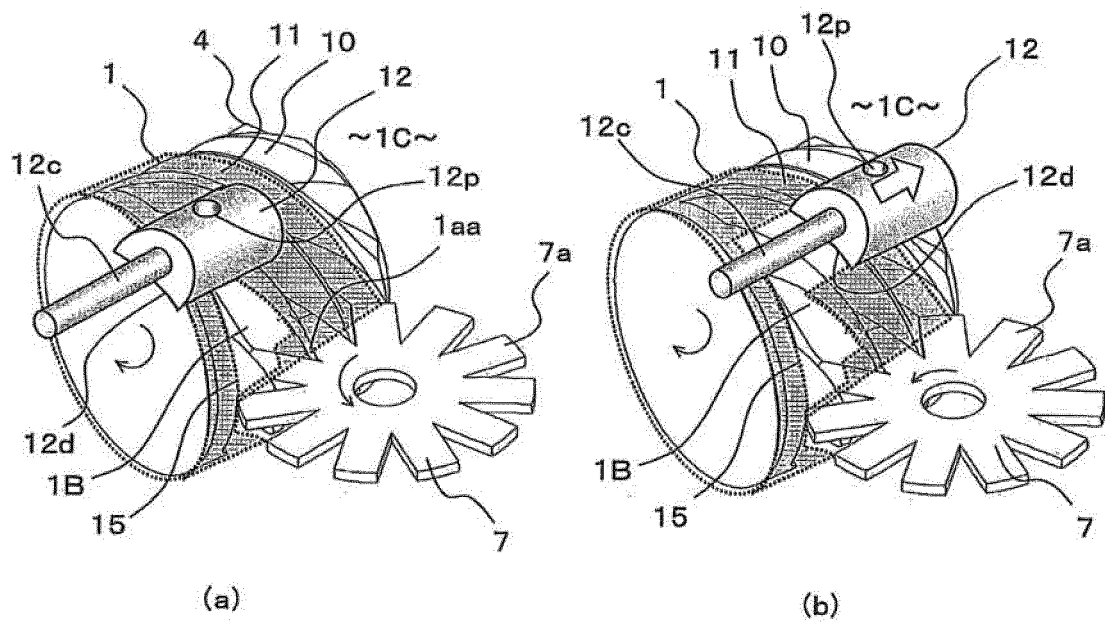


FIG. 5

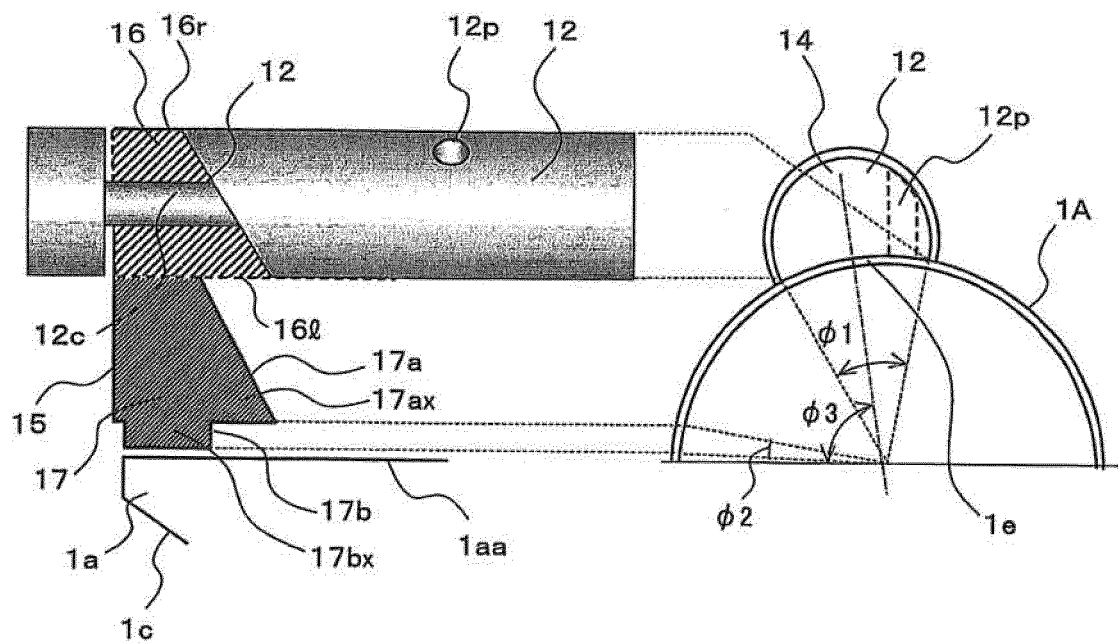


FIG. 6

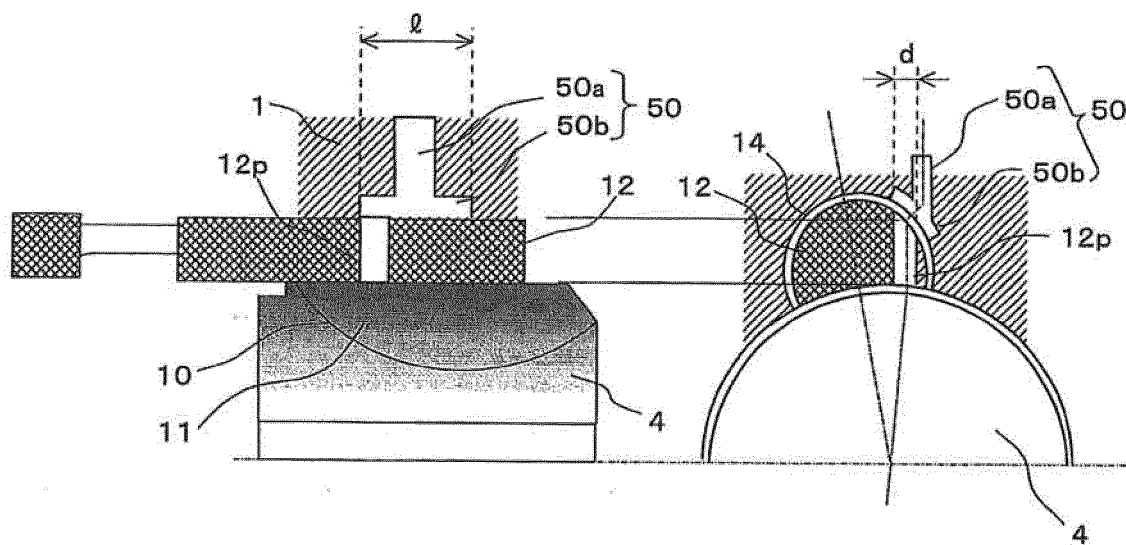


FIG. 7

