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(72) Inventors:
• **SUN, Hongzhi**
Osaka 540-6207 (JP)
• **SHOYAMA, Tadayoshi**
Osaka 540-6207 (JP)
• **KAWANO, Bunki**
Osaka 540-6207 (JP)
• **TAMURA, Tomoichiro**
Osaka 540-6207 (JP)
• **HAYASHI, Yoshimi**
Osaka 540-6207 (JP)
• **MATSUI, Masaru**
Osaka 540-6207 (JP)

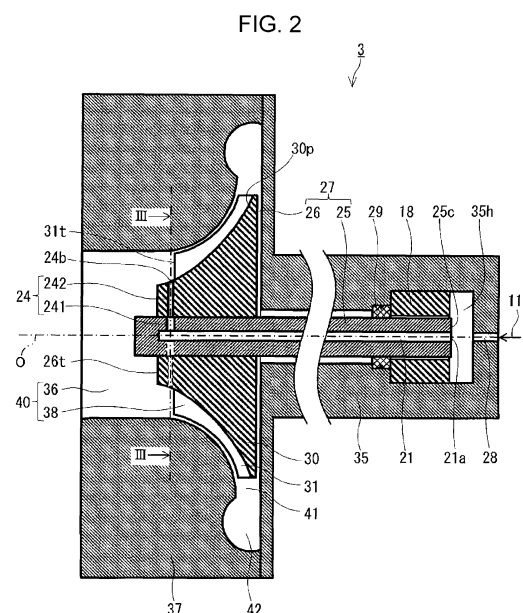
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(71) Applicant: **Panasonic Corporation**
Osaka 571-8501 (JP)

(74) Representative: **Eisenführ Speiser**
Patentanwälte Rechtsanwälte PartGmbB
Postfach 31 02 60
80102 München (DE)

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

(57) A dynamic compressor includes a rotating body including a rotating shaft and at least one impeller, a refrigerant flow path that is located around the rotating body and that enables a gas-phase refrigerant to flow there-through, a main flow path that extends in the axial direction of the rotating body inside the rotating body and that enables a liquid-phase refrigerant to flow therethrough, and an injection flow path that is located inside the rotating body and that branches off from the main flow path and extends from the main flow path to the refrigerant flow path so as to lead a liquid-phase refrigerant from the main flow path to the refrigerant flow path.



Description

Technical Field

5 **[0001]** The present disclosure relates to a dynamic compressor and a refrigeration cycle apparatus.

Background Art

10 **[0002]** Some existing refrigeration cycle apparatuses include a two-stage compressor and are configured to cool refrigerant vapor discharged from the first-stage compressor before the refrigerant vapor is sucked into a second-stage compressor.

15 **[0003]** As illustrated in Fig. 21, an air conditioner 500 described in PTL 1 includes an evaporator 510, a centrifugal compressor 531, a steam cooler 533, a Roots type compressor 532, and a condenser 520. The centrifugal compressor 531 is provided in the front stage, and the Roots type compressor 532 is provided in the rear stage. The evaporator 510 generates saturated refrigerant vapor. The refrigerant vapor is sucked into the centrifugal compressor 531 and is compressed. The refrigerant vapor compressed by the centrifugal compressor 531 is further compressed by the Roots type compressor 532. The refrigerant vapor is cooled by the steam cooler 533 disposed between the centrifugal compressor 531 and the Roots type compressor 532.

20 **[0004]** The steam cooler 533 is provided between the centrifugal compressor 531 and the Roots type compressor 532. In the steam cooler 533, water is directly sprayed on the refrigerant vapor. Alternatively, in the steam cooler 533, heat exchange is indirectly performed between a cooling medium, such as air, and the refrigerant vapor.

Citation List

25 Patent Literature

[0005] PTL 1: Japanese Unexamined Patent Application Publication No. 2008-122012

Summary of Invention

30 **[0006]** According to the technique described in PTL 1, in the steam cooler 533, the degree of superheat of the refrigerant to be sucked into the Roots type compressor 532 can be reduced. However, the degree of superheat generated in the compression process performed by the centrifugal compressor 531 and the degree of superheat generated in the compression process performed by the Roots type compressor 532 cannot be removed in the compression process.

35 **[0007]** The present disclosure provides a dynamic compressor including a rotating body having a rotating shaft and at least one impeller, a refrigerant flow path that is located around the rotating body and that enables a gas-phase refrigerant to flow therethrough, a main flow path that extends in the axial direction of the rotating body inside the rotating body and that enables a liquid-phase refrigerant to flow therethrough, and an injection flow path that is located inside the rotating body and that branches off from the main flow path and extends from the main flow path to the refrigerant flow path so as to lead a liquid-phase refrigerant from the main flow path to the refrigerant flow path.

40 **[0008]** According to the present disclosure, the degree of superheat generated in the compression process can be removed in the compression process. In this manner, the efficiency of the refrigeration cycle apparatus can be improved.

Brief Description of Drawings

45 **[0009]**

[Fig. 1] Fig. 1 is a configuration diagram of a refrigeration cycle apparatus according to a first embodiment of the present disclosure.

50 [Fig. 2] Fig. 2 is a cross-sectional view of the dynamic compressor according to the first embodiment of the present disclosure.

[Fig. 3] Fig. 3 is a cross-sectional view of a rotating body taken along line III - III.

[Fig. 4A] Fig. 4A is a cross-sectional view of a rotating body according to a modification.

[Fig. 4B] Fig. 4B is a partial side view of a rotating shaft according to the modification.

55 [Fig. 5] Fig. 5 is a cross-sectional view of a compressor according to the modification.

[Fig. 6] Fig. 6 is a cross-sectional view of a compressor according to another modification.

[Fig. 7] Fig. 7 is a plan projection view of an impeller, illustrating the vicinity of an injection flow path in an enlarged manner.

[Fig. 8] Fig. 8 is a plan projection view of the impeller, illustrating the vicinity of an injection flow path in an enlarged manner (a stationary coordinate system).

[Fig. 9] Fig. 9 is a graph denoting an outflow angle necessary for preventing collision of refrigerant droplets.

[Fig. 10] Fig. 10 is a cross-sectional view of a multi-stage dynamic compressor according to still another modification.

[Fig. 11] Fig. 11 is a cross-sectional view of a first impeller and a second impeller.

[Fig. 12] Fig. 12 is a cross-sectional view of the impeller at a position including an injection flow path.

[Fig. 13] Fig. 13 is a cross-sectional view of a multi-stage dynamic compressor according to still another modification.

[Fig. 14] Fig. 14 is a cross-sectional view of a dynamic compressor according to yet another modification.

[Fig. 15] Fig. 15 is a cross-sectional view of a dynamic compressor according to a further modification.

[Fig. 16] Fig. 16 is a cross-sectional view of a dynamic compressor according to a still further modification.

[Fig. 17] Fig. 17 is a configuration diagram of a refrigeration cycle apparatus according to a second embodiment of the present disclosure.

[Fig. 18] Fig. 18 is a configuration diagram of a refrigeration cycle apparatus according to a third embodiment of the present disclosure.

[Fig. 19] Fig. 19 is a configuration diagram of a refrigeration cycle apparatus according to a fourth embodiment of the present disclosure.

[Fig. 20] Fig. 20 is a flowchart illustrating a compression method according to the present disclosure.

[Fig. 21] Fig. 21 is a configuration diagram of an existing air conditioner. Description of Embodiments

(Underlying Knowledge Forming Basis of Aspect of Present Disclosure)

[0010] According to the air conditioner described in PTL 1, in the steam cooler 533, the degree of superheat of the refrigerant sucked into the Roots type compressor 532 can be reduced. However, the degree of superheat generated in the compression process performed by the centrifugal compressor 531 and the degree of superheat generated in the compression process performed by the Roots type compressor 532 cannot be removed in the compression process. If the degree of superheat of the refrigerant increases, the enthalpy of the refrigerant also increases.

[0011] The ideal compression process performed by a compressor progresses along a perfectly insulated isentropic curve. In the p-h diagram of the refrigerant, the slope of the isentropic curve becomes gentler with increasing enthalpy of the refrigerant and, thus, larger compression power is required. As the degree of superheat of the refrigerant increases, more compression power is required to raise the pressure of the unit mass refrigerant to a predetermined pressure. That is, the load imposed on the compressor increases, and the power consumption of the compressor increases.

[0012] The present disclosure provides a technique for removing, in a compression process, the degree of superheat generated in a compression process. In addition, the present disclosure provides a technique for improving the efficiency of the refrigeration cycle apparatus.

(Overview of One Aspect According to Present Disclosure)

[0013] A dynamic compressor according to a first aspect of the present disclosure includes

a rotating body having a rotating shaft and at least one impeller,

a refrigerant flow path that is located around the rotating body and that enables a gas-phase refrigerant to flow there-through,

a main flow path that extends in the axial direction of the rotating body inside the rotating body and that enables a liquid-phase refrigerant to flow therethrough, and

an injection flow path that is located inside the rotating body and that branches off from the main flow path and extends from the main flow path to the refrigerant flow path so as to lead a liquid-phase refrigerant from the main flow path to the refrigerant flow path.

[0014] According to the first aspect, the liquid-phase refrigerant is pressurized by centrifugal force and is injected toward the refrigerant flow path inside the compressor through the main flow path and the injection flow path. When the liquid-phase refrigerant is brought into contact with the gas-phase refrigerant in the refrigerant flow path, heat exchange occurs between the liquid-phase refrigerant and the gas-phase refrigerant, and the superheated gas-phase refrigerant is continuously cooled by the sensible heat or evaporation latent heat of the liquid-phase refrigerant. For this reason, an increase in the enthalpy of the refrigerant caused by an increase in the degree of superheat of the refrigerant in the compression process is continuously suppressed. The compression power required by the compressor can be reduced to less than the compression power required for perfectly adiabatic, isentropic compression. The work to be performed by the compressor to increase the pressure of the refrigerant to a predetermined pressure can be significantly reduced. That is, the power consumption of the compressor can be significantly reduced.

[0015] According to a second aspect of the present disclosure, for example, in the dynamic compressor according to the first aspect, the impeller may have a hub and a blade fixed to the hub, and the injection flow path may have an

outflow port facing the refrigerant flow path. The outflow port may be located upstream of an upstream end of the blade in a flow direction of the gas-phase refrigerant. According to such a configuration, heat can be efficiently removed from the gas-phase refrigerant in the compression process.

[0016] According to a third aspect of the present disclosure, for example, in the dynamic compressor according to the first or second aspect, the impeller may have a hub and a blade fixed to the hub, and the injection flow path may have an outflow port located on a surface of the hub. In addition, the injection flow path may penetrate the hub in a radial direction of the rotating shaft. According to such a configuration, the gas-phase refrigerant and the liquid-phase refrigerant can be mixed with each other before the gas-phase refrigerant enters the inter-blade flow path between the blades. In this manner, heat can be efficiently removed from the gas-phase refrigerant in the compression process.

