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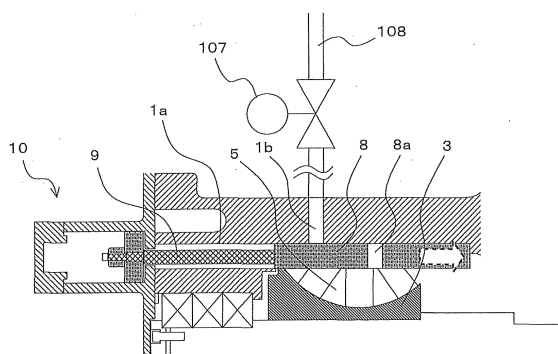
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(54) **SCREW COMPRESSOR AND REFRIGERATION CYCLE DEVICE**

(57) A screw compressor includes a casing (1), a screw rotor (3) rotatable in the casing (1), a compression chamber (5) defined between the casing (1) and the screw rotor (3) to compress refrigerant gas, a slide groove (1a) disposed on the inner peripheral surface of the casing (1) and extending along the direction of the rotational axis of the screw rotor (3), an economizer gas passage (1b) extending through the casing (1) to communicate the outside of the casing (1) with the slide groove (1a), a slidable valve (8) disposed in the slide groove (1a) so as to be slidable along the direction of the rotational axis

of the screw rotor (3), and an economizer port (8a) penetrating through the slidable valve (8) to communicate the economizer gas passage (1b) with the compression chamber (5) depending on the position of the slidable valve (8), wherein the slidable valve (8) moves between a first position, which allows the economizer gas passage (1b) to communicate with the compression chamber (5), and a second position, which prevents the economizer port (8a) from communicating with the compression chamber (5).

FIG. 6



Description

Technical Field

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

Background Art

[0002] Some traditional refrigeration cycle apparatuses are equipped with an intermediate cooler in the refrigeration cycle in order to increase the refrigeration capacity and improve the performance or coefficient of performance (ratio of the refrigeration capacity to an input to a compressor) of the refrigeration cycle (see, for example, Patent Literature 1). After cooling main-stream liquid in the intermediate cooler with refrigerant gas, the refrigeration cycle apparatus performs economizing operation to introduce the refrigerant gas (hereinafter referred to as "economizer gas") into an intermediate part of the compressor. In the refrigeration cycle apparatus, the intermediate cooler is disposed between a condenser and an evaporator in the refrigeration cycle. The refrigeration cycle apparatus further includes an economizer pipe branching from an intermediate portion of the passage from the condenser to the evaporator, an expansion valve for intermediate cooling disposed on the economizer pipe, and a screw compressor having an economizer port connected to the economizer pipe.

[0003] Some traditional screw compressors include a screw rotor and a casing accommodating the screw rotor. The casing has an economizer port for injecting refrigerant into a compression chamber defined between the screw rotor and the inner surface of the casing (see, for example, Patent Literature 2).

Citation List

Patent Literature

[0004]

Patent Literature 1: Japanese Unexamined Patent Application Publication No. H11-248264 (p. 4, Fig. 1)
Patent Literature 2: Japanese Patent No. 4140488 (p.5, Fig. 1)

Summary of Invention

Technical Problem

[0005] For these refrigeration cycle apparatuses equipped with the screw compressors, the energy saving performance was generally represented by the coefficient of performance (ratio of the refrigeration capacity to an electric power consumption) under the rated condition (full load mode: 100% load). Recently, interest has focused on indexes based on approximately the actual

operational conditions, for example, an integrated part load value (IPLV) standardized in the United States.

[0006] A typical refrigeration cycle apparatus runs under the rated condition in a very short period of the year. In specific, more than 90% of the annual operation is operated in a partial load mode. Most of the partial load mode is operated under 75% to 50% of the full load. The full load mode differs from the partial load mode in the flow rate of refrigerant circulation, operational compression ratio, and coefficient of performance. These circumstances of the actual operation draw attention on the IPLV. In specific, the IPLV is an index based on the coefficient of performance in the partial load mode.

[0007] The full load mode has a large pressure difference in the refrigeration cycle, which indicates high capacity operation, whereas the partial load mode has a small pressure difference in the refrigeration cycle, which indicates low capacity operation. In the full load mode having a large pressure difference, the economizing operation is effective to increase the coefficient of performance. In the partial load mode having a smaller pressure difference, however, the economizing operation becomes less effective. Under some conditions in the partial load mode, the economizing operation is less effective for an increase in the refrigeration capacity and adversely increases the electric power consumption, resulting in a decrease in the coefficient of performance. To increase the IPLV, the economizing operation can be switched between the drive and halt depending on operational conditions, such as the full load mode and the partial load mode.

[0008] Unfortunately, the economizer ports disclosed in Patent Literatures 1 and 2 always communicate with the compression chambers. Each economizer port is thus a volume part (dead volume) that is subject to useless compression from low pressure to high pressure. The economizer port passing over the compression chamber during the halt of the economizing operation causes re-expansion loss that impairs the refrigeration capacity.

[0009] An object of the invention, which has been accomplished to overcome the above problems, is to provide a screw compressor and a refrigeration cycle apparatus that include an economizer port in a better position and can achieve high coefficient of performance and high refrigeration capacity in a wide range of operation. Solution to Problem

[0010] A screw compressor of the invention includes

a casing; a screw rotor rotatable in the casing; a compression chamber defined between the casing and the screw rotor and configured to compress refrigerant gas; a slide groove disposed on an inner peripheral surface of the casing and extending along a direction of a rotational axis of the screw rotor; an economizer gas passage extending through the casing to communicate between the outside of the casing and the slide groove; a first slidable valve dis-

posed in the slide groove to be slidable along the direction of the rotational axis of the screw rotor; and an economizer port penetrating through the first slidable valve to communicate between the economizer gas passage and the compression chamber depending on a position of the first slidable valve, the first slidable valve being configured to move between a first position and a second position, the first slidable valve at the first position allowing the economizer gas passage to communicate with the compression chamber, the first slidable valve at the second position preventing the economizer port from communicating with the compression chamber.

[0011] A refrigeration cycle apparatus of the invention includes a refrigerant circuit including the screw compressor, a condenser, a high-pressure unit of an intermediate cooler, an expansion device, and an evaporator connected in sequence with a refrigerant pipe; and an economizer pipe branching from a portion between the intermediate cooler and the expansion device and connected to the economizer gas passage of the screw compressor through an expansion valve for the intermediate cooler and a low-pressure unit of the intermediate cooler.

Advantageous Effects of Invention

[0012] The invention can provide a screw compressor and a refrigeration cycle apparatus that include an economizer port in an optimized position and can achieve high coefficient of performance and high refrigeration capacity in a wide range of operation.

Brief Description of Drawings

[0013]

[Fig. 1] Fig. 1 illustrates a refrigerant circuit of a refrigeration cycle apparatus including a screw compressor according to Embodiment 1 of the invention.
 [Fig. 2] Fig. 2 is a schematic longitudinal-sectional view of a screw compressor according to Embodiment 1 of the invention.
 [Fig. 3] Fig. 3 illustrates the principle of compression in a screw compressor according to Embodiment 1 of the invention.
 [Fig. 4] Fig. 4 is a schematic sectional view illustrating the position of an economizer port of a screw compressor according to Embodiment 1 of the invention under a large-pressure-difference condition, such as a full load mode.
 [Fig. 5] Fig. 5 is a development view of the inner peripheral surface of a casing and a screw rotor of a screw compressor according to Embodiment 1 of the invention under a large-pressure-difference condition, such as a full load mode.
 [Fig. 6] Fig. 6 is a schematic sectional view illustrating the position of an economizer port of a screw com-

pressor according to Embodiment 1 of the invention under a small-pressure-difference condition, such as a partial load mode.