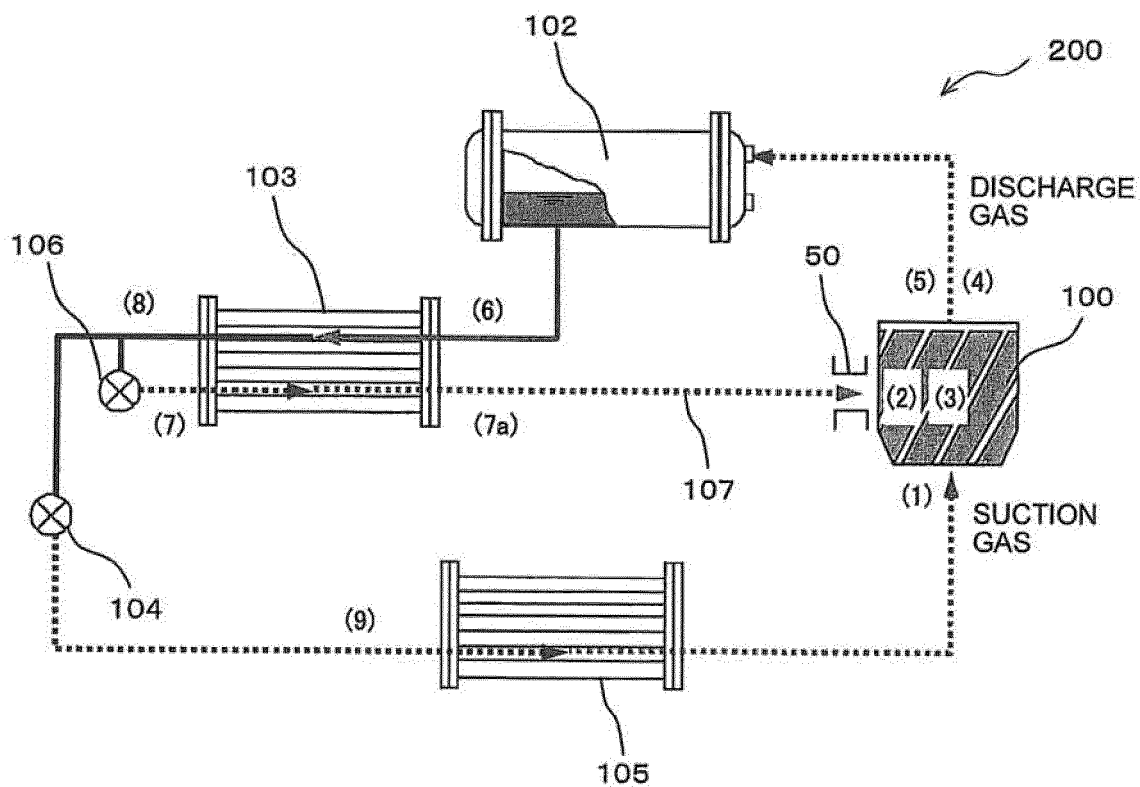


FIG. 8

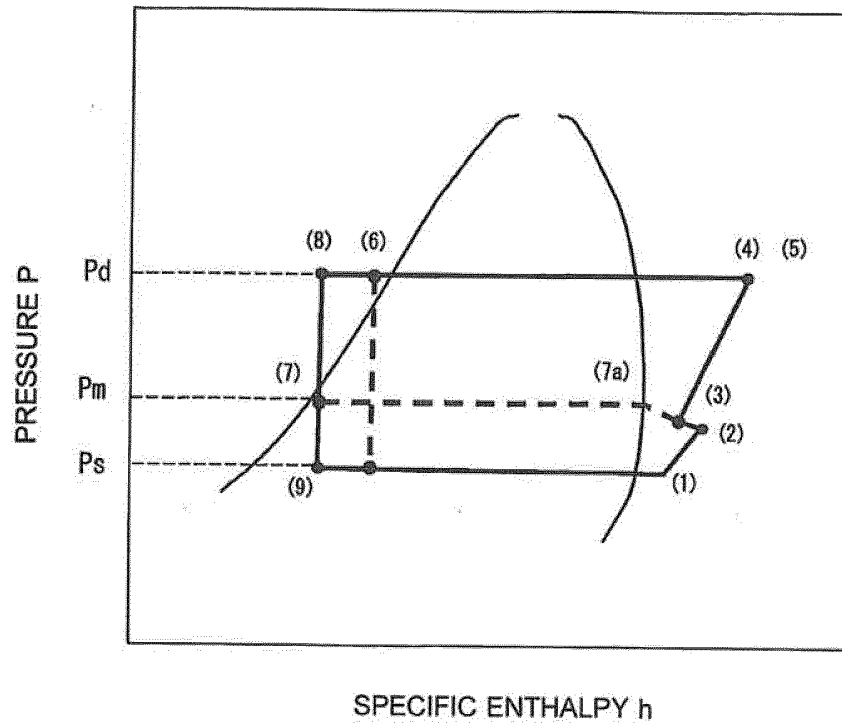


FIG. 9

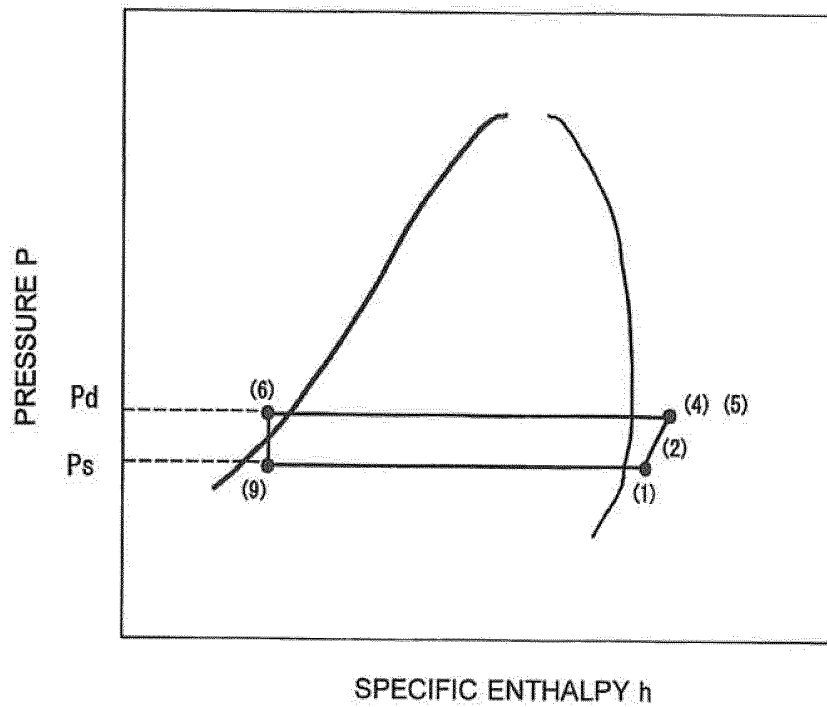