[0017] According to a fourth aspect of the present disclosure, for example, in the dynamic compressor according to any one of the first to third aspects, the injection flow path may include a first portion extending from the main flow path in the radial direction of the rotating shaft inside the rotating shaft and a second portion located between the first portion and the refrigerant flow path. According to such a configuration, a sufficient length of the injection flow path can be ensured. The centrifugal acceleration applied to the liquid-phase refrigerant increases with increasing length of the injection flow path, and the liquid-phase refrigerant is more easily injected into the refrigerant flow path.

[0018] According to a fifth aspect of the present disclosure, for example, in the dynamic compressor according to the fourth aspect, the number of the injection flow paths each including the first portion and the second portion may be greater than or equal to two. According to such a configuration, the gas-phase refrigerant can be cooled uniformly in the circumferential direction of the rotating shaft.

[0019] According to a sixth aspect of the present disclosure, for example, in the dynamic compressor according to the fourth or fifth aspect, the first portion may include a groove provided on a side surface of the rotating shaft and extending in a circumferential direction of the rotating shaft, and the second portion may be connected to the groove. According to such a configuration, since the alignment of the first portion with the second portion in the circumferential direction of the rotating shaft is extremely easily accomplished or is not needed, the work of attaching the impeller to the rotating shaft is facilitated.

[0020] According to a seventh aspect of the present disclosure, for example, in the dynamic compressor according to any one of the first to sixth aspects, the main flow path may have an inflow port located on an end face of the rotating shaft. According to such a configuration, the liquid-phase refrigerant can be smoothly fed into the main flow path.

[0021] According to an eighth aspect of the present disclosure, for example, the dynamic compressor according to any one of the first to seventh aspects may further include a supply tank storing the liquid-phase refrigerant, a buffer chamber in contact with the inflow port of the main flow path, and a pressure pump that pumps the liquid-phase refrigerant from the supply tank to a buffer chamber via a refrigerant supply path connected to the buffer chamber. According to such a configuration, the liquid-phase refrigerant is pressurized by the pressure pump, and the pressure of the liquid-phase refrigerant rises. Thus, the boiling point of the liquid-phase refrigerant rises. For this reason, the liquid-phase refrigerant is less likely to evaporate inside the main flow path, and clogging of the flow path by vapor can be prevented.

[0022] According to a ninth aspect of the present disclosure, for example, the dynamic compressor according to the eighth aspect may further include a heat exchanger that exchanges heat with an external heat source. The refrigerant supply path may be a flow path connected to the buffer chamber and the pressure pump, and the heat exchanger may be provided in the refrigerant supply path between the buffer chamber and the pressure pump. According to such a configuration, since the liquid-phase refrigerant is cooled by a heat exchanger 23, the supercooled liquid-phase refrigerant is supplied to the main flow path 21. For this reason, the liquid-phase refrigerant is less likely to evaporate inside the main flow path 21.

[0023] According to a tenth aspect of the present disclosure, for example, in the dynamic compressor according to any one of the first to ninth aspects, the impeller may include a hub and a plurality of blades fixed to the hub. The injection flow path may have an outflow port facing the refrigerant flow path. One of the blades located closest to the outflow port in a rotational direction opposite to the rotational direction of the rotating body is defined as a first blade. In a projection view obtained by projecting a blade root line of the first blade onto a plane perpendicular to the rotating shaft, an outermost peripheral portion of the blade root line is defined as a first trailing edge portion. A line extending from the central axis of the rotating body in the radial direction through the outflow port is defined as an r-axis. The rotational direction of the rotating body is defined as a positive direction. Then, an angle formed by a line extending between the first trailing edge portion and the central axis and the r-axis as measured from the r-axis in the rotational direction of the rotating body may be greater than or equal to -40 degrees, and the ratio of a distance between the central axis of the rotating body and the first trailing edge portion to a distance between the central axis of the rotating body and the outflow port may be greater than or equal to three. In a projection view obtained by projecting an outflow direction of the liquid-phase refrigerant injected from the outflow port onto a plane perpendicular to the rotating shaft, an angle formed by the outflow direction of the liquid-phase refrigerant and the r-axis as measured from the r-axis in the rotational direction of the rotating body may be greater than or equal to -25 degrees. According to such a configuration, the angular displacement of the refrigerant droplet by the Coriolis force is less than the angle formed by the line extending between the trailing edge

portion of the blade and the rotating shaft and the r-axis, and collision of a large-sized refrigerant droplet with the trailing edge portion of the blade can be prevented. As a result, erosion of the impeller can be prevented.

[0024] According to an eleventh aspect of the present disclosure, for example, in the dynamic compressor according to any one of the first to tenth aspects, the at least one impeller may include a first impeller and a second impeller. Each of the first impeller and the second impeller may be provided with the injection flow path. The relationship ($R_2/R_1 \leq S_1/S_2$) may be satisfied, where an opening area of the outflow port of the injection flow path provided in the first impeller is denoted by S_1 , an opening area of the outflow port of the injection flow path provided in the second impeller is denoted by S_2 , a distance between the central axis of the rotating body and the outflow port provided in the first impeller is denoted by R_1 , and a distance between the central axis of the rotating body and the outflow port provided in the second impeller is denoted by R_2 . According to such a configuration, the injection quantity from the injection flow path of the second impeller is less than or equal to the injection quantity from the injection flow path of the first impeller. As a result, it is possible to prevent the liquid-phase refrigerant that has a large particle diameter and that does not follow the gas-phase refrigerant from colliding with the wall surface of the impeller and remaining around the wall surface.

[0025] According to a twelfth aspect of the present disclosure, for example, in the dynamic compressor according to any one of the first to eleventh aspects, the at least one impeller may include the first impeller and the second impeller, and the dynamic compressor may further include a first diffuser facing the first impeller. The first impeller may be provided with a downstream injection flow path that is located inside the first impeller and that branches off from the main flow path and extends from the main flow path to the refrigerant flow path. The downstream injection flow path may be located downstream of the injection flow path in the flow direction of the gas-phase refrigerant, and the central axis of the downstream injection flow path may intersect with an inlet of the first diffuser. According to such a configuration, the amount of refrigerant droplets present in each of the refrigerant flow path around the first impeller and the refrigerant flow path around the second impeller decreases. As a result, the probability of collision of the refrigerant droplets with the first impeller and the second impeller is reduced and, thus, the erosion risk of the first impeller and the second impeller is reduced.

[0026] According to a thirteenth aspect of the present disclosure, for example, the dynamic compressor according to the twelfth aspect may further include a second diffuser facing the second impeller. The second impeller may be provided with a second injection flow path that is located inside the second impeller and that branches off from the main flow path and extends from the main flow path to the refrigerant flow path. The central axis of the second injection flow path may intersect with an inlet of the second diffuser. According to such a configuration, heat can be removed from the gas-phase refrigerant at the time of performing pressure recovery in the second diffuser.

[0027] A refrigeration cycle apparatus according to a fourteenth aspect of the present disclosure includes an evaporator, the dynamic compressor according to any one of the first to thirteenth aspects, and a condenser.

[0028] According to the fourteenth aspect, the efficiency of the refrigeration cycle apparatus is improved by significantly reducing the power consumption of the dynamic compressor.

[0029] According to a fifteenth aspect of the present disclosure, for example, in the refrigeration cycle apparatus according to the fourteenth aspect, the evaporator may have a liquid-phase refrigerant stored therein, and the condenser may have a liquid-phase refrigerant stored therein. The refrigeration cycle apparatus may further include a refrigerant supply path that leads one of the liquid-phase refrigerant stored in the evaporator and the liquid-phase refrigerant stored in the condenser to the dynamic compressor. According to such a configuration, the liquid-phase refrigerant can be reliably supplied to the main flow path of the dynamic compressor.

[0030] A compression method according to a sixteenth aspect of the present disclosure is a compression method for use of a dynamic compressor, where the dynamic compressor has a rotating body including a rotating shaft and an impeller and a refrigerant flow path that is located around the rotating body and that enables a gas-phase refrigerant to flow therethrough from a suction port for the gas-phase refrigerant to a discharge port for the gas-phase refrigerant. The method includes

causing the dynamic compressor to suck the gas-phase refrigerant, accelerating and compressing the sucked gas-phase refrigerant in the dynamic compressor, and injecting, through a flow path that communicates with an outflow port disposed in a surface of the rotating body and that is located inside the rotating body, a liquid-phase refrigerant from the outflow port toward the gas-phase refrigerant present in the refrigerant flow path.

[0031] According to the sixteenth aspect, the same effect as in the first aspect can be obtained.

[0032] According to a seventeenth aspect of the present disclosure, for example, in the compression method according to the sixteenth aspect, the flow path located inside the rotating body may include a main flow path that may extend in an axial direction of the rotating body inside the rotating body and that enables the liquid-phase refrigerant to flow therethrough and an injection flow path that is located inside the rotating body and that branches off from the main flow path and extends from the main flow path to the refrigerant flow path so as to lead the liquid-phase refrigerant from the

main flow path to the refrigerant flow path, and the liquid-phase refrigerant flowing in the main flow path may flow in a direction opposite to a direction in which the gas-phase refrigerant is sucked and flows.

[0033] According to an eighteenth aspect of the present disclosure, for example, in the compression method according to the sixteenth or seventeenth aspect, the liquid-phase refrigerant may be injected from the outflow port by centrifugal force generated by rotation of the rotating body, and the injected liquid-phase refrigerant may be sucked by an inter-blade flow path of the dynamic compressor. The liquid-phase refrigerant can be efficiently injected by the centrifugal force of the rotating body.

[0034] According to a nineteenth aspect of the present disclosure, for example, in the compression method according to any one of the sixteenth to eighteenth aspects, the impeller may have a hub and a blade fixed to the hub, and the outflow port may be located upstream of an upstream end of the blade in the flow direction of the gas-phase refrigerant. According to such a configuration, heat can be efficiently removed from the gas-phase refrigerant in the compression process.

[0035] Embodiments of the present disclosure are described below with reference to the accompanying drawings. The present disclosure is not limited to the following embodiments.

(First Embodiment)

[0036] Fig. 1 illustrates the configuration of a refrigeration cycle apparatus according to a first embodiment of the present disclosure. A refrigeration cycle apparatus 100 includes an evaporator 2, a compressor 3, a condenser 4, and a refrigerant supply path 11. The compressor 3 is connected to the evaporator 2 by a suction pipe 6 and is connected to the condenser 4 by a discharge pipe 8. More specifically, the suction pipe 6 is connected to an outlet of the evaporator 2 and a suction port of the compressor 3. The discharge pipe 8 is connected to a discharge port of the compressor 3 and an inlet of the condenser 4. The condenser 4 is connected to the evaporator 2 by a return path 9. The evaporator 2, the compressor 3, and the condenser 4 are connected in a ring fashion in this order to form a refrigerant circuit 10.