[Fig. 7] Fig. 7 is a development view of the inner peripheral surface of a casing and a screw rotor of a screw compressor according to Embodiment 1 of the invention under a small-pressure-difference condition, such as a partial load mode.

[Fig. 8] Fig. 8 is a schematic sectional view illustrating the position of an economizer port of a screw compressor according to Embodiment 1 of the invention under a large-pressure-difference condition, such as a full load mode.

[Fig. 9] Fig. 9 is a schematic sectional view illustrating the position of an economizer port of a screw compressor according to Embodiment 1 of the invention under a small-pressure-difference condition, such as a partial load mode.

[Fig. 10] Fig. 10 is a development view of the inner peripheral surface of a casing and a screw rotor of a screw compressor according to Embodiment 2 of the invention.

[Fig. 11] Fig. 11 is a development view of the inner peripheral surface of a casing and a screw rotor of a screw compressor according to Embodiment 3 of the invention.

Description of Embodiments

[0014] Embodiments of the invention will now be described with reference to the accompanying drawings. The description of the embodiments of the invention focuses on an example single-screw compressor equipped with a single screw rotor engaged with two gate rotors.

Embodiment 1

[0015] Fig. 1 illustrates a refrigerant circuit of a refrigeration cycle apparatus including a screw compressor according to Embodiment 1 of the invention. In the accompanying drawings including Fig. 1, the components referred to by the same reference sign are same as or equivalent to each other throughout the following description. The embodiments of the components disclosed in the entire specification are given for mere illustration and should not be construed to limit the invention. In specific, the combinations of the components in the embodiments should not be construed to limit the invention, and the components in one embodiment may be appropriately applied to another embodiment. High and low pressures are not absolutely determined relative to a fixed reference value, but relatively determined based on states and operations of the system and the apparatus, etc.

[0016] A refrigeration cycle apparatus 100 is equipped with a refrigerant circuit including a screw compressor 102 driven by an inverter 101, a condenser 103, a high-pressure unit of an intermediate cooler 104, an expansion valve 105 (expansion device), and an evaporator 106,

which are connected in sequence with a refrigerant pipe. The refrigeration cycle apparatus 100 further includes an economizer pipe 108, which branches from a portion between the intermediate cooler 104 and the expansion valve 105 and is connected to the screw compressor 102 through an intermediate-cooler expansion valve 107 (expansion valve for the intermediate cooler) and a low-pressure unit of the intermediate cooler 104.

[0017] The condenser 103 cools and condenses gas discharged from the screw compressor 102. The expansion valve 105 performs throttle expansion to main-stream refrigerant flowing out from the intermediate cooler 104. The evaporator 106 evaporates the main-stream refrigerant from the expansion valve 105. The intermediate cooler 104 has the high-pressure unit and the low-pressure unit, as described above. High-pressure refrigerant (main-stream refrigerant between the condenser 103 and the expansion valve 105) passes through the high-pressure unit, whereas intermediate-pressure refrigerant (the pressure of part of the high-pressure refrigerant is reduced by the intermediate-cooler expansion valve 107 to an intermediate pressure within the whole pressure range in the refrigeration cycle) passes through the low-pressure unit. The intermediate cooler 104 then causes heat exchange between the high-pressure refrigerant and the intermediate-pressure refrigerant to cool the high-pressure refrigerant.

[0018] The refrigeration cycle apparatus 100 further includes a controller 109. The controller 109 controls the inverter 101, the expansion valve 105, and the intermediate-cooler expansion valve 107, controls the position of at least one slidable valve (described below) of the screw compressor 102, and controls the drive and halt of economizing operation for injecting economizer gas into a compression chamber.

(Screw compressor)

[0019] The screw compressor 102 according to Embodiment 1 of the invention will now be described with reference to Fig. 2.

[0020] Fig. 2 is a schematic longitudinal-sectional view of the screw compressor according to Embodiment 1 of the invention.

[0021] As illustrated in Fig. 2, the screw compressor 102 includes a tubular casing 1 accommodating a motor 2. The motor 2 is equipped with a stator 2a fixed to the inner surface of the casing 1 and a motor rotor 2b disposed inside the stator 2a.

[0022] The casing 1 also accommodates a screw rotor 3. The screw rotor 3 and the motor rotor 2b are disposed on the same axis and fixed to a screw shaft 4. The screw rotor 3 has helical screw grooves 5a on the outer peripheral surface, and is coupled to the motor rotor 2b fixed to the screw shaft 4 to be rotated. The screw grooves 5a engage with teeth 6a of gate rotors 6. A space surrounded by the teeth 6a of the gate rotors 6, the screw grooves 5a, and the inner peripheral surface of the casing 1 de-

fines a compression chamber 5. The casing 1 is divided by a partition (not shown) into a low-pressure compartment (adjacent to the suction end) and a high-pressure compartment (adjacent to the discharge end). The high-pressure compartment has an outlet 7 (Fig. 3 described below) in communication with a discharge chamber (not shown).

[0023] With reference to Fig. 2, the inner peripheral surface of the casing 1 has a slide groove 1a extending along the direction of the rotational axis of the screw rotor 3. The slide groove 1a slidably accommodates a slidable valve 8 (first slidable valve). The slidable valve 8 constitutes part of the inner peripheral surface with the casing 1 to define the compression chamber 5. The slidable valve 8 has an economizer port 8a. The economizer port 8a penetrates the slidable valve 8 from its outer surface to slide on the slide groove 1a to its inner surface to slide on the screw rotor 3. Fig. 2 illustrates an example of the casing 1 that accommodates a single slidable valve 8 having the economizer port 8a.

[0024] The casing 1 has an economizer gas passage 1b for introducing refrigerant gas from the intermediate cooler 104 into the compression chamber 5 (the screw grooves 5a during a compression stroke). The economizer gas passage 1b communicates with the compression chamber 5 through the economizer port 8a. The economizer gas passage 1b is also connected to the economizer pipe 108. In this configuration, the refrigerant gas from the intermediate cooler 104 separates from the main stream to cool the main-stream liquid, and then flows into the compression chamber 5 through the economizer pipe 108, the economizer gas passage 1b, and the economizer port 8a. The economizer gas passage 1b of the casing 1 may have a space (not shown) for reducing the pulsation of flowing gas and communicate with the compression chamber 5 through the space, for example.

[0025] The slidable valve 8 is coupled to a drive unit 10 including a piston or the like, with a coupling rod 9, and is driven by the drive unit 10 to slide in the slide groove 1a along the direction of the rotational axis of the screw rotor 3. The drive unit 10 for driving the slidable valve 8 is a unit such as those powered by gas pressure or oil pressure, or powered by a motor other than a piston, that is, the driving method is not limited.