FIG. 10

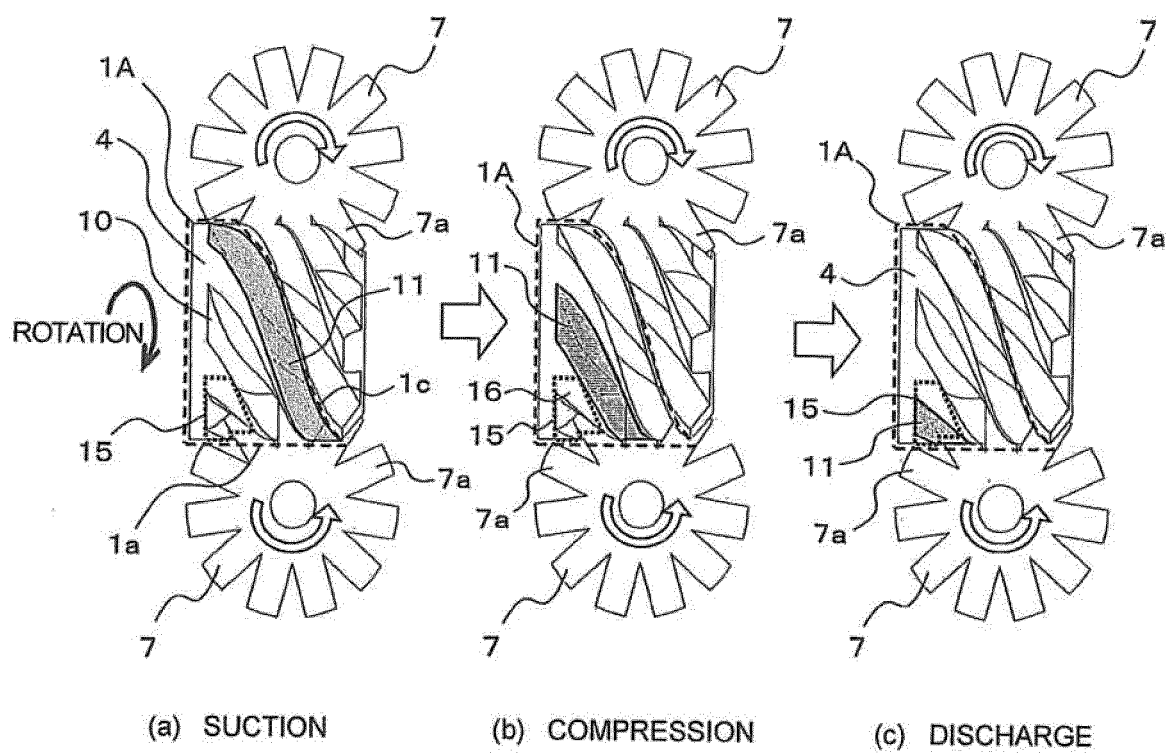
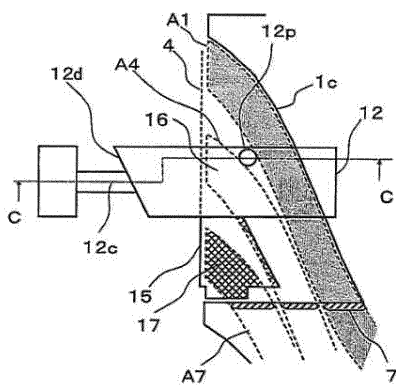
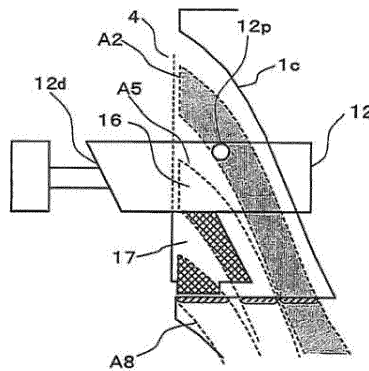


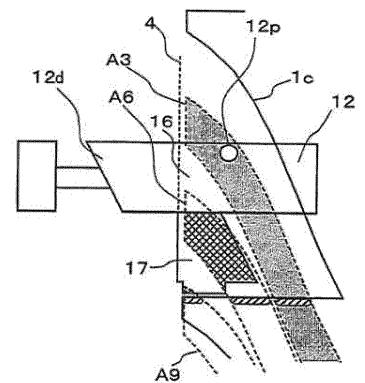
FIG. 11



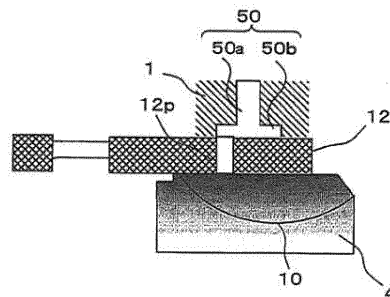
(a) SCREW GROOVES OF $\theta A(1)$, $\theta A(4)$, AND $\theta A(7)$