[0037] The refrigerant evaporates in the evaporator 2. Thus, a gas-phase refrigerant (refrigerant vapor) is generated. The gas-phase refrigerant generated in the evaporator 2 is sucked into the compressor 3 through the suction pipe 6 and is compressed. The compressed gas-phase refrigerant is supplied to the condenser 4 through the discharge pipe 8. The gas-phase refrigerant is cooled in the condenser 4. Thus, a liquid-phase refrigerant (refrigerant liquid) is generated. The liquid-phase refrigerant is sent from the condenser 4 to the evaporator 2 through the return path 9.

[0038] As the refrigerant of the refrigeration cycle apparatus 100, a fluorocarbon refrigerant, a low GWP (Global Warming Potential) refrigerant, or a natural refrigerant can be used. Examples of the fluorocarbon refrigerant include HCFC (hydrochlorofluorocarbon) and HFC (hydrofluorocarbon). An example of the low GWP refrigerant is HFO-1234yf. Examples of the natural refrigerant include CO₂ and water.

[0039] For example, the refrigeration cycle apparatus 100 has, filled therein, a refrigerant consisting mainly of a substance having a negative saturated vapor pressure (an absolute pressure lower than atmospheric pressure) at normal temperature (Japanese Industrial Standards: 20°C ± 15°C/JIS Z8703). An example of such a refrigerant is a refrigerant consisting mainly of water. The term "consisting mainly of a substance" means that the substance is contained in the largest amount in terms of the ratio of mass.

[0040] If water is used as the refrigerant, the pressure ratio in the refrigeration cycle increases and, therefore, the degree of superheat of the refrigerant tends to be excessive. According to the present embodiment, the liquid-phase refrigerant is injected toward the refrigerant flow path inside the compressor 3 and, thus, an increase in the enthalpy of the refrigerant due to an increase in the degree of superheat of the refrigerant in the compression process is continuously suppressed. In this way, the work to be performed by the compressor 3 to increase the pressure of the refrigerant to a predetermined pressure can be significantly reduced. That is, the power consumption of the compressor 3 can be significantly reduced.

[0041] The refrigeration cycle apparatus 100 further includes a heat absorption circuit 12 and a heat radiation circuit 14.

[0042] The heat absorption circuit 12 is a circuit for using the liquid-phase refrigerant cooled by the evaporator 2. The heat absorption circuit 12 has necessary devices, such as a pump and an indoor heat exchanger. Part of the heat absorption circuit 12 is located inside the evaporator 2. Inside the evaporator 2, part of the heat absorption circuit 12 may be located above the liquid level of the liquid-phase refrigerant or may be located below the liquid level of the liquid-phase refrigerant. The heat absorption circuit 12 is filled with a heat medium, such as water or brine.

[0043] The liquid-phase refrigerant stored in the evaporator 2 is brought into contact with a member (a pipe) that constitutes the heat absorption circuit 12. Accordingly, heat exchange is performed between the liquid-phase refrigerant and the heat medium inside the heat absorption circuit 12 and, thus, the liquid-phase refrigerant evaporates. The heat medium inside the heat absorption circuit 12 is cooled by the latent heat of vaporization of the liquid-phase refrigerant. For example, if the refrigeration cycle apparatus 100 is an air conditioner that cools indoor air, the indoor air is cooled by the heat medium of the heat absorption circuit 12. The indoor heat exchanger is, for example, a fin tube heat exchanger.

[0044] The heat radiation circuit 14 is a circuit used to remove heat from the refrigerant inside the condenser 4. The

heat radiation circuit 14 has necessary devices, such as a pump and a cooling tower. Part of the heat radiation circuit 14 is located inside the condenser 4. More specifically, inside the condenser 4, the part of the heat radiation circuit 14 is located above the liquid level of the liquid-phase refrigerant. The heat radiation circuit 14 is filled with a heat medium, such as water or brine. If the refrigeration cycle apparatus 100 is an air conditioner that cools indoor air, the condenser 4 is disposed outside the room, and the refrigerant in the condenser 4 is cooled by the heat medium of the heat radiation circuit 14.

[0045] The high-temperature gas-phase refrigerant discharged from the compressor 3 is brought into contact with a member (a pipe) that constitutes the heat radiation circuit 14 inside the condenser 4. Accordingly, heat exchange is performed between the gas-phase refrigerant and the heat medium inside the heat radiation circuit 14 and, thus, the gas-phase refrigerant is condensed. The heat medium inside the heat radiation circuit 14 is heated by the latent heat of condensation of the gas-phase refrigerant. The heat medium heated by the gas-phase refrigerant is cooled by, for example, outside air or cooling water in a cooling tower (not illustrated) of the heat radiation circuit 14.

[0046] The evaporator 2 is composed of, for example, a container having heat insulation and pressure resistance. The evaporator 2 stores the liquid-phase refrigerant and evaporates the liquid-phase refrigerant therein. The liquid-phase refrigerant inside the evaporator 2 absorbs heat transferred from the outside of the evaporator 2 and, thus, evaporates. That is, the liquid-phase refrigerant heated by absorbing heat from the heat absorption circuit 12 evaporates in the evaporator 2. According to the present embodiment, the liquid-phase refrigerant stored in the evaporator 2 is brought into indirect contact with the heat medium circulating in the heat absorption circuit 12. That is, part of the liquid-phase refrigerant stored in the evaporator 2 is heated by the heat medium in the heat absorption circuit 12 and is used to heat the saturated liquid-phase refrigerant. The temperature of the liquid-phase refrigerant stored in the evaporator 2 and the temperature of the gas-phase refrigerant generated in the evaporator 2 are, for example, 5°C.

[0047] According to the present embodiment, the evaporator 2 is an indirect contact heat exchanger (for example, a shell tube heat exchanger). However, the evaporator 2 may be a direct contact heat exchanger, such as a spray type or a filler type heat exchanger. That is, the liquid-phase refrigerant may be heated by circulating the liquid-phase refrigerant through the heat absorption circuit 12. Alternatively, the heat absorption circuit 12 may be removed.

[0048] The compressor 3 sucks and compresses the gas-phase refrigerant generated by the evaporator 2. The compressor 3 is a dynamic compressor. A dynamic compressor is a compressor that gives momentum to a gas-phase refrigerant and, thereafter, increases the pressure of the gas-phase refrigerant by decelerating the gas-phase refrigerant. Examples of the dynamic compressor include a centrifugal compressor, a mixed flow compressor, and an axial flow compressor. Dynamic compressors are also referred to as turbo compressors. The compressor 3 may include a variable speed mechanism for changing the rotational speed. An example of the variable speed mechanism is an inverter that drives a motor of the compressor 3. The temperature of the refrigerant at the outlet of the compressor 3 is in the range of, for example, 100°C to 150°C.

[0049] The condenser 4 is composed of, for example, a container having heat insulation and pressure resistance. The condenser 4 condenses the gas-phase refrigerant compressed by the compressor 3 and stores a liquid-phase refrigerant generated by condensing the gas-phase refrigerant. According to the present embodiment, the gas-phase refrigerant is brought into indirect contact with the heat medium cooled by dissipating heat to the external environment and, thus, condenses. That is, the gas-phase refrigerant is cooled by the heat medium of the heat radiation circuit 14 and, thus, condenses. The temperature of the gas-phase refrigerant introduced into the condenser 4 is in the range of, for example, 100°C to 150°C. The temperature of the liquid-phase refrigerant stored in the condenser 4 is, for example, 35°C.

[0050] According to the present embodiment, the condenser 4 is an indirect contact heat exchanger (for example, a shell tube heat exchanger). However, the condenser 4 may be a direct contact type heat exchanger, such as a spray type or a filler type heat exchanger. That is, the liquid-phase refrigerant may be cooled by circulating the liquid-phase refrigerant through the heat radiation circuit 14. Alternatively, the heat radiation circuit 14 may be removed.

[0051] The suction pipe 6 is a flow path for leading the gas-phase refrigerant from the evaporator 2 to the compressor 3. The outlet of the evaporator 2 is connected to the suction port of the compressor 3 via the suction pipe 6.

[0052] The discharge pipe 8 is a flow path for leading the compressed gas-phase refrigerant from the compressor 3 to the condenser 4. The discharge port of the compressor 3 is connected to the inlet of the condenser 4 via the discharge pipe 8.

[0053] The return path 9 is a flow path for leading the liquid-phase refrigerant from the condenser 4 to the evaporator 2. The return path 9 connects the evaporator 2 to the condenser 4. A pump, a flow control valve, and the like may be disposed in the return path 9. The return path 9 can be constituted by at least one pipe.

[0054] The refrigerant supply path 11 connects the evaporator 2 to the compressor 3. The liquid-phase refrigerant stored in the evaporator 2 is supplied to the compressor 3 through the refrigerant supply path 11. The liquid-phase refrigerant is injected toward the refrigerant flow path inside the compressor 3. The refrigerant supply path 11 can be constituted by at least one pipe. In the evaporator 2, the inlet of the refrigerant supply path 11 is located below the liquid level of the liquid-phase refrigerant stored in the evaporator 2. A pump, a valve, and the like may be disposed in the refrigerant supply path 11.

[0055] The refrigeration cycle apparatus 100 may include a spare tank for storing a liquid-phase refrigerant. The spare tank is connected to, for example, the evaporator 2. The liquid-phase refrigerant is transferred from the evaporator 2 to the spare tank. The refrigerant supply path 11 connects the spare tank to the compressor 3 so that the liquid-phase refrigerant is supplied from the spare tank to the compressor 3. The spare tank may be connected to the suction pipe 6. In this case, the spare tank may store the liquid-phase refrigerant supplied from the inside of the refrigeration cycle or may store the liquid-phase refrigerant generated by being cooled by an external heat source via the inner peripheral surface of the suction pipe 6 or the like.

[0056] The structure of the compressor 3 is described in detail below.

[0057] As illustrated in Fig. 2, the compressor 3 is a centrifugal compressor. The compressor 3 includes a rotating body 27, a housing 35, and a shroud 37. The rotating body 27 is disposed in a space surrounded by the housing 35 and the shroud 37. A motor (not illustrated) for rotating the rotating body 27 may be disposed inside the housing 35.

[0058] The rotating body 27 includes a rotating shaft 25 and an impeller 26. The impeller 26 is attached to the rotating shaft 25. The impeller 26 rotates at high speed together with the rotating shaft 25. The impeller 26 may be formed integrally with the rotating shaft 25. The rotational speed of the rotating shaft 25 and the impeller 26 is in the range of, for example, 5000 rpm to 100000 rpm. The rotating shaft 25 is made of a strong iron-based material, such as S45CH. The impeller 26 is made of a material, such as aluminum, duralumin, iron, or ceramic.

[0059] The impeller 26 has a hub 30 and a plurality of blades 31. The hub 30 is a portion fitted to the rotating shaft 25. In a cross section including a central axis O of the rotating shaft 25, the hub 30 has a flared profile. The plurality of blades 31 are arranged on a surface 30p of the hub 30 in the circumferential direction of the rotating shaft 25.