(Operation of refrigerant circuit)

[0026] The operation according to Embodiment 1 will now be explained.

[0027] With reference to Fig. 1, the operation of the refrigerant circuit will now be explained under a large-pressure-difference condition, such as a full load (100% load) mode, in the refrigeration cycle.

[0028] The screw compressor 102 sucks refrigerant gas flowing from the evaporator 106, and compresses and discharges the refrigerant gas. The discharged refrigerant gas is cooled in the condenser 103. The refrigerant

erant cooled in the condenser 103 flows into the intermediate cooler 104. The intermediate cooler 104 causes heat exchange between high-pressure refrigerant, which flows from the condenser 103 into the high-pressure unit, and intermediate-pressure refrigerant, which branches off after passing through the intermediate cooler 104, undergoes decompression in the intermediate-cooler expansion valve 107, and then enters the low-pressure unit. In other words, the high-pressure refrigerant that flows directly from the condenser 103 into the high-pressure unit of the intermediate cooler 104 is subcooled by the heat exchange with the intermediate-pressure refrigerant. The addition of subcooling degree enhances the refrigerating effects of the evaporator 106.

[0029] The intermediate-pressure refrigerant entering the low-pressure unit of the intermediate cooler 104, cools the high-pressure refrigerant in the high-pressure unit, flows through the economizer pipe 108 and the economizer gas passage 1b, and is injected from the economizer port 8a of the slidable valve 8 into the compression chamber 5. In specific, the difference of the high-pressure side pressure and the intermediate pressure of the economizer gas as the high-pressure side from the pressure in the compression chamber 5 causes the economizer gas to be injected from the economizer port 8a into the compression chamber 5. The injected economizer gas is mixed with compressed gas.

[0030] The operation of the refrigerant circuit will now be explained under a small-pressure-difference condition, such as a partial load (lower than 100% load) mode, in the refrigeration cycle.

[0031] Under the small-pressure-difference condition, the pressure difference is small between the exit of the intermediate cooler and the compression chamber 5. Such a small pressure difference prevents the economizer gas from readily entering the compression chamber 5. The small pressure difference thus destabilizes the economizing operation. In addition, the small pressure difference impairs the effects of the increased refrigeration capacity and adversely increases the electric power consumption due to the injection of the economizer gas during compression, resulting in a decrease in the coefficient of performance. To address this problem, the intermediate-cooler expansion valve 107 is closed to halt the economizing operation under a small-pressure-difference condition.

(Operation of screw compressor)

[0032] The operation of the screw compressor 102 according to Embodiment 1 will now be explained.

[0033] Fig. 3 illustrates the principle of compression in the screw compressor according to Embodiment 1.

[0034] With reference to Fig. 3, the rotation of the screw rotor 3 driven by the motor 2 (see Fig. 2) with the screw shaft 4 (see Fig. 2) relatively moves the teeth 6a of the gate rotors 6 in the compression chamber 5 (screw grooves 5a). With this operation, an intake stroke, com-

pression stroke, and discharge stroke are carried out at the compression chamber 5 as one cycle, and the cycle is repeated. The following will explain the individual strokes focusing on the compression chamber 5, which is illustrated with dotted hatching in Fig. 3.

[0035] Fig. 3(a) illustrates a state of the compression chamber 5 during the intake stroke. The screw rotor 3 is driven by the motor 2 to rotate along the direction of the solid arrow. This rotation reduces the volume of the compression chamber 5, as illustrated in Fig. 3(b).

[0036] Further rotation of the screw rotor 3 causes the compression chamber 5 to communicate with the outlet 7, as illustrated in Fig. 3(c). The high-pressure refrigerant gas compressed in the compression chamber 5 is thus discharged to the outside through the outlet 7. The same compression is performed again behind the screw rotor 3.

[0037] Although the economizer port 8a, the slidable valve 8 having the economizer port 8a, and the slide groove 1a are not illustrated in Fig. 3, the economizer gas in the economizing operation enters the compression chamber 5 from the economizer port 8a during the compression stroke. The economizer gas in the compression chamber 5 is compressed together with sucked gas, and is discharged to the outside during the discharge stroke.

(Large-pressure-difference condition: drive of economizing operation)

[0038] The following explanation focuses on the positional relationship of the economizer port 8a with the economizer gas passage 1b and the compression chamber 5 (screw grooves 5a) under a large-pressure-difference condition, such as a full load mode.

[0039] Fig. 4 is a schematic sectional view illustrating the position of the economizer port of the screw compressor according to Embodiment 1 of the invention under a large-pressure-difference condition, such as the full load mode. Fig. 5 is a development view of the inner peripheral surface of the casing and the screw rotor of the screw compressor according to Embodiment 1 of the invention under a large-pressure-difference condition, such as the full load mode.

[0040] During the drive of the economizing operation, the controller 109 controls the slidable valve 8 having the economizer port 8a to move toward the discharge end (the left of Fig. 4 or 5), as illustrated with the outline arrow of Fig. 4 or 5, such that the economizer port 8a is disposed in a position (first position) so as to communicate with the economizer gas passage 1b and the compression chamber 5. The economizer gas passage 1b of the casing 1 thus communicates with the compression chamber 5 through the economizer port 8a.

[0041] While the compression chamber 5 is being in communication with the economizer port 8a in the compression stroke, the economizer gas is injected through the economizer gas passage 1b and the economizer port 8a into the compression chamber 5. In this configuration, as the pressure (intermediate pressure) in the economiz-

er port 8a in communication with the compression chamber 5 increases, the effects of increasing the refrigeration capacity by the economizing operation decrease. In addition, the economizer gas, which is injected into the compression chamber 5 before completion of closing of the compression chamber 5, flows from the compression chamber 5 toward the suction end and inhibits sucked gas from entering the screw grooves 5a. To address this problem, the slidable valve 8 is moved for shifting the economizer port 8a to the position illustrated in Fig. 5, so that the economizer gas is injected into the compression chamber 5 at a low pressure as much as possible without inhibiting the sucked gas from entering the compression chamber 5. The details will be explained below.

[0042] The compression chamber 5, which is surrounded by a thick broken line in Fig. 5, is in the position at which suction of gas (trapping of sucked gas) is completed. The economizer port 8a is disposed in the position illustrated in Fig. 5, i.e., the position from which the economizer port 8a is open to communication with the compression chamber 5 upon the completion of trapping of the sucked gas (upon the start of compression). The economizer gas can thus be injected into the compression chamber 5 at a low pressure as much as possible without inhibiting the sucked gas from entering the compression chamber 5.

[0043] Even in the partial load mode, the controller 109 also performs the economizing operation if the pressure difference is relatively large to ensure the economization. In specific, the controller 109 controls the slidable valve 8 to move to the position illustrated in Fig. 5 such that the economizer gas passage 1b of the casing 1, the economizer port 8a, and the compression chamber 5 communicate with each other. The economizer gas is thus injected into the compression chamber 5.

(Small-pressure-difference condition: halt of economizing operation)

[0044] The following explanation focuses on the positional relationship of the economizer port 8a with the economizer gas passage 1b and the compression chamber 5 (screw grooves 5a) under a small-pressure-difference condition, such as a partial load mode.