(b) SCREW GROOVES OF $\theta A(2)$, $\theta A(5)$, AND $\theta A(8)$

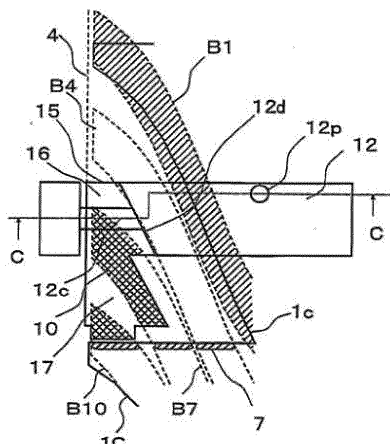


(c) SCREW GROOVES OF $\theta A(3)$, $\theta A(6)$, AND $\theta A(9)$

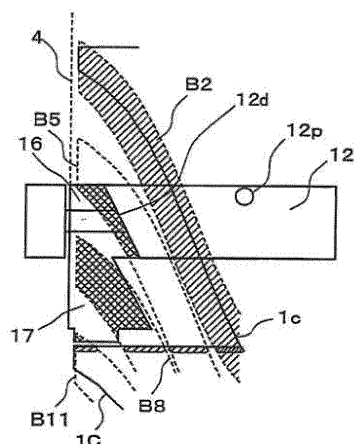


(d) C-C CROSS-SECTION

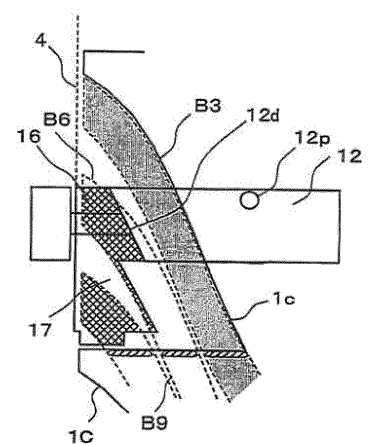
FIG. 12



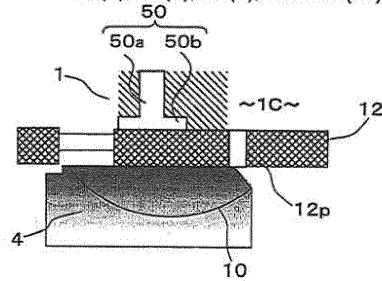
(a) SCREW GROOVES OF
0B(1), 0B(4), 0B(7), AND 0B(10)



(b) SCREW GROOVES OF
0B(2), 0B(5), 0B(8), AND 0B(11)



(c) SCREW GROOVES OF
0B(3), 0B(6), AND 0B(9)