[0060] The space around the impeller 26 includes a refrigerant flow path 40, a diffuser 41, and a volute chamber 42. The refrigerant flow path 40 is a flow path that is located around the rotating body 27 and that enables a gas-phase refrigerant to be compressed to pass therethrough. The refrigerant flow path 40 includes a suction flow path 36 and a plurality of inter-blade flow paths 38. The suction flow path 36 is located upstream of an upstream end 31t of the blade 31 in the flow direction of the gas-phase refrigerant. Each of the inter-blade flow paths 38 is located between the neighboring blades 31 in the circumferential direction of the rotating shaft 25. When the impeller 26 rotates, a speed in the rotational direction is given to the gas-phase refrigerant flowing through each of the plurality of inter-blade flow paths 38.

[0061] The diffuser 41 is a flow path for leading, to the volute chamber 42, the gas-phase refrigerant accelerated in the rotational direction by the impeller 26. The flow path cross-sectional area of the diffuser 41 increases from the refrigerant flow path 40 toward the volute chamber 42. This structure reduces the flow velocity of the gas-phase refrigerant accelerated by the impeller 26 and increases the pressure of the gas-phase refrigerant. The diffuser 41 is, for example, a vaneless diffuser consisting of a flow path extending in a radial direction. In order to effectively increase the pressure of the refrigerant, the diffuser 41 may be a vaned diffuser having a plurality of vanes and a plurality of flow paths partitioned by the vanes.

[0062] The volute chamber 42 is a volute space in which the gas-phase refrigerant that has passed through the diffuser 41 is collected. The compressed gas-phase refrigerant is led to the outside of the compressor 3 (the discharge pipe 8) via the volute chamber 42. The cross-sectional area of the volute chamber 42 increases in the circumferential direction. In this manner, the flow speed and the angular momentum of the gas-phase refrigerant in the volute chamber 42 are kept constant.

[0063] The shroud 37 covers the impeller 26 so as to define the refrigerant flow path 40, the diffuser 41, and the volute chamber 42. The shroud 37 is produced using an iron-based material or an aluminum-based material. Examples of the iron-based material include FC250, FCD400, SS400, and the like. An example of the aluminum-based material is ACD12 or the like.

[0064] The housing 35 plays a role of a casing that accommodates a variety of components of the compressor 3. The volute chamber 42 is formed by combining the housing 35 and the shroud 37. The housing 35 can be produced using the above-described iron-based material or aluminum-based material. If the diffuser is a vaned diffuser, the plurality of vanes can also be produced using the above-described iron-based material or aluminum-based material.

[0065] The housing 35 has a bearing 18 and a seal 29 arranged therein. The bearing 18 rotatably supports the rotating shaft 25. The bearing 18 may be a sliding bearing or a rolling bearing. If the bearing 18 is a sliding bearing, the refrigerant of the refrigeration cycle apparatus 100 can be used as a lubricant. The bearing 18 is connected to the housing 35 directly or via a bearing box (not illustrated). The seal 29 prevents the lubricant of the bearing 18 from flowing toward the impeller 26. An example of the seal 29 is a labyrinth seal.

[0066] The rotating body 27 has a main flow path 21 and an injection flow path 24 provided therein. The main flow path 21 extends in the axial direction of the rotating body 27 inside the rotating body 27. More specifically, the main flow path 21 is provided inside the rotating shaft 25 and extends in the axial direction of the rotating shaft 25. The injection flow path 24 branches off from the main flow path 21 inside the rotating body 27 and extends from the main flow path 21 to the refrigerant flow path 40. The main flow path 21 is connected to the evaporator 2 through the refrigerant supply path 11. The liquid-phase refrigerant introduced from the refrigerant supply path 11 located outside the rotating body 27 flows into the main flow path 21. The injection flow path 24 is a flow path that leads the liquid-phase refrigerant from

the main flow path 21 to the refrigerant flow path 40.

[0067] The liquid-phase refrigerant is supplied from the evaporator 2 to the main flow path 21 through the refrigerant supply path 11. The liquid-phase refrigerant is pressurized by centrifugal force and is injected toward the refrigerant flow path 40 inside the compressor 3 through the main flow path 21 and the injection flow path 24. When the liquid-phase refrigerant is brought into contact with the gas-phase refrigerant in the refrigerant flow path 40, heat exchange occurs between the liquid-phase refrigerant and the gas-phase refrigerant, and the superheated gas-phase refrigerant is continuously cooled by the sensible heat or evaporation latent heat of the liquid-phase refrigerant. For this reason, an increase in the enthalpy of the refrigerant caused by an increase in the degree of superheat of the refrigerant in the compression process is continuously suppressed. The compression power required by the compressor 3 can be reduced to less than the compression power required for perfectly adiabatic, isentropic compression. The work to be performed by the compressor 3 to increase the pressure of the refrigerant to a predetermined pressure can be significantly reduced. That is, the power consumption of the compressor 3 can be significantly reduced. As a result, the efficiency of the refrigeration cycle apparatus 100 is improved.

[0068] The main flow path 21 has an inflow port 21a located in an end face 25c of the rotating shaft 25. The end face 25c is an end face located opposite to an end face adjacent to the impeller 26. A liquid-phase refrigerant is introduced into the main flow path 21 through the inflow port 21a. According to such a configuration, the liquid-phase refrigerant can be smoothly fed into the main flow path 21. The main flow path 21 includes a central axis O of the rotating shaft 25. In the cross section of the rotating shaft 25, the main flow path 21 has, for example, a circular cross-sectional shape. In the cross section of the rotating shaft 25, the center of the main flow path 21 coincides with the central axis O. However, the center of the main flow path 21 may be offset from the central axis O of the rotating shaft 25. In the axial direction of the rotating shaft 25, the main flow path 21 extends to near an upper surface 26t of the impeller 26.

[0069] The refrigerant supply path 11 can be connected to a connection port 28 of the housing 35. The housing 35 has a buffer chamber 35h provided therein. The buffer chamber 35h communicates with the connection port 28, and a liquid-phase refrigerant is supplied from the refrigerant supply path 11 to the buffer chamber 35h. The end face 25c of the rotating shaft 25 faces the buffer chamber 35h. That is, the main flow path 21 opens toward the buffer chamber 35h. According to such a configuration, it is possible to smoothly feed the liquid-phase refrigerant from the refrigerant supply path 11 to the main flow path 21 via the buffer chamber 35h.

[0070] The location of the inflow port 21a of the main flow path 21 is not limited to the end face 25c of the rotating shaft 25. As described below, the inflow port 21a may be provided on a side surface of the rotating shaft 25. In this case, the buffer chamber 35h may surround the side surface of the rotating shaft 25 inside the housing 35. The detailed structure is described below with reference to Fig. 6.

[0071] The injection flow path 24 branches off from the main flow path 21 and extends in the radial direction of the rotating shaft 25. A centrifugal force acts on the liquid-phase refrigerant in the injection flow path 24. The liquid-phase refrigerant is injected into the refrigerant flow path 40 by the centrifugal force and is mixed with the gas-phase refrigerant sucked into the compressor 3. According to the present embodiment, the injection flow path 24 extends in a direction perpendicular to the axial direction of the rotating shaft 25. The injection flow path 24 has an outflow port 24b facing the refrigerant flow path 40. The outflow port 24b is located upstream of the upstream end 31t of the blade 31 in the flow direction of the gas-phase refrigerant. According to such a configuration, heat can be efficiently removed from the gas-phase refrigerant in the compression process. The injection flow path 24 may have an orifice shape so that the mist-like liquid-phase refrigerant is supplied to the refrigerant flow path 40.

[0072] According to the present embodiment, the outflow port 24b is located in the surface 30p of the hub 30 of the impeller 26. The injection flow path 24 penetrates the hub 30 in the radial direction of the rotating shaft 25. According to such a configuration, the gas-phase refrigerant and the liquid-phase refrigerant can be mixed with each other before the gas-phase refrigerant enters the inter-blade flow path 38 between the blades 31. In this manner, heat can be efficiently removed from the gas-phase refrigerant in the compression process.

[0073] The location of the outflow port 24b is not limited to the position illustrated in Fig. 2. The outflow port 24b may be located downstream of the upstream end 31t of the blade 31 in the flow direction of the gas-phase refrigerant. Alternatively, the outflow port 24b may be located upstream of the upper surface 26t of the impeller 26 in the flow direction of the gas-phase refrigerant. In this case, the outflow port 24b can be located on the side surface of the rotating shaft 25. By employing one of these configurations, heat can be removed from the gas phase refrigerant in the compression process.

[0074] According to the present embodiment, the injection flow path 24 includes a first portion 241 and a second portion 242. The first portion 241 is a portion extending from the main flow path 21 in the radial direction of the rotating shaft 25 inside the rotating shaft 25. The second portion 242 is a portion located between the first portion 241 and the refrigerant flow path 40. The first portion 241 is located inside the rotating shaft 25. The second portion 242 is located inside the impeller 26. According to such a configuration, a sufficient length of the injection flow path 24 can be ensured. The centrifugal acceleration applied to the liquid-phase refrigerant increases with increasing length of the injection flow path 24, and the liquid-phase refrigerant is more easily injected into the refrigerant flow path 40. If the tip of the rotating

shaft 25 protrudes in the axial direction from the upper surface 26t of the impeller 26, a component that differs from the impeller 26 may be attached to the tip of the rotating shaft 25, and the second portion 242 may be located inside the component.

[0075] If the outflow port 24b is located upstream of the upper surface 26t of the impeller 26, the second portion 242 is removed, and the injection flow path 24 may be composed of only the first portion 241.

[0076] The flow path cross-sectional area of the injection flow path 24 is less than the flow path cross-sectional area of the main flow path 21. According to such a configuration, mist-like liquid-phase refrigerant can be easily supplied to the refrigerant flow path 40.

[0077] As illustrated in Fig. 3, according to the present embodiment, a plurality (two or more) of the injection flow paths 24 are provided. The plurality of injection flow paths 24 extend radially from the main flow path 21. A liquid-phase refrigerant is injected from each of the injection flow paths 24 into the refrigerant flow path 40. According to such a configuration, the gas-phase refrigerant can be uniformly cooled in the circumferential direction of the rotating shaft 25. Note that if the compressor 3 has at least one injection flow path 24, the effects of the present disclosure can be obtained. The injection flow path 24 may extend parallel to the radial direction of the impeller 26 as in the present embodiment or may extend in a direction inclined with respect to the radial direction.

[0078] More specifically, the outflow ports 24b of the injection flow paths 24 are arranged at equal angular intervals in the circumferential direction of the rotating shaft 25. The outflow port 24b of each of the injection flow paths 24 is located between every two of the neighboring blades 31 arranged in the circumferential direction. A liquid-phase refrigerant is injected from each of the outflow ports 24b into one of the inter-blade flow paths 38 at a uniform flow rate. According to such a configuration, the gas-phase refrigerant can be cooled more uniformly in the circumferential direction of the rotating shaft 25. The number of outlets 24b may be different from or equal to the number of inter-blade flow paths 38. There may be a one-to-one correspondence between the outflow ports 24b of the injection flow paths 24 and the inter-blade flow paths 38.