[0045] Fig. 6 is a schematic sectional view illustrating the position of the economizer port of the screw compressor according to Embodiment 1 of the invention under a small-pressure-difference condition, such as the partial load mode. Fig. 7 is a development view of the inner peripheral surface of the casing and the screw rotor of the screw compressor according to Embodiment 1 of the invention under a small-pressure-difference condition, such as the partial load mode.

[0046] During the halt of the economizing operation, the controller 109 controls the slidable valve 8 having the economizer port 8a to move toward the suction end (the right of Fig. 6 or 7), as illustrated with the outline arrow of Fig. 6 or 7. In specific, the economizer port 8a is shifted

to a position (hereinafter referred to as "second position") along the axis direction so as not to communicate with the compression chamber 5 (screw grooves 5a). The economizer port 8a thus does not communicate with the economizer gas passage 1b of the casing 1 or the compression chamber 5. In other words, the economizer port 8a is completely separated from the compression chamber 5 during the halt of the economizing operation.

[0047] The economizer port 8a is shifted to the position along the axis direction so as not to communicate with the compression chamber 5 (screw grooves 5a), in other words, so as to be separated from the compression chamber 5 (screw grooves 5a) during the halt of the economizing operation. Accordingly, the economizer port 8a and the economizer gas passage 1b do not affect the compression chamber 5 from the intake stroke to the discharge stroke during the halt of the economizing operation. This configuration prevents the economizer port 8a and the economizer gas passage 1b from being a volume part (dead volume) that is subject to useless compression. That is, the screw compressor 102 according to Embodiment 1 has no dead volume.

[0048] Even in the full load mode, the controller 109 halts the economizing operation under the condition of a relatively small pressure difference not causing effective economization. In specific, the controller 109 controls the slidable valve 8 to move to the second position such that the economizer gas passage 1b of the casing 1, the economizer port 8a, and the compression chamber 5 do not communicate with each other.

[0049] As described above, the slidable valve 8 having the economizer port 8a is accommodated in the casing 1 so as to be slidable along the direction of the rotational axis of the screw rotor 3 according to Embodiment 1. The slidable valve 8 can move between the first position, which allows the economizer gas passage 1b, the economizer port 8a, and the compression chamber 5 to communicate with each other, and the second position, which prevents the economizer gas passage 1b, the economizer port 8a, and the compression chamber 5 from communicating with each other.

[0050] With this configuration, the economizing operation is effectively performed under a large-pressure-difference condition (where the pressure difference is larger than a predetermined pressure difference in the refrigeration cycle), resulting in an increase in the coefficient of performance. The economizing operation is halted under a small-pressure-difference condition (where the pressure difference is equal to or smaller than the predetermined pressure difference in the refrigeration cycle). The configuration has no dead volume during the halt, as described above, and thus causes no re-expansion loss, resulting in an increase in the coefficient of performance. That is, Embodiment 1 can provide the screw compressor 102 and the refrigeration cycle apparatus 100 that can achieve a high coefficient of performance in a wide range of operation.

[0051] The slidable valve 8 may have any range of

movement. The slidable valve 8, the coupling rod 9, and the drive unit 10 may be disposed, such that the first position of the slidable valve 8 is "most adjacent to the discharge end" in the range of movement of the slidable valve 8 whereas the second position of the slidable valve 8 is "most adjacent to the suction end" in the range of movement of the slidable valve 8.

[0052] Alternatively, the economizer port 8a is shifted toward the suction end (the right of Fig. 8) during performing the economizing operation, as illustrated with the outline arrow of Fig. 8. In specific, the economizer port 8a is disposed in the position along the axis direction so as to communicate with the economizer gas passage 1b and the compression chamber 5. The economizer gas passage 1b of the casing 1 thus communicates with the compression chamber 5 through the economizer port 8a. In this case, the economizer port 8a is shifted toward the discharge end (the left of Fig. 9) during the halt of the economizing operation, as illustrated with the outline arrow of Fig. 9. In specific, the economizer port 8a is disposed in the position along the axis direction so as not to communicate with the compression chamber 5 (screw grooves 5a). As described above, the position to communicate the economizer passage with the compression chamber and the position not to communicate may be interchangeable.

Embodiment 2

[0053] Embodiment 2 differs from Embodiment 1 only in the shape of the suction end surface of the slidable valve 8 having the economizer port 8a.

[0054] Fig. 10 is a development view of the inner peripheral surface of the casing and the screw rotor of the screw compressor according to Embodiment 2 of the invention. The following explanation of Embodiment 2 focuses on the difference from Embodiment 1. The components not described in Embodiment 2 are identical to those in Embodiment 1.

[0055] According to Embodiment 2, the slidable valve 8 having the economizer port 8a has a suction end surface 8b, which extends along the slope of each of the screw grooves 5a. This shape of the suction end surface 8b, in comparison to a suction end surface 8b of the slidable valve 8 perpendicular to the screw shaft 4 in Embodiment 1, can bring about the following advantageous effects: the configuration does not require an extra space for movement of the slidable valve 8 and can thus achieve a reduction in size of the components in addition to the effects comparable to those in Embodiment 1. Although the suction end surface 8b of the slidable valve 8 extends along the slope of the screw groove 5a in this embodiment, the suction end surface 8b may be any inclined surface. It should be noted that the suction end surface 8b of the slidable valve 8 extending along the slope of the screw groove 5a can ensure to have necessary surface for closing the screw groove 5a, leading to further optimization of the shape (a reduction in size). In addition,

a reduction in the surfaces of the slidable valve 8 useless for the closing can decrease the viscous resistance of the oil between the slidable valve 8 and the outer surface of the screw rotor.

Embodiment 3

[0056] In comparison to the screw compressor 102 according to Embodiment 1 or 2 including the slidable valve 8 for varying the position of the economizer port 8a, a volume-controllable screw compressor 102 according to Embodiment 3 further includes a slidable valve for varying the internal volume ratio.

[0057] Fig. 11 is a development view of the inner peripheral surface of the casing and the screw rotor of the screw compressor according to Embodiment 3 of the invention. The following explanation of Embodiment 3 focuses on the differences from Embodiment 1. The components not described in Embodiment 3 are identical to those in Embodiment 1.

[0058] In the screw compressor 102 according to Embodiment 3, the casing 1 further accommodates a slidable valve 11 (second slidable valve) for varying the internal volume ratio, in addition to the slidable valve 8, such that the slidable valve 11 is slidable along the direction of the rotational axis of the screw rotor 3. The slidable valve 11 adjusts the timing to start discharge of high-pressure gas compressed in the compression chamber 5 (the timing of completion of compression) depending on the slide position of the slidable valve 11. The slidable valve 11 has a discharge end surface 11a constituting part of the outlet 7. A change in the discharging area of the outlet 7 depending on the slide position varies the discharge timing and the internal volume ratio. In specific, advanced discharge timing provides an operation with a small internal volume ratio, whereas delayed discharge timing provides an operation with a large internal volume ratio.

[0059] The internal volume ratio indicates the ratio of the volume of the compression chamber 5 just before the discharge to the volume of the compression chamber 5 upon the completion of an intake operation (start of compression), i.e., the ratio of the volume upon the opening of the outlet 7 to the volume upon the completion of the intake operation. In general, a screw compressor does not cause loss due to improper compression under an operational condition of a proper compression ratio, i.e., in the case of the actual compression ratio matching the internal volume ratio. In an operation of a small compression ratio, however, the gas is over-compressed before the opening of an outlet to have a pressure higher than the discharge pressure, resulting in excess compression. In an operation of a high compression ratio, the outlet opens before achieving the discharge pressure, resulting in insufficient compression that causes reverse flow of gas. To address this problem, the position of the slidable valve 11 is adjusted for optimizing the discharge timing.