(d) C-C CROSS-SECTION

FIG. 13

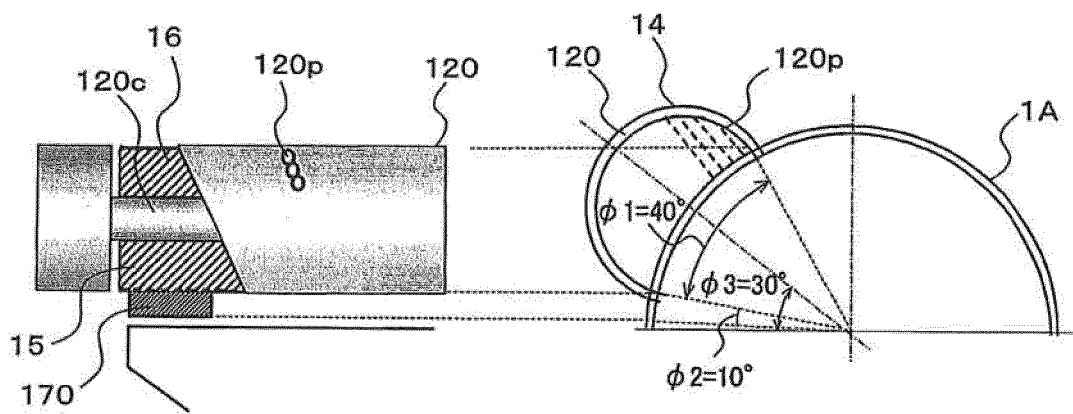


FIG. 14

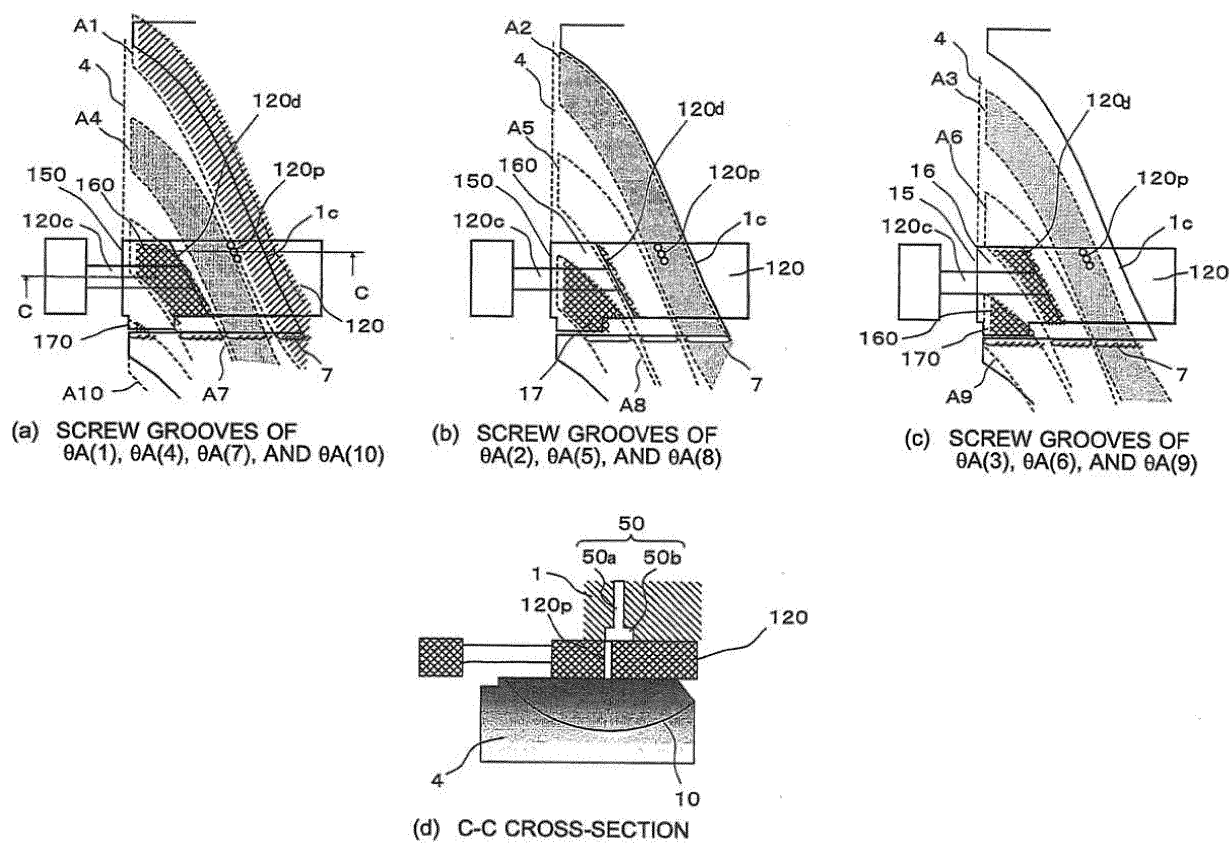


FIG. 15

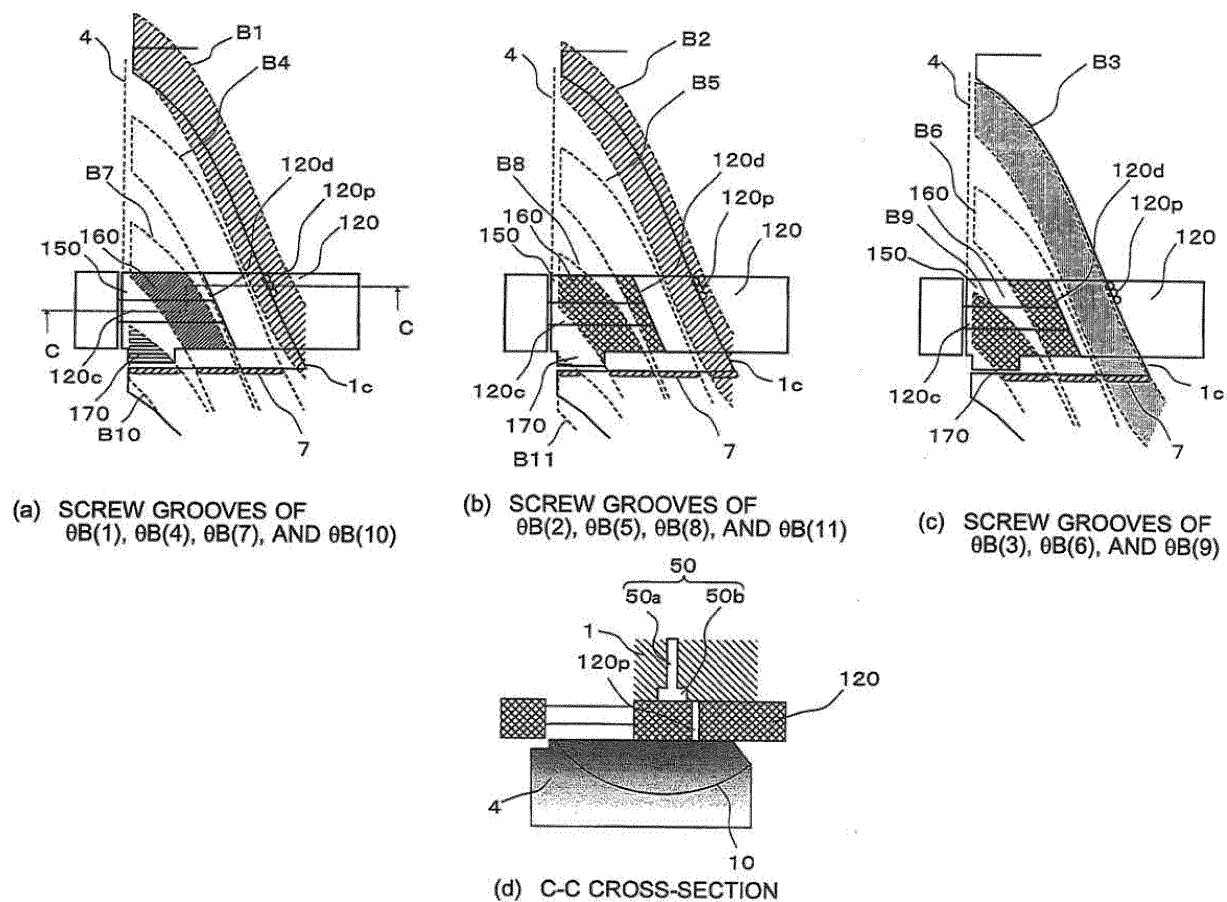


FIG. 15A

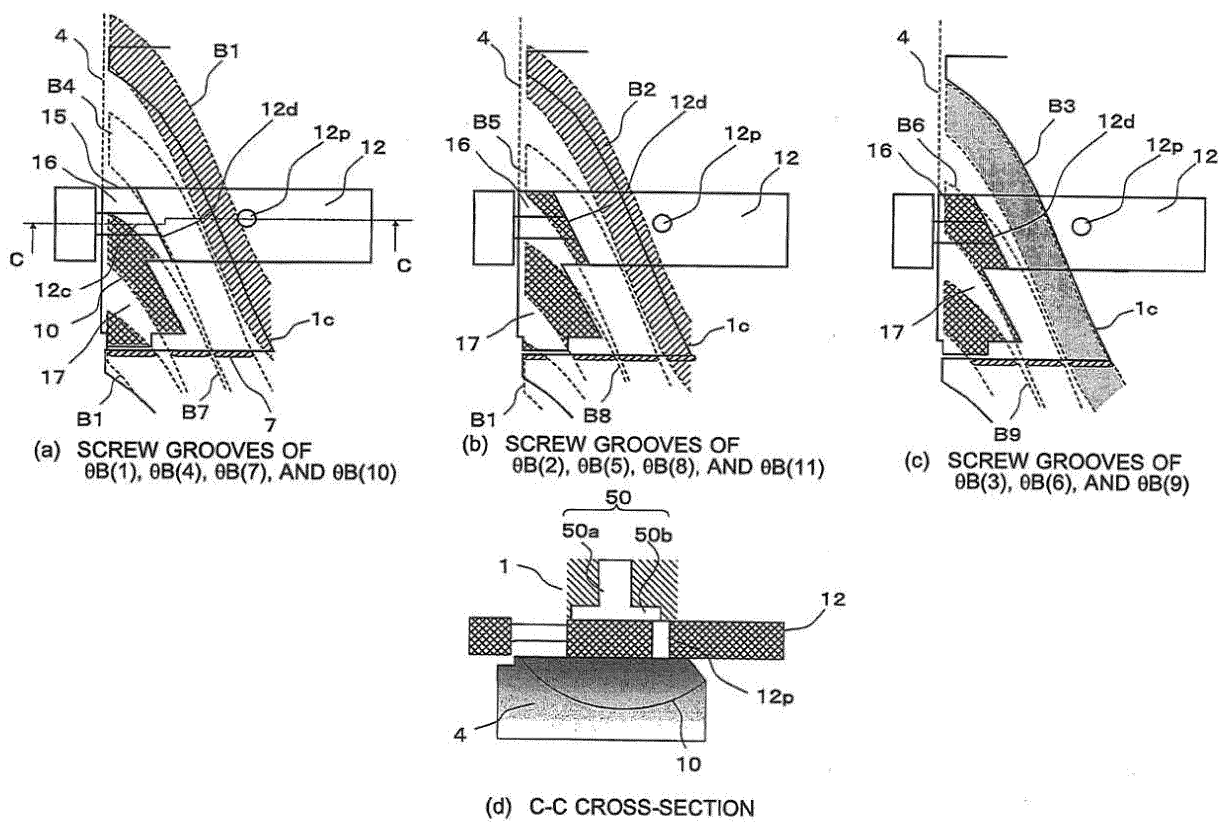
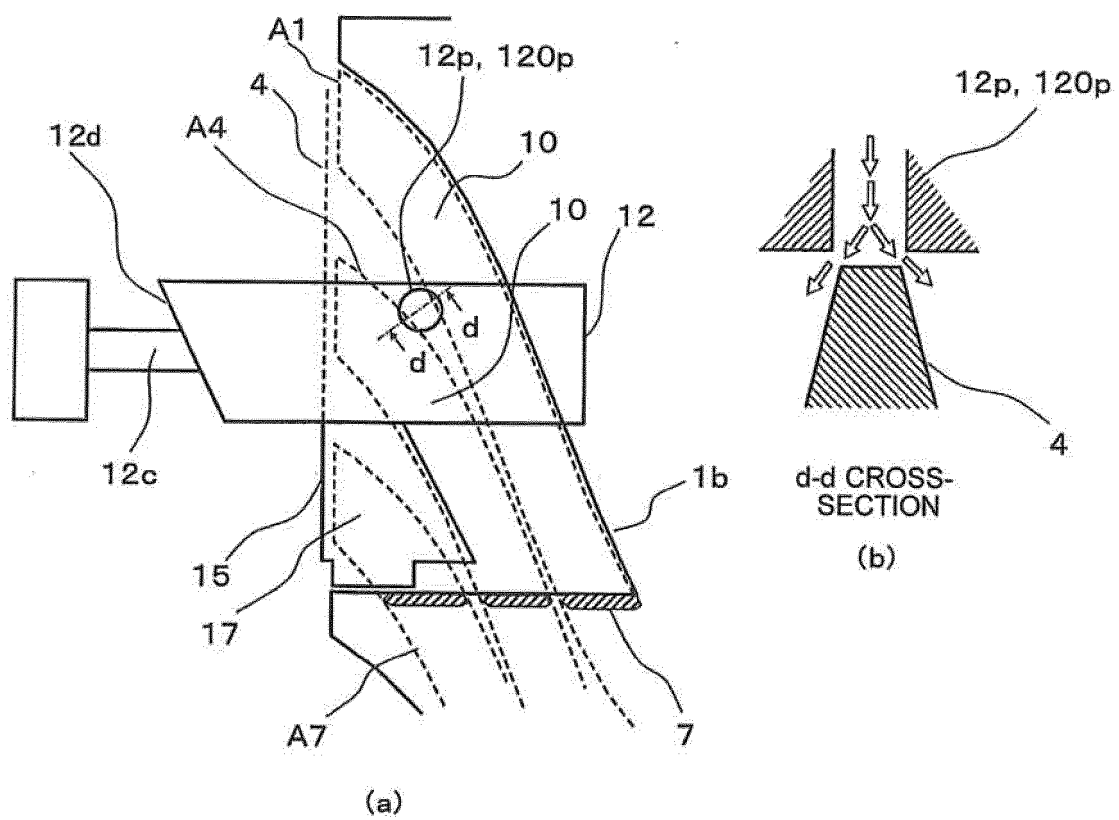


FIG. 16



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2014/064346

A. CLASSIFICATION OF SUBJECT MATTER

F04C18/52(2006.01)i, F04C18/16(2006.01)i, F04C28/26(2006.01)i, F25B1/047 (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)

F04C18/52, F04C18/16, F04C28/26, F25B1/047

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 1996-2014

Kokai Jitsuyo Shinan Koho 1971-2014 Toroku Jitsuyo Shinan Koho 1994-2014

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	JP 2009-19624 A (Daikin Industries, Ltd.), 29 January 2009 (29.01.2009), claim 1; fig. 1 & JP 4183021 B1 & US 2010/0229595 A1 & EP 2166229 A1 & WO 2008/153061 A1 & CN 101680450 A	1-6
A	JP 5-106572 A (Daikin Industries, Ltd.), 27 April 1993 (27.04.1993), fig. 2, 6 (Family: none)	1-6
A	JP 4-136663 A (Daikin Industries, Ltd.), 11 May 1992 (11.05.1992), fig. 1 to 4 (Family: none)	1-6

☐ 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 August, 2014 (04.08.14)

Date of mailing of the international search report

12 August, 2014 (12.08.14)

Name and mailing address of the ISA/
Japanese Patent Office

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REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 5010614 A [0010]
- JP 4136663 A [0010]