[0079] If the plurality of blades 31 include a plurality of full blades and a plurality of splitter blades, each of the outflow ports 24b may be located between neighboring ones of the full blades in the circumferential direction of the rotating shaft 25. Alternatively, each of the outflow ports 24b may be located between neighboring full blade and splitter blade in the circumferential direction. A splitter blade is a blade that is shorter than a full blade. The plurality of full blades and the plurality of splitter blades may be alternately disposed on the surface 30p of the hub 30 in the circumferential direction of the rotating shaft 25.

[0080] According to the present embodiment, the rotating shaft 25 is fitted to the impeller 26 without any gap therebetween by a method such as shrink fitting or cold fitting. This prevents leakage of the liquid-phase refrigerant from a connection portion between the first portion 241 and the second portion 242 of the injection flow path 24. A seal structure, such as a seal ring, may be provided to prevent leakage.

[0081] The structure of the compressor 3 of the present disclosure is applicable to each of compressors of the multi-stage compressor. A desired effect can be obtained in the compressor in each stage. For example, if the compressor 3 is a multi-stage compressor including a plurality of impellers, each of the plurality of impellers may be provided with an injection flow path 24, and a liquid-phase refrigerant may be injected into the refrigerant flow path in each stage.

[0082] The operation and action performed by the refrigeration cycle apparatus 100 are described below.

[0083] If the refrigeration cycle apparatus 100 is left undisturbed for a certain period (for example, at all hours of the night), the temperature inside the refrigeration cycle apparatus 100 (the refrigerant circuit 10) is substantially the same as the ambient temperature. The pressure inside the refrigeration cycle apparatus 100 is substantially the same as a specific pressure. If the compressor 3 is started, the pressure inside the evaporator 2 gradually decreases, and the liquid-phase refrigerant absorbs heat from the heat medium of the heat absorption circuit 12 that exchanges heat with the inside air. As a result, the liquid-phase refrigerant evaporates and, thus, a gas-phase refrigerant is generated. The gas-phase refrigerant is sucked into the compressor 3, is compressed, and is discharged from the compressor 3. The high-pressure gas-phase refrigerant is introduced into the condenser 4 and is condensed by dissipating heat to the outside air or the like via the heat radiation circuit 14. Thus, a liquid-phase refrigerant is generated. The liquid-phase refrigerant is sent from the condenser 4 to the evaporator 2 through the return path 9.

[0084] Inside the compressor 3, the liquid-phase refrigerant is injected into the refrigerant flow path 40 through the main flow path 21 and the injection flow path 24. Heat exchange occurs between the gas-phase refrigerant pressurized by the compressor 3 and having an increased temperature and mist of the liquid-phase refrigerant. Thus, the superheated gas-phase refrigerant is continuously cooled by evaporation of the mist of the liquid-phase refrigerant. As a result, an increase in the enthalpy of the refrigerant caused by the increase in the degree of superheat of the refrigerant in the compression process is continuously suppressed. The compression power required by the compressor 3 can be reduced to less than the compression power required for perfectly adiabatic, isentropic compression. The work to be performed by the compressor 3 to increase the pressure of the refrigerant to a predetermined pressure can be significantly reduced. That is, the power consumption of the compressor 3 can be significantly reduced. As a result, the efficiency of the refrigeration cycle apparatus 100 is improved.

[0085] According to the present embodiment, the liquid-phase refrigerant stored in the evaporator 2 is supplied to the main flow path 21 of the compressor 3 through the refrigerant supply path 11. Mist of the liquid-phase refrigerant having substantially the same temperature as the temperature (saturation temperature) of the gas-phase refrigerant sucked into the compressor 3 is injected into the refrigerant flow path 40. In this case, flash evaporation of the liquid-phase refrigerant can be prevented and, thus, a rapid increase in vapor content inside the compressor 3 can be prevented. As a result, an increase in compression power due to an increase in vapor content is suppressed. Since the increase in the compression power due to the increase in the vapor content is suppressed, the effect of reducing the compressive power can be obtained without drastically reducing the refrigeration power by the above-described mechanism, even under operating conditions where the compressor input is excessive, such as during overload operation. In addition, chalking of the compressor 3 due to an increase in the vapor content can be prevented.

[0086] Fig. 20 is a flowchart illustrating a method for compressing a gas-phase refrigerant by using the compressor 3. In step S1, a gas-phase refrigerant is sucked into the compressor 3. The gas-phase refrigerant is sucked by the impeller 26 and flows through the suction flow path 36 of the refrigerant flow path 40 in a direction parallel to the central axis O. Therefore, the flow direction of the liquid-phase refrigerant in the main flow path 21 is opposite to the direction in which the gas-phase refrigerant is sucked into the compressor 3 and flows. In step S2, the sucked gas-phase refrigerant is accelerated in the compressor 3. More specifically, the gas-phase refrigerant is accelerated by the impeller 26. In step S4, the liquid-phase refrigerant is injected from the outflow port 24b of the injection flow path 24 toward the gas-phase refrigerant existing in the refrigerant flow path 40. The injected liquid-phase refrigerant is sucked into the inter-blade flow paths 38 of the compressor 3. Thus, the degree of superheat of the gas-phase refrigerant is decreased. The accelerated gas-phase refrigerant flows from the refrigerant flow path 40 toward the diffuser 41. In step S4, the static pressure of the gas-phase refrigerant is recovered in the diffuser 41.

[0087] Note that since the compressor 3 is a dynamic compressor, the steps described in the flowchart are not completely separated. The steps are performed continuously.

(Modifications)

[0088] Fig. 4A is a cross-sectional view of a rotating body according to a modification. Fig. 4B is a partial side view of a rotating shaft according to the modification. Fig. 4A corresponds to the cross-sectional view of Fig. 3. A rotating body 47 according to the present modification includes a rotating shaft 45 and an impeller 26. The impeller 26 is attached to the rotating shaft 45 and rotates together with the rotating shaft 45. The first portion 241 of the injection flow path 24 is connected to the second portion 242 on the side surface of the rotating shaft 45. At the connection position between the first portion 241 and the second portion 242, the angle range in which the first portion 241 exists in the circumferential direction of the rotating shaft 45 is larger than the angle range in which the second portion 242 exists in the circumferential direction of the rotating shaft 45. According to such a configuration, connection between the first portion 241 and the second portion 242 can be easily accomplished. Alignment of the first portion 241 with the second portion 242 in the circumferential direction of the rotating shaft 45 is easily achieved, and the impeller 26 is easily attached to the rotating shaft 45.

[0089] More specifically, the first portion 241 of the injection flow path 24 includes a radial portion 241a and a groove 241b. The radial portion 241a is a portion located inside the rotating shaft 45. The groove 241b is a portion provided on the side surface of the rotating shaft 45 which extends in the circumferential direction of the rotating shaft 45. The second portion 242 is connected to the groove 241b. According to such a configuration, the liquid-phase refrigerant can be supplied to each of the second portions 242 of the injection flow path 24 at a uniform flow rate. Because the groove 241b serves as a distributor, the number of the first portions 241 (the radial portions 241a) may differ from the number of the second portions 242. According to the present modification, the number of the first portions 241 is smaller than the number of the second portions 242. Furthermore, since the alignment of the first portion 241 with the second portion 242 in the circumferential direction of the rotating shaft 45 is extremely easily accomplished or is not needed, the work of attaching the impeller 26 to the rotating shaft 45 is facilitated. Note that the groove 241b need not be completely annular, and the groove 241b may be arc-shaped.

[0090] The liquid to be injected from the injection flow path 24 may be a liquid other than a refrigerant. Such a liquid may be any other liquid that can evaporate at the temperature of the gas-phase refrigerant and can cool the gas-phase refrigerant.

[0091] As illustrated in Fig. 5, in a compressor 50 according to another modification, the injection flow path 24 extends in a direction inclined with respect to both the radial direction and the axial direction of the rotating shaft 25. The outflow port 24b of the injection flow path 24 is located between the blades 31 of the impeller 26. According to such a configuration, the injected liquid-phase refrigerant easily flows along the flow of the gas-phase refrigerant between the blades 31. As a result, efficient heat exchange between the gas-phase refrigerant and the liquid-phase refrigerant can be expected to occur.

[0092] As illustrated in Fig. 6, in a compressor 60 according to another modification, the main flow path 21 has an

inflow port 21a located on a side surface of the rotating shaft 25. The connection port 28 of the housing 35 is provided at a position facing the side surface of the rotating shaft 25. In this manner, the inflow port 21a of the main flow path 21 may be located on the side surface of the rotating shaft 25.

5 (Advantageous Structure)

[0093] A dynamic compressor according to the present disclosure may have the structure described below.

[0094] To obtain a required pressure ratio in a dynamic compressor, it is necessary to increase the rotational speed and, thus, increase the peripheral speed of the impeller. The liquid-phase refrigerant discharged from the outflow port of the injection flow path does not have a certain particle size. The particle sizes of the liquid-phase refrigerant vary following a certain particle size distribution. The small-diameter particles follow the flow of the gas-phase refrigerant so as to flow out of the refrigerant flow path. Alternatively, the small-diameter particles evaporate before flowing out.

[0095] However, the Coriolis force acts in the circumferential direction in a coordinate system that rotates with the impeller. For large-sized refrigerant droplets, the Coriolis force exceeds the drag received from the gas-phase refrigerant. As a result, the refrigerant droplets may not follow the flow of the gas-phase refrigerant and collide with the trailing edge portion of the blade adjacent to the outflow port and, thus, erosion may occur in the impeller.

[0096] According to the configuration described below, erosion of the impeller caused by the collision of large-diameter refrigerant droplets injected from the outflow port can be prevented.

[0097] Fig. 7 is a plan projection view obtained by projecting the impeller 26 onto a plane perpendicular to the central axis O. Curves A_1B_1 and A_2B_2 represent the blade root line of a first blade 311 and the blade root line of a second blade 312 in the projection view, respectively. The outflow port 24b is located on the surface of the hub 30 between the first blade 311 and the second blade 312. In addition, the outflow port 24b is provided at a position of a radius R_1 from the central axis O, which is the center of rotation. The first blade 311 is a blade closest to the outflow port 24b in a rotational direction opposite to the rotational direction of the rotating body 27. The second blade 312 is a blade closest to the outflow port 24b in the rotational direction of the rotating body 27.

[0098] The term "blade root line" refers to a boundary line between the hub 30 and each of the blades. More specifically, since the blade has a thickness, the hub 30 and the blade are separated by a long narrow boundary plane. A blade root line means a line drawn in the length direction of the boundary plane such that the boundary plane is bisected in the thickness direction of the blade.

[0099] In the projection view illustrated in Fig. 7, the outflow port 24b is represented by a curved surface. A radius R_1 is represented by the distance between the central axis O and a point at which the curved surface is bisected.