[0060] The slidable valve 8 having the economizer port

8a moves between two positions, i.e., the position allowing for communication and the position preventing the communication between the economizer gas passage 1b, the economizer port 8a, and the compression chamber 5. In contrast, the slidable valve 11 for varying the internal volume ratio can freely move in accordance with any proper discharge timing.

[0061] As described above, the configuration according to Embodiment 3 further includes the slidable valve 11 for varying the internal volume ratio that can move to a position for optimizing the discharge timing. The configuration can thus prevent over-compression and insufficient compression and increase the coefficient of performance in addition to bringing about the effects comparable to those in Embodiment 1. That is, Embodiment 3 can provide the screw compressor 102 and the refrigeration cycle apparatus 100 that can achieve a higher coefficient of performance in a wide range of operation.

[0062] The screw compressor according to an embodiment of the invention may be replaced with a twin-screw compressor including male and female screw rotors that engage with each other to define a compression chamber 5, other than a single-screw compressor.

Reference Signs List

[0063]

1 casing 1a slide groove 1b economizer gas passage
2 motor 2a stator 2b motor rotor 3 screw rotor 4 screw shaft 5 compression chamber 5a screw groove 6 gate rotor 6a tooth 7 outlet 8 slidable valve 8a economizer port 8b intake end surface 9 coupling rod 10 drive unit 11 slidable valve 11a discharge end surface 100 refrigeration cycle apparatus 101 inverter 102 screw compressor 103 condenser 104 intermediate cooler 105 expansion valve 106 evaporator 107 intermediate-cooler expansion valve 108 economizer pipe 109 controller

Claims

1. A screw compressor comprising:

a casing;
a screw rotor rotatable in the casing;
a compression chamber defined between the casing and the screw rotor and configured to compress refrigerant gas;
a slide groove disposed on an inner peripheral surface of the casing and extending along a direction of a rotational axis of the screw rotor;
an economizer gas passage extending through the casing to communicate between an outside of the casing and the slide groove;
a first slidable valve disposed in the slide groove to be slidable along the direction of the rotational

axis of the screw rotor; and
an economizer port penetrating through the first slidable valve to communicate between the economizer gas passage and the compression chamber depending on a position of the first slidable valve,
the first slidable valve being configured to move between a first position and a second position, the first slidable valve at the first position allowing the economizer gas passage to communicate with the compression chamber, the first slidable valve at the second position preventing the economizer port from communicating with the compression chamber.

2. The screw compressor of claim 1, further comprising a second slidable valve disposed in the casing, the second slidable valve being slidable along the direction of the rotational axis of the screw rotor to adjust timing of discharge from the compression chamber.

3. The screw compressor of claim 1 or 2, wherein, the economizer port is configured to open from the first position to communicate with the compression chamber upon completion of trapping of the refrigerant gas in the compression chamber.

4. The screw compressor of any one of claims 1 to 3, wherein
the screw compressor is included in a refrigeration cycle, and
the first slidable valve is situated in the first position under a large-pressure-difference condition where a high-low pressure difference in the refrigeration cycle is larger than a predetermined pressure difference in the refrigeration cycle, and the first slidable valve is situated in the second position under a small-pressure-difference condition where the high-low pressure difference is equal to or smaller than the predetermined pressure difference in the refrigeration cycle.

5. The screw compressor of any one of claims 1 to 4, wherein a suction end surface of the first slidable valve is an inclined surface.

6. The screw compressor of claim 5, wherein the inclined surface extends along a slope of a screw groove defining the compression chamber.

7. The screw compressor of any one of claims 1 to 6, further comprising an electric motor driven by an inverter and configured to rotate the screw rotor.

8. A refrigeration cycle apparatus comprising:

a refrigerant circuit comprising the screw compressor of any one of claims 1 to 7, a condenser,

a high-pressure unit of an intermediate cooler,
an expansion device, and an evaporator con-
nected in sequence with a refrigerant pipe; and
an economizer pipe branching from a portion
between the intermediate cooler and the expan- 5
sion device, the economizer pipe being connect-
ed to the economizer gas passage of the screw
compressor through an expansion valve for the
intermediate cooler and a low-pressure unit of
the intermediate cooler. 10