[0100] An axis centered on the central axis O and passing through the outflow port 24b is defined as an r-axis, an angle in the rotational direction of the rotating body 27 is defined as θ (degrees), and a rotating polar coordinate system fixed to the impeller 26 is defined. In the present specification, the rotational direction of the rotating body 27 (the counterclockwise direction) is a positive direction, and the opposite rotational direction (the clockwise direction) is a negative direction. The angle formed by the outflow direction of the liquid-phase refrigerant and the r-axis is represented by an angle ϕ . In the example illustrated in Fig. 7, $\phi \neq 0$. The outflow direction of the liquid-phase refrigerant means the center direction of the injection of the liquid-phase refrigerant from the injection flow path 24. A trailing edge portion B_1 of the first blade 311 is located at a radius R_2 from the central axis O. An angle formed by a line OB_1 connecting the central axis O to the trailing edge portion B_1 and the r-axis, as measured from the r-axis in the rotational direction of the rotating body 27, is represented by an angle θ_{B1} . In Fig. 7, the angle θ_{B1} is a negative value. A trailing edge portion B_2 of the second blade 312 is located at a radius R_2 from the central axis O. An angle formed by a line OB_2 connecting the central axis O and the trailing edge portion B_2 and the r-axis, as measured from the r-axis in the rotational direction of the rotating body 27, is represented by an angle θ_{B2} . In Fig. 7, the angle θ_{B2} is a positive value.

[0101] The refrigerant droplet flowing out of the outflow port 24b at a velocity U at time zero flies in front of the rotating first blade 311, reaches the position of the radius R_2 at time t_p , and is discharged from the impeller 26. The angle formed by the direction OP of the position P and the r-axis, as measured from the r-axis in the rotational direction at this time, is denoted as an angle θ_p . The direction OP refers to a line OP connecting the central axis O to the position P.

[0102] When observed in the rotating polar coordinate system, the first blade 311 is stationary, but the centrifugal force and the Coriolis force act on the droplet. Consequently, the droplet follows a flight path turning right while accelerating in the r-axis direction. If $\theta_{B1} < \theta_p < \theta_{B2}$, the refrigerant droplets are discharged from the impeller 26 without colliding with the trailing edge portion.

[0103] Fig. 8 illustrates Fig. 7 in a stationary coordinate system. When observed in the stationary coordinate system, the first blade 311 rotates and follows the refrigerant droplet that linearly moves at a constant velocity U' from the r-axis in the direction of an angle α . The trailing edge portion B_1 of the first blade 311 moves to $\theta'_{B1} = \theta_{B1} + \omega t_p$. In contrast, the trailing edge portion B_2 of the second blade 312 moves to $\theta'_{B2} = \theta_{B2} + \omega t_p$. At the time t_p , the refrigerant droplet arrives at an intersection point P' between the extension of the straight line represented by the velocity U' and the outer edge of the impeller 26 having a radius of R_2 . Let θ'_p be the angle of the straight line OP' measured from the r-axis.

Then, if $\theta'_B < \theta'_P < \theta'_{B2}$, the refrigerant droplet is discharged from the impeller 26 without colliding with the trailing edge portion.

[0104] The velocity U is given by the centrifugal effect caused by the rotation of the rotating body 27. The velocity U is further increased by attaching a nozzle having a small sectional area to the outflow port 24b. The time t_p at which the refrigerant droplet reaches the trailing edge portion of the radius R_2 increases with decreasing velocity U , and the angle θ'_B at which the trailing edge portion moves by that time increases. For this reason, it is sufficient to consider the minimum velocity U . The total pressure that increases due to the centrifugal effect when the refrigerant droplets passing through the injection flow path 24 is given by $0.5\rho\omega^2(R_1^2 - R_0^2)$. The radius R_0 of the main flow path 21 is sufficiently small as compared with the radius R_1 and, thus, can be ignored. Accordingly, the total pressure that increases by the centrifugal effect when the refrigerant droplets pass through the injection flow path 24 is given by $0.5\rho\omega^2 R_1^2$. When the nozzle is not attached, the velocity U is the smallest. In that case, an increase in the total pressure becomes exactly the dynamic pressure. Therefore, $U = R_1\omega$ holds.

[0105] Since the centrifugal force $r\omega^2$ acts in the r direction on the refrigerant droplet flowing out of the outflow port 24b and flying in the radius r , the following equation (1) is obtained from the equation of motion.

$$\frac{d^2 r}{dt^2} = \frac{du_r}{dt} = r\omega^2 \quad (1)$$

[0106] Among the solutions of Equation (1), a solution that satisfies $r = R_1$ and $u_r = U\cos\phi = R_1\omega\cos\phi \approx R_1\omega$ at $t = 0$ is given by the following equation (2).

[Math. 2]

$$r = R_1 e^{\omega t} \quad (2)$$

[0107] The time t_p at which $r = R_2$ is given by Equation (3).

[Math. 3]

$$t_p = \frac{1}{\omega} \ln(R_2/R_1) \quad (3)$$

[0108] However, the approximation of $\cos\phi \approx 1$ used above holds when $\phi \approx 0$ degrees. In addition, this approximation slightly increases t_p , which results in a condition where the refrigerant droplets are more likely to collide with the trailing edge portion than in reality. However, if such consideration is made, the design is on the safe side.

[0109] In a rotating coordinate system, the velocity vector of velocity U at the angle ϕ is a velocity vector of the angle α and velocity U' in the stationary coordinate system, as illustrated in the upper right hand corner of Fig. 8. The velocity U' is given by the following equation (4).

[Math. 4]

$$U'^2 = U^2 + (U\sin\phi + R_1\omega)^2 \quad (4)$$

[0110] Here again, if $U = R_1\omega$, the following equation (5) holds.

[Math. 5]

$$U' = R_1\omega\sqrt{2(\sin\phi + 1)} \quad (5)$$

[0111] In addition, the angle α is obtained by the following equation (6).

[Math. 6]

$$\tan\alpha = \frac{U\sin\phi + R_1\omega}{U\cos\phi} = \frac{\sin\phi + 1}{\cos\phi} \quad (6)$$

[0112] From the sine theorem, the following equation (7) holds for the triangle formed by the straight line OP and the outflow port 24b.
[Math. 7]

$$\sin \gamma = \frac{R_1}{R_2} \sin \alpha \quad (7)$$

[0113] Therefore, as can be seen from Fig. 8, Equation (8) holds.
[Math. 8]

$$\theta'_P = \alpha - \gamma = \tan^{-1} \frac{\sin \varphi + 1}{\cos \varphi} - \sin^{-1} \left(\frac{R_1}{R_2} \sin \left(\tan^{-1} \frac{\sin \varphi + 1}{\cos \varphi} \right) \right) \quad (8)$$

[0114] If $\theta'_{B1} < \theta'_P < \theta'_{B2}$ holds, that is, if the angle ϕ representing the outflow direction of the injection flow path 24 satisfies the following relation equation (9), then the first blade 311 and the second blade 312 do not collide with the refrigerant droplets.
[Math. 9]

$$\theta_{B1} + \ln \left(\frac{R_2}{R_1} \right) < \tan^{-1} \frac{\sin \varphi + 1}{\cos \varphi} - \sin^{-1} \left(\frac{R_1}{R_2} \sin \left(\tan^{-1} \frac{\sin \varphi + 1}{\cos \varphi} \right) \right) < \theta_{B2} + \ln \left(\frac{R_2}{R_1} \right) \quad (9)$$

[0115] Since the outflow port 24b is present on the surface of the hub 30 near the inlet of the refrigerant flow path of the impeller 26, the ratio (R_2/R_1) is a value in the range of 3 to 6. In addition, the angle θ_{B2} representing the position of the trailing edge portion B_2 of the second blade 312 is normally less than or equal to +20 degrees. In this range, the above condition for θ'_{B2} on the right side is satisfied even when $\phi > 90^\circ$ which is the physical upper limit value. Consequently, the droplets do not collide with the second blade 312.

[0116] The lower limit value of the angle ϕ is determined by the collision condition regarding the first blade 311 on the left side above. The lower limit value of the range of the angle ϕ depends on the ratio (R_2/R_1) . If $3 \leq (R_2/R_1) \leq 6$, the lower limit value of the range of the angle ϕ is minimum when $R_2/R_1 = 3$.

[0117] Fig. 9 illustrates an outflow angle ϕ that satisfies the collision condition on the left side above for an angle θ_{B1} representing the position of the trailing edge portion B_1 of the first blade 311. The angle θ_{B1} is set to -40 degrees or more in a general design of the impeller 26. At this time, the condition $\phi \geq -25^\circ$ is necessary for preventing the impeller 26 from colliding with the trailing edge portion B_1 of the first blade 311. The upper limit value of the angle ϕ is determined so as to be in the range that enables drilling. For example, the upper limit value is 60 degrees.

[0118] As described above, collision between a refrigerant droplet and the trailing edge portion can be prevented by setting the outflow direction ϕ of the injection flow path 24 $\geq -25^\circ$. As a result, erosion of the impeller 26 caused by collision of liquid droplets can be prevented.

(Another Modification)

[0119] The case where the technology of the present disclosure is applied to a multi-stage dynamic compressor is described below. The same reference symbols are used for elements of the modification that are the same as those of the compressor 3 illustrated in Fig. 2, and description of the elements is not necessarily repeated. Descriptions of the compressors can be interchangeably used as long as no technical conflicts occur. The configurations of the compressor may be combined with one another as long as no technical conflicts occur.

[0120] A multi-stage compressor is designed within a range of an optimum specific speed NS that can achieve highly efficient operation. The gas-phase refrigerant is compressed in each of the stages and, thus, is gradually reduced in volume. For this reason, in general, the pressure ratio in the subsequent stage can be set to be less than or equal to the pressure ratio in the previous stage. In other words, to reduce the degree of superheat to below that in the perfectly adiabatic, isentropic compression, the degree of superheat to be removed in the subsequent stage can be less than the degree of superheat to be removed in the previous stage. Therefore, the injection quantity of the liquid-phase refrigerant in a subsequent stage can be set to be less than or equal to the injection quantity of the liquid-phase refrigerant in the previous stage.

[0121] However, in the multi-stage compressor, if the radial position distance of the injection flow path in the subsequent stage is larger than the radial position distance of the injection flow path in the previous stage, the injection quantity in

the subsequent stage at a constant rotational angular velocity becomes excessive. Some of the liquid-phase refrigerant that cannot be completely evaporated by the degree of superheat of the gas-phase refrigerant and that has a large particle diameter so as not to follow the gas-phase refrigerant may collide with the wall surface of the impeller and remain there. The latent heat of evaporation when the remaining liquid-phase refrigerant evaporates due to the heat of the wall surface of the impeller does not contribute to the refrigeration power of the system and, thus, the theoretical power of the compressor increases and the COP decreases.

[0122] The present inventors have intensively studied the above-described problem and conceived a technique for a multi-stage compressor to prevent the liquid-phase refrigerant that has a large particle size and that does not follow the gas-phase refrigerant from colliding with the wall surface of the impeller and remaining around the wall surface, due to excessive injection of liquid-phase refrigerant.

[0123] The details are described below.

[0124] Fig. 10 illustrates a cross section of a multi-stage dynamic compressor 70 according to another modification. According to the present modification, the compressor 70 is a two-stage compressor. However, the compressor 70 may have three or more stages.

[0125] As illustrated in Fig. 10, the compressor 70 is a multi-stage centrifugal compressor. The compressor 70 includes a rotating body 77, a housing 35, and a shroud 37. The rotating body 77 is disposed in a space surrounded by the housing 35 and the shroud 37. A motor and a bearing (neither is illustrated) for rotating the rotating body 77 may be disposed inside the housing 35.

[0126] The rotating body 77 includes a rotating shaft 25, a first impeller 26, and a second impeller 71. The first impeller 26 and the second impeller 71 are attached to the rotating shaft 25 and rotate together with the rotating shaft 25 at high speed. The first impeller 26 and the second impeller 71 may be formed integrally with the rotating shaft 25. The rotational speed of the rotating shaft 25, the first impeller 26, and the second impeller 71 is, for example, in the range of 5000 rpm to 100000 rpm. The rotating shaft 25 is produced using a strong iron-based material, such as S45CH. The first impeller 26 and the second impeller 71 are produced using a material, such as aluminum, duralumin, iron, or ceramic.

[0127] The direction of the first impeller 26 is the same as the direction of the second impeller 71. In other words, both the upper surface of the first impeller 26 and the upper surface of the second impeller 71 are located on the same side in a direction parallel to the rotating shaft 25. However, the first impeller 26 may be attached to one end of the rotating shaft 25, and the second impeller 71 may be attached to the other end of the rotating shaft 25. In this case, the upper surface of the first impeller 26 is located on the side opposite to the upper surface of the second impeller 71 in the direction parallel to the rotating shaft 25. The back surface of the first impeller 26 and the back surface of the second impeller 71 face each other.

[0128] The space around the first impeller 26 and the second impeller 71 includes a refrigerant flow path 40, a refrigerant flow path 80, a first diffuser 41, a second diffuser 51, a volute chamber 42, and a return channel 79. The refrigerant flow path 40 and the refrigerant flow path 80 are flow paths that are located around the rotating body 27 and that enables a gas-phase refrigerant to be compressed to flow therethrough. The refrigerant flow path 40 includes a suction flow path 36 and a plurality of inter-blade flow paths 38. The refrigerant flow path 80 includes a suction flow path 76 and a plurality of inter-blade flow paths 78. When the first impeller 26 and the second impeller 71 rotate, a speed in the rotational direction is given to the gas-phase refrigerant flowing through each of the plurality of inter-blade flow paths 38 and the inter-blade flow paths 78.

[0129] The first diffuser 41 is provided so as to surround the first impeller 26. The second diffuser 51 is provided so as to surround the second impeller 71. The first diffuser 41 is a flow path for leading the gas-phase refrigerant accelerated in the rotational direction by the first impeller 26 to the return channel 79. The second diffuser 51 is a flow path for leading the gas-phase refrigerant accelerated in the rotational direction by the second impeller 71 to the volute chamber 42. The flow path cross-sectional area of the first diffuser 41 increases from the refrigerant flow path 40 toward the return channel 79. The flow path cross-sectional area of the second diffuser 51 increases from the refrigerant flow path 80 toward the volute chamber 42. This structure reduces the flow velocity of the gas-phase refrigerant accelerated by the first impeller 26 and the second impeller 71 and increases the pressure of the gas-phase refrigerant. Each of the first diffuser 41 and the second diffuser 51 is, for example, a vaneless diffuser formed by a flow path extending in a radial direction. To effectively increase the pressure of the refrigerant, each of the first diffuser 41 and the second diffuser 51 may be a vaned diffuser having a plurality of vanes and a plurality of flow paths partitioned by the vanes.

[0130] The return channel 79 is a flow path that leads the gas-phase refrigerant compressed when passing through the first impeller 26 to the second impeller 71. The return channel 79 extends inward from the first diffuser 41 toward the suction flow path 76.

[0131] The volute chamber 42 is a voluted space in which the gas-phase refrigerant that has passed through the second diffuser 51 is collected. The compressed gas-phase refrigerant is led to the outside of the compressor 70 (the discharge pipe 8) via the volute chamber 42. The cross-sectional area of the volute chamber 42 increases in the circumferential direction. Thus, the flow velocity and the angular momentum of the gas-phase refrigerant in the volute chamber 42 are kept constant.

[0132] The shroud 37 covers the first impeller 26 and the second impeller 71 and defines the refrigerant flow path 40, the first diffuser 41, the second diffuser 51, the volute chamber 42, and the return channel 79. The shroud 37 is produced using an iron-based material or an aluminum-based material. Examples of the iron-based material include FC250, FCD400, SS400, and the like. An example of the aluminum-based material is ACD12 or the like.

[0133] The housing 35 plays a role of a casing that houses a variety of components of the compressor 70. The volute chamber 42 is formed by combining the housing 35 and the shroud 37. The housing 35 can be produced using the above-described iron-based material or aluminum-based material. If the diffuser is a vaned diffuser, the plurality of vanes can also be produced using the iron-based or aluminum-based materials described above.

[0134] The rotating body 77 has a main flow path 21, a first injection flow path 24, and a second injection flow path 74 provided therein. The main flow path 21 extends in the axial direction of the rotating body 27 inside the rotating body 27. More specifically, the main flow path 21 is provided inside the rotating shaft 25 and extends in the axial direction of the rotating shaft 25. The first injection flow path 24 branches off from the main flow path 21 inside the first impeller 26 and extends from the main flow path 21 to the refrigerant flow path 40. The second injection flow path 74 branches off from the main flow path 21 inside the second impeller 71 and extends from the main flow path 21 to the refrigerant flow path 80. The main flow path 21 is connected to the evaporator 2 through the refrigerant supply path 11. The liquid-phase refrigerant introduced from the refrigerant supply path 11 located outside the rotating body 27 flows into the main flow path 21. The first injection flow path 24 is a flow path that leads the liquid-phase refrigerant from the main flow path 21 to the refrigerant flow path 40. The second injection flow path 74 is a flow path that leads the liquid-phase refrigerant from the main flow path 21 to the refrigerant flow path 80.

[0135] The liquid-phase refrigerant is supplied from the evaporator 2 to the main flow path 21 through the refrigerant supply path 11. The liquid-phase refrigerant is pressurized by centrifugal force and is injected toward the refrigerant flow path 40 and the refrigerant flow path 80 inside the compressor 70 through the main flow path 21, the first injection flow path 24, and the second injection flow path 74. When the liquid-phase refrigerant is brought into contact with the gas-phase refrigerant in the refrigerant flow path 40 and the refrigerant flow path 80, heat exchange occurs between the liquid-phase refrigerant and the gas-phase refrigerant. Thus, the superheated gas-phase refrigerant is continuously cooled by the sensible heat or evaporative latent heat of the liquid-phase refrigerant.

[0136] Fig. 11 illustrates a cross section of the first impeller 26 at a location including an outflow port 24b and a cross section of the second impeller 71 at a location including an outflow port 74b. Let S_1 be the opening area of the outflow port 24b of the first injection flow path 24, let S_2 be the opening area of the outflow port 74b of the second injection flow path 74, let R_1 be the radial distance from the central axis O of the rotating body 77 to the outflow port 24b, and let R_2 be the radial distance from the central axis O of the rotating body 77 to the outflow port 74b. Then, the compressor 70 satisfies the relationship of $(R_2/R_1 \leq S_1/S_2)$.

[0137] The opening area S_1 can be a flow path cross-sectional area of the first injection flow path 24. The opening area S_2 can be the flow path cross-sectional area of the second injection flow path 74. A radial distance R_1 means the distance from the central axis O to the center or the center of gravity of the outflow port 24b. A radial distance R_2 means the distance from the central axis O to the center or the center of gravity of the outflow port 74b.

[0138] As illustrated in Fig. 11, if $R_1 \leq R_2$, the radius of the hub 30 of the first impeller 26, that is, the radial distance R_1 of the outflow port 24b is decreased and, thus, the inlet area of the gas-phase refrigerant is increased, for example. In this manner, the inlet Mach number can be decreased. As a result, high efficiency operation can be performed.

[0139] As illustrated in Fig. 12, a centrifugal force acts on the liquid-phase refrigerant inside the main flow path 21, and a pressure gradient dp/dr is formed in the radial direction so as to balance the centrifugal force. The balance between the two in the radial direction is expressed by the following equation (10).

[Math. 10]

$$\frac{dp}{dr} = \rho \omega^2 r \quad (10)$$

[0140] By integrating the equation (10) from a radius of zero to r , the pressure is given by $P_1 = (\rho \omega^2 r^2)/2$. The supply static pressure of the liquid-phase refrigerant is set to $P_s = 0$. Note that the pressure head due to the gravity force is so small as to be negligible, as compared with the pressure head due to centrifugal force. Accordingly, the pressure head due to the gravity force is ignored.

[0141] Let v be the outflow velocity, and let A be the cross-sectional area of the outflow port. Then, in terms of the liquid-phase refrigerant present inside the injection flow path, considering a micro-columnar portion of the thickness dr in the injection direction, the acceleration in the injection direction is given as $a = dv/dr$ from $a = dv/dt$ and $dt = dr/v$.

[0142] The forces acting in the injection direction are the centrifugal force $(\rho A \omega^2 r dr)$ and the force $-A(dp/dr)dr$ due to the pressure difference between the front and back of the micro-columnar portion. Accordingly, the equation of motion in the injection direction is given by the following equation (11).

[Math. 11]

$$\rho A dr v dv/dr = \rho A \omega^2 r dr - A(dp/dr) dr \quad (11)$$

[0143] By integrating equation (11), the following equation (12) is obtained.

[Math. 12]

$$\frac{v^2}{2} - \frac{\omega^2 r^2}{2} + \frac{P}{\rho} = \text{const.} \quad (12)$$

[0144] If the inlet of the injection flow path is denoted by the suffix "1", and the outlet of the injection flow path is denoted by the suffix "2", the following equation (13) is obtained.

[Math. 13]

$$\frac{v_1^2}{2} - \frac{\omega^2 r^2}{2} + \frac{P_1}{\rho} = \frac{v_2^2}{2} - \frac{\omega^2 R^2}{2} + \frac{P_2}{\rho} \quad (13)$$

[0145] Herein, if $v_1 = 0$ and $P_1 = (\rho \omega^2 r^2)/2$, the outflow velocity v_2 is given by the following equation (14).

[Math. 14]

$$v_2^2 = \omega^2 R^2 - \frac{2P_2}{\rho} \quad (14)$$

[0146] The flow rate Q is obtained as the product of the opening area S (S = the number N of the injection flow paths $\times$ the cross-sectional area A of the outflow port) and the outflow velocity v_2 . Assuming that the liquid-phase refrigerant flows out at the theoretical flow rate (the loss is ignored), the flow rate Q is given by the following equation (15).