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FIG. 1

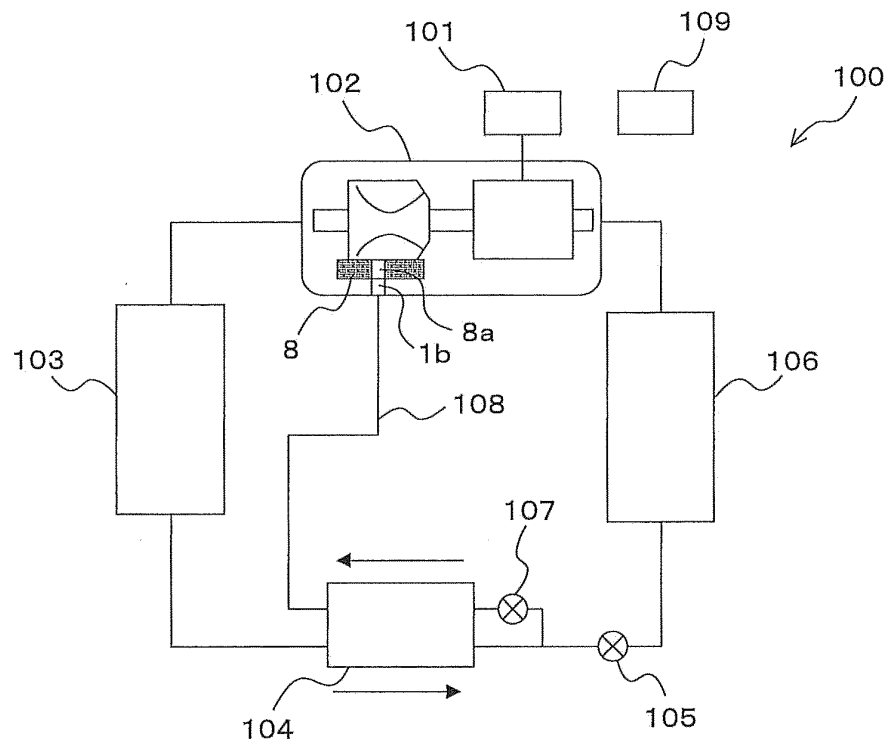


FIG. 2

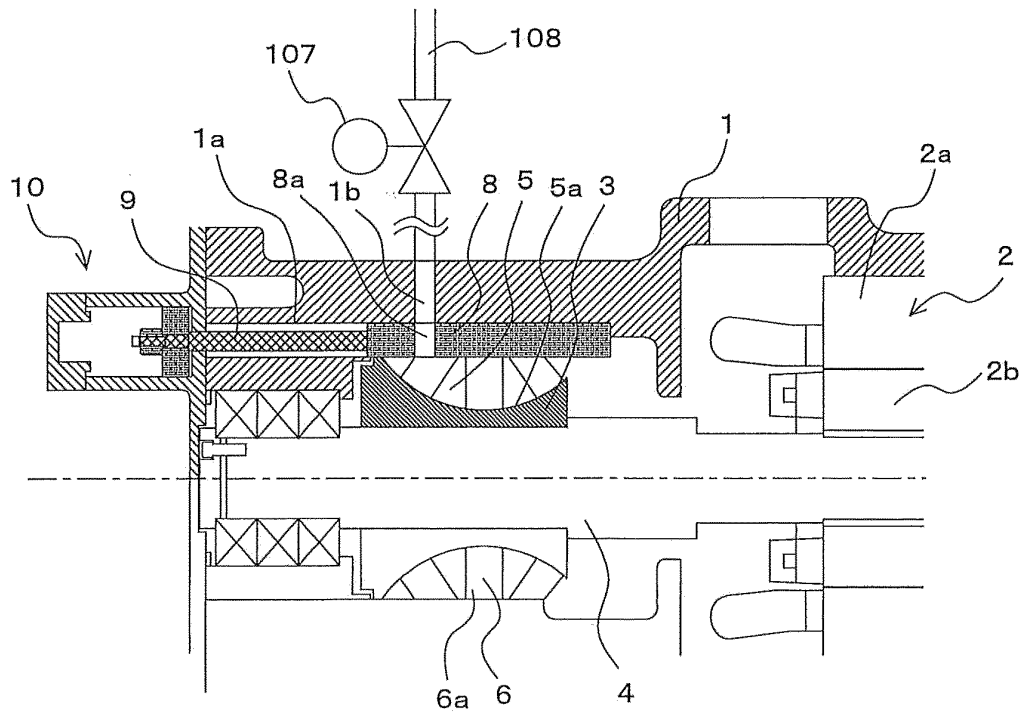


FIG. 3

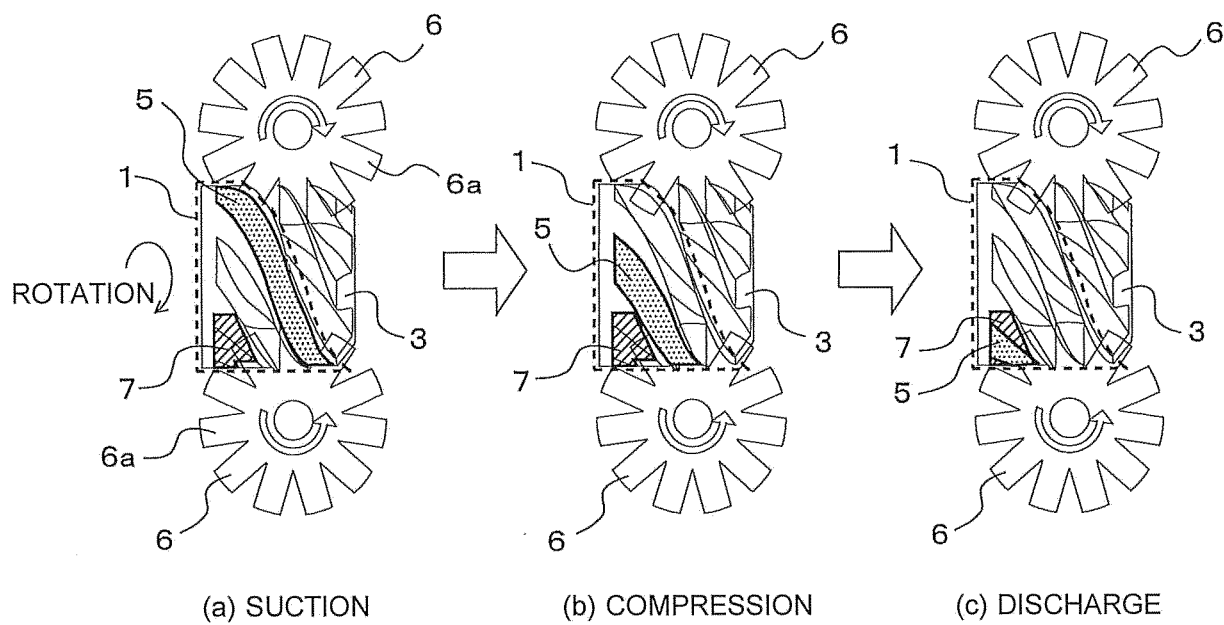


FIG. 4

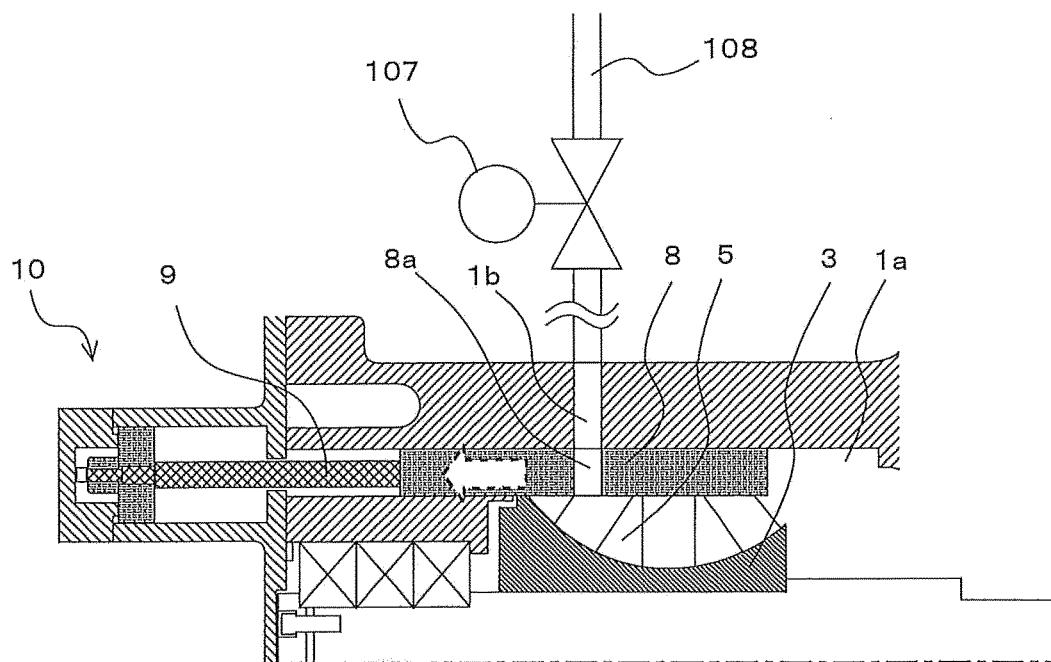


FIG. 5

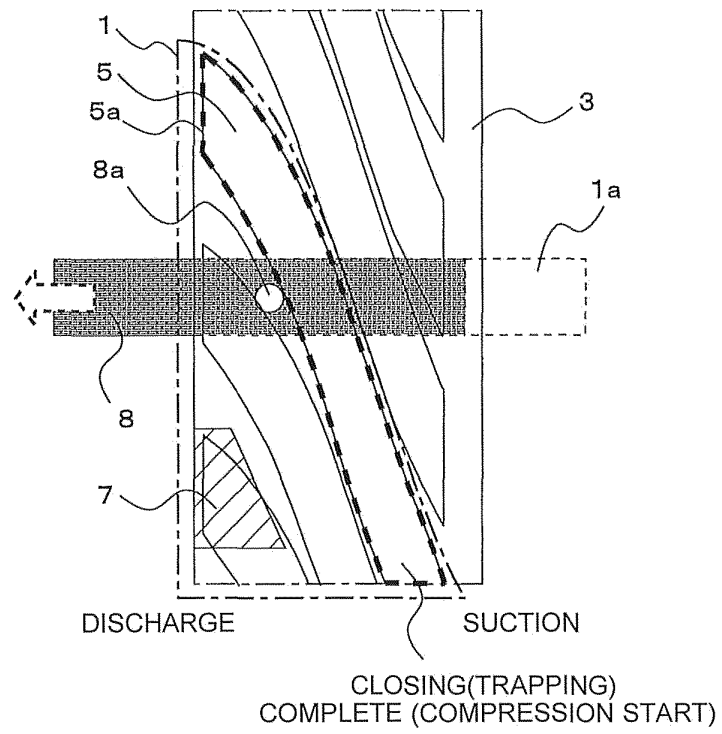


FIG. 6

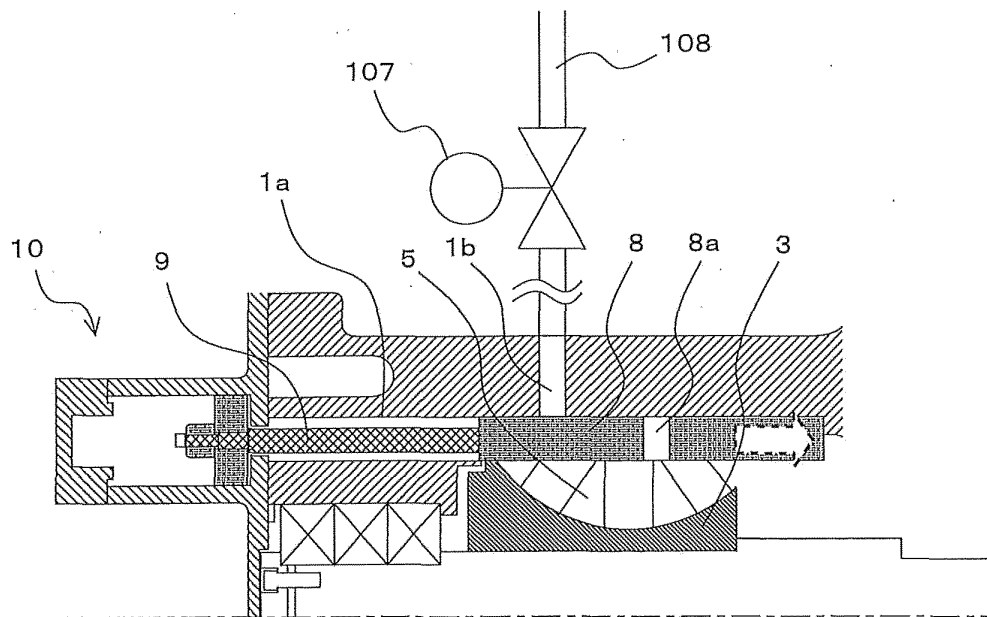


FIG. 7

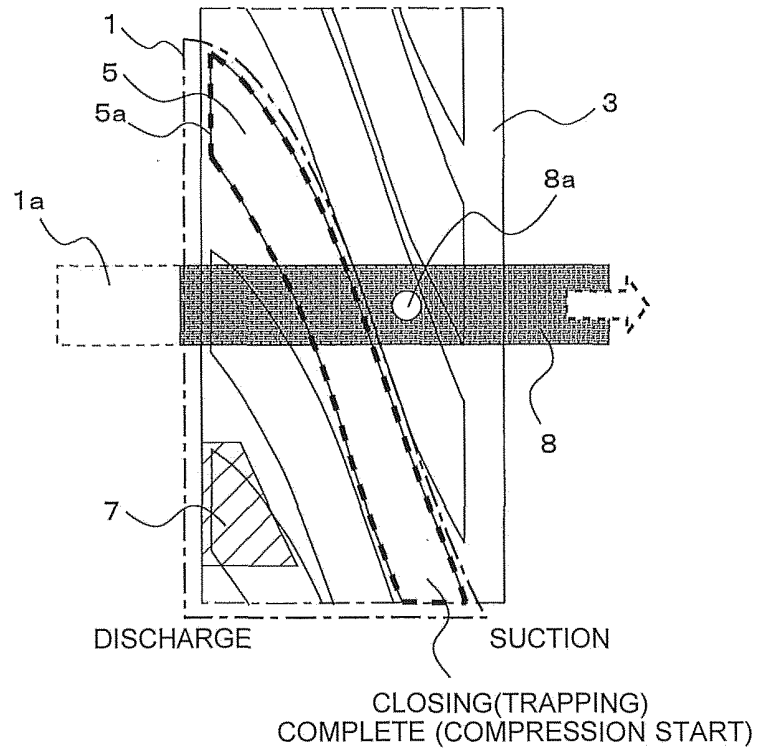


FIG. 8

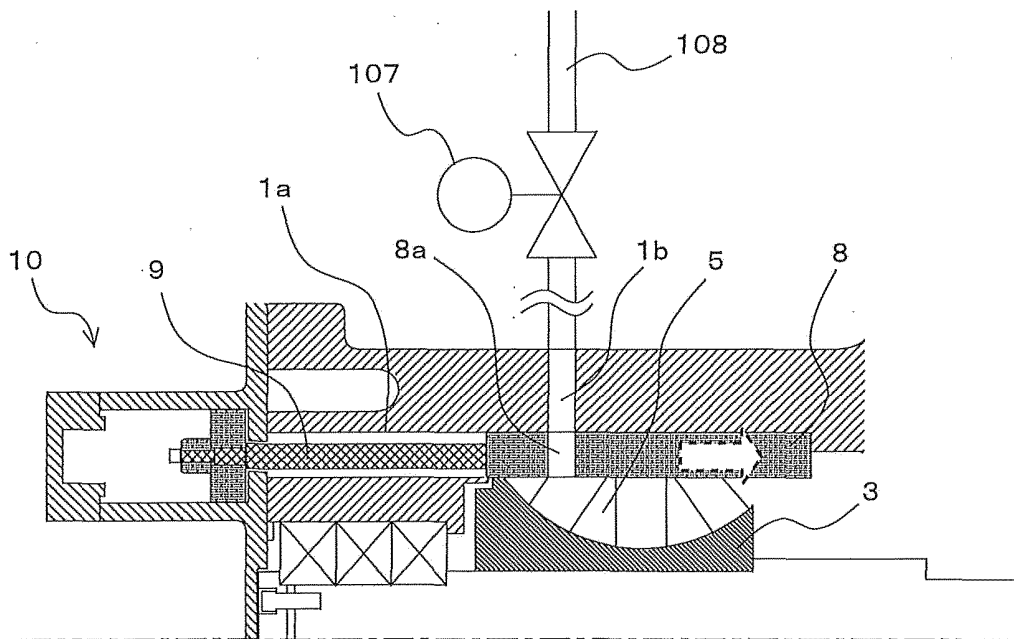


FIG. 9

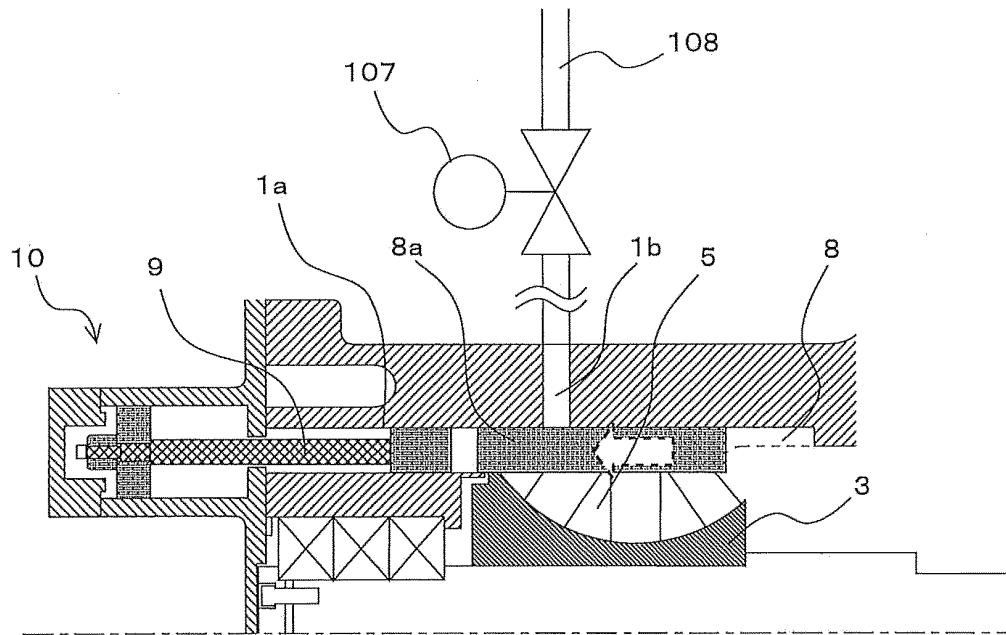


FIG. 10

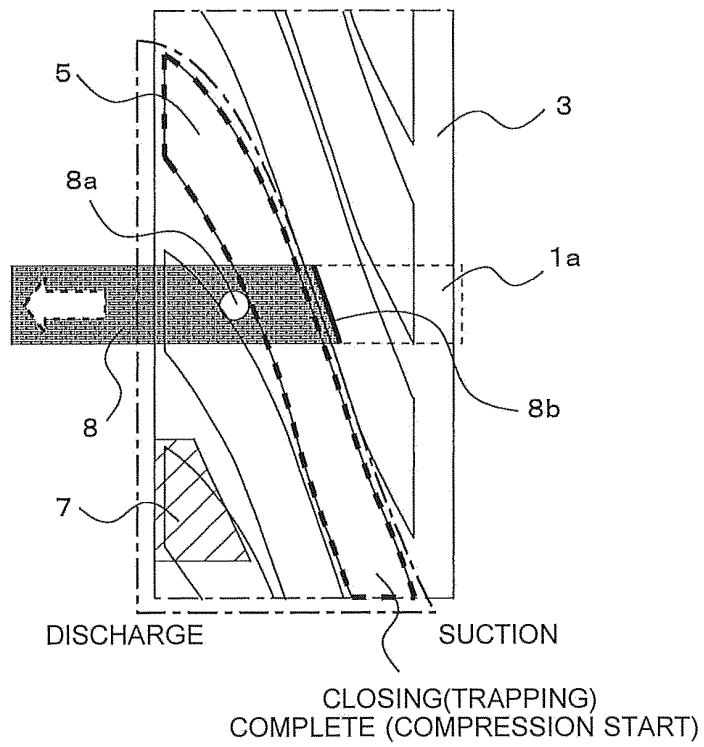
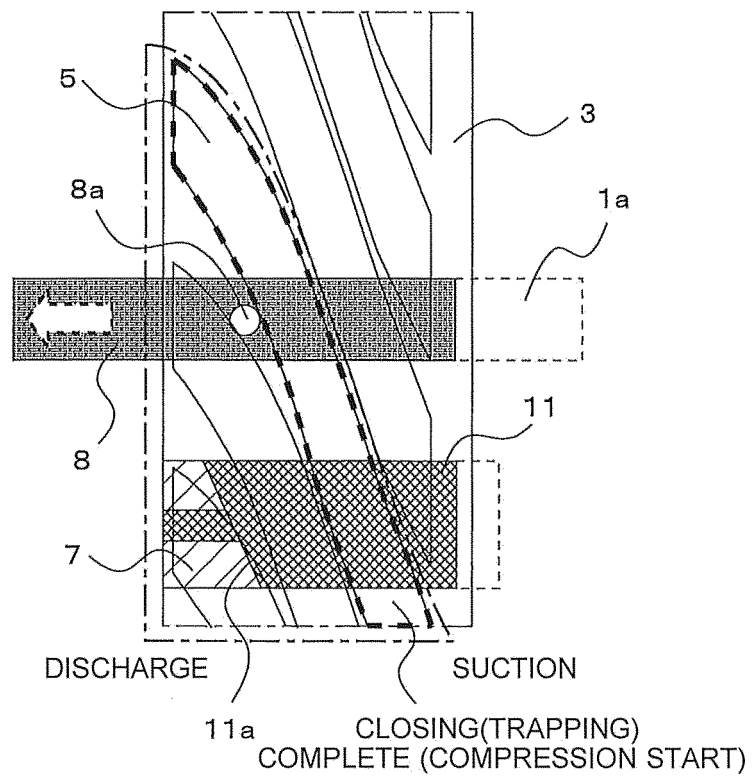


FIG. 11



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2014/075230

A. CLASSIFICATION OF SUBJECT MATTER

F04C29/04(2006.01)i, F04C18/16(2006.01)i, F04C18/52(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)

F04C29/04, F04C18/16, F04C18/52, 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), paragraph [0040]; fig. 1 & JP 4183021 B1 & US 2010/0229595 A1 & EP 2166229 A1 & WO 2008/153061 A1 & CN 101680450 A	1-8
A	Microfilm of the specification and drawings annexed to the request of Japanese Utility Model Application No. 127661/1983 (Laid-open No. 35169/1985) (Kobe Steel, Ltd.), 11 March 1985 (11.03.1985), fig. 3 (Family: none)	1-8

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

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

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"&" document member of the same patent family

Date of the actual completion of the international search

09 December 2014 (09.12.14)

Date of mailing of the international search report

22 December 2014 (22.12.14)

Name and mailing address of the ISA/
Japan Patent Office

Authorized officer

Facsimile No.

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INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2014/075230

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	JP 4-136663 A (Daikin Industries, Ltd.), 11 May 1992 (11.05.1992), fig. 1 (Family: none)	1-8
A	JP 5-106572 A (Daikin Industries, Ltd.), 27 April 1993 (27.04.1993), fig. 2 (Family: none)	1-8

Form PCT/ISA/210 (continuation of second sheet) (July 2009)

REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP H11248264 B [0004]
- JP 4140488 B [0004]