[Math. 15]

$$Q = S \sqrt{\omega^2 R^2 - \frac{2P_2}{\rho}} \quad (15)$$

[0147] Since in reality, there are a variety of losses, let C be the flow coefficient. Then, equation (15) can be defined as the following equation (16).

[Math. 16]

$$Q = CS \sqrt{\omega^2 R^2 - \frac{2P_2}{\rho}} \quad (16)$$

[0148] Herein, P_2 denotes the vapor pressure of the gas-phase refrigerant flowing through the refrigerant flow path. P_2 is so small as to be negligible, as compared with the pressure head due to centrifugal force. Thus, P_2 can be ignored, and the equation $Q = CS\omega R$ holds.

[0149] That is, the injection flow rate Q is proportional to the product of the opening area S of the outflow port, the rotational angular velocity ω , and the radial distance R from the central axis to the outflow port.

[0150] Since in the multi-stage compressor 70, the gas-phase refrigerant is compressed and is gradually reduced in volume as it passes through the stages, the pressure ratio in a stage can be set to be less than or equal to the pressure ratio in the immediately previous stage. In other words, to reduce the degree of superheat to below the perfectly adiabatic, isentropic compression, the degree of superheat to be removed in a stage can be less than or equal to the degree of superheat to be removed in the immediately previous stage. Therefore, the injection quantity of the liquid-phase refrigerant in a stage can be set to be less than or equal to the injection quantity of the liquid-phase refrigerant in the immediately previous stage.

[0151] That is, the expression (the injection quantity Q_1 of the liquid phase refrigerant in a stage) $\geq$ (the injection quantity Q_2 of the liquid phase refrigerant in the immediately subsequent stage) can hold.

[0152] As described above, if the relationship of $S_1 \times R_1 \geq S_2 \times R_2$, that is, the relationship of $R_2/R_1 \leq S_1/S_2$ is satisfied, the injection quantity from the second injection flow path 74 of the second impeller 71 is less than or equal to the injection quantity from the injection flow path 24 of the first impeller 26, since the rotational angular velocity ω is constant.

[0153] In this way, the liquid-phase refrigerant injected in an amount commensurate with the degree of superheat to be removed reliably evaporates in the refrigerant flow path.

[0154] Therefore, in the multi-stage compressor 70, it is possible to prevent the liquid-phase refrigerant that has a large particle diameter and that does not follow the gas-phase refrigerant due to excessive injection of the liquid-phase refrigerant from colliding with the wall surface of the impeller and remaining around the wall surface.

(Another Modification)

[0155] Even in the multistage compressor, a problem of erosion of an impeller due to refrigerant droplets arises.

[0156] When the liquid-phase refrigerant is injected into the refrigerant flow path around the impeller in the first-stage in an amount necessary to remove the degree of superheat generated in each stage of the multi-stage compressor, the amount of droplets present around the impeller in the first stage is excessive. As a result, the probability of a refrigerant droplet colliding with the impeller increases and, thus, the risk of erosion of the impeller increases.

[0157] In the compressor 70 described with reference to Fig. 10, the first impeller 26 and the second impeller 71 are provided with the injection flow path 24 and the injection flow path 74, respectively. This configuration is effective in preventing erosion of the impeller. However, there is still room for improvement.

[0158] As a result of further studies by the present inventors, the present inventors have found a configuration capable of injecting a more appropriate amount of liquid-phase refrigerant in a multi-stage compressor. The configuration is described below.

[0159] Fig. 13 illustrates a cross section of a multi-stage dynamic compressor 90 according to another modification. The difference between the compressor 70 described with reference to Fig. 10 and the compressor 90 according to the present modification lies in the number and locations of the injection flow paths.

[0160] As illustrated in Fig. 13, the rotating body 77 has a main flow path 21, a first injection flow path 24, a downstream injection flow path 32, and a second injection flow path 74 provided therein. The main flow path 21 extends in the axial direction of the rotating body 77 inside the rotating body 77. More specifically, the main flow path 21 is provided inside the rotating shaft 25 and extends in the axial direction of the rotating shaft 25.

[0161] The first injection flow path 24 is located inside the first impeller 26, branches off from the main flow path 21, and extends from the main flow path 21 to the refrigerant flow path 40. The first injection flow path 24 is located upstream of the inter-blade flow path 38 in the flow direction of the gas-phase refrigerant. The first injection flow path 24 is provided upstream of the upstream end 31t of the blade of the first impeller 26. By injecting the liquid-phase refrigerant from the first injection flow path 24 toward the refrigerant flow path 40, only the amount of liquid-phase refrigerant necessary to remove the degree of superheat generated in the first impeller 26 can be supplied. The first injection flow path 24 may be provided downstream of the upstream end 31t of the blade of the first impeller 26.

[0162] The downstream injection flow path 32 is located inside the first impeller 26, branches off from the main flow path 21, and extends from the main flow path 21 to the refrigerant flow path 40. The downstream injection flow path 32 is located downstream of the first injection flow path 24 in the flow direction of the gas-phase refrigerant. The central axis of the downstream injection flow path 32 intersects with the inlet of the first diffuser 41. An outflow port 32b of the downstream injection flow path 32 is located on the surface of the hub 30 of the first impeller 26. The downstream injection flow path 32 penetrates the hub 30 in the radial direction of the rotating shaft 25. The outflow port 32b faces the inlet of the first diffuser 41.

[0163] The amount of liquid-phase refrigerant necessary for removing the degree of superheat generated in the second impeller 71 is injected through the downstream injection flow path 32. The liquid-phase refrigerant injected from the downstream injection flow path 32 partially evaporates in the first diffuser 41. The second impeller 71 sucks the liquid-phase refrigerant in an amount necessary only to remove the degree of superheat generated in the second impeller 71. The amount of refrigerant droplets present in each of the refrigerant flow path 40 around the first impeller 26 and the refrigerant flow path 80 around the second impeller 71 decreases. As a result, the probability of collision of the refrigerant droplets with the first impeller 26 and the second impeller 71 is reduced and, thus, the risk of erosion of the first impeller 26 and the second impeller 71 is reduced.

[0164] The second injection flow path 74 is located inside the second impeller 71, branches off from the main flow path 21, and extends from the main flow path 21 to the refrigerant flow path 80. The central axis of the second injection flow path 74 intersects with the inlet of the second diffuser 51. The outflow port 74b of the second injection flow path 74 is located on the surface of a hub 33 of the second impeller 71. The second injection flow path 74 penetrates the hub 33 in the radial direction of the rotating shaft 25. The outflow port 74b faces the inlet of the second diffuser 51. According

to the second injection flow path 74, heat can also be removed from the gas-phase refrigerant in the second diffuser 51 at the time of performing pressure recovery. This configuration is also effective for a multi-stage dynamic compressor having three or more stages.

[0165] The term "central axis of the injection flow path" refers to an axis that passes through the center or the center of gravity of the cross section of the injection flow path and extends parallel to the injection flow path. The term "inlet of the diffuser" refers to an inlet to a space that serves as a diffuser.

[0166] The liquid-phase refrigerant is supplied from the evaporator 2 or the condenser 4 to the main flow path 21 through the refrigerant supply path 11. The first injection flow path 24, the downstream injection flow path 32, and the second injection flow path 74 are flow paths that lead the liquid-phase refrigerant from the main flow path 21 to the refrigerant flow path 40 and the refrigerant flow path 80. The liquid-phase refrigerant is pressurized by centrifugal force and is injected toward the refrigerant flow path 40 and the refrigerant flow path 80 in the compressor 90 through the main flow path 21, the first injection flow path 24, the downstream injection flow path 32, and the second injection flow path 74. When the liquid-phase refrigerant is brought into contact with the gas-phase refrigerant in the refrigerant flow path 40 and the refrigerant flow path 80, heat exchange occurs between the liquid-phase refrigerant and the gas-phase refrigerant. Thus, the superheated gas-phase refrigerant is continuously cooled by the sensible heat or the evaporative latent heat of the liquid-phase refrigerant.

(Another Modification)

[0167] Fig. 14 illustrates a compressor 3a obtained by adding a motor 16 to the compressor 3 described with reference to Fig. 2. In addition to the configuration of the compressor 3, the compressor 3a has the motor 16 attached to the rotating shaft 25. The motor 16 is disposed inside the housing 35. The motor 16 includes a rotor 16a and a stator 16b. The rotor 16a is fixed to the rotating shaft 25. When the motor 16 is driven, the rotating body 27 rotates. Bearings 18a and 18b that support the rotating shaft 25 are disposed on either side of the motor 16.

[0168] The temperature of the liquid-phase refrigerant in the main flow path 21 provided inside the rotating shaft 25 is increased by the exhaust heat of the motor 16. Since the liquid-phase refrigerant is centrifugally pressurized in the injection flow path 24 inside the rotating body 27 of the compressor 3a which rotates at high speed, the power of the motor 16 is further increased and, thus, the temperature increase increases. The heat value of the rotor 16a of the motor 16 is, for example, about 0.8 kW when the refrigeration power is 880 kW as a rated condition. In particular, under the high load operating condition, the rotational speed of the compressor 3a increases, and the amount of exhaust heat from the motor 16 increases in accordance with the rotational speed. For this reason, the liquid-phase refrigerant may evaporate inside the main flow path 21, and the gas-phase refrigerant may remain inside the main flow path 21. In this case, the main flow path 21 is clogged by the gas-phase refrigerant, and the liquid-phase refrigerant does not flow. As a result, the motor 16 cannot be continuously cooled, and the efficiency of the motor 16 decreases.

[0169] The present modification solves the above-mentioned problem and provides a technique for preventing clogging of the flow path due to evaporation of the liquid-phase refrigerant inside the main flow path while reducing the compressor power due to an increase in enthalpy during compression. At the same time, the motor is continuously cooled to improve the efficiency of the motor.

[0170] Fig. 15 illustrates a cross section of the compressor 3b that can solve the problem by the heat generated by the motor 16. The compressor 3b includes a motor 16 having a rotor 16a and a stator 16b. The rotor 16a is fixed to the rotating shaft 25 between the impeller 26 and the bearing 18b in the axial direction of the rotating shaft 25. The rotor 16a is made from a steel material, such as a silicon nitride steel plate. The stator 16b is disposed so as to surround the rotor 16a in the circumferential direction of the rotating shaft 25. A rotating magnetic field induced by the stator 16b generates a rotational torque in the rotor 16a. The rotational torque drives the rotating shaft 25 and the impeller 26 to rotate at high speed.

[0171] The buffer chamber 35h is provided so as to be in contact with the inflow port 21a. The buffer chamber 35h communicates with the main flow path 21.

[0172] The buffer chamber 35h is described in detail below.

[0173] As illustrated in Fig. 15, the compressor 3b further includes a supply tank 20 and a pressure pump 19. The buffer chamber 35h is connected to a refrigerant supply path 22 provided outside the housing 35. The refrigerant supply path 22 enables the buffer chamber 35h and the supply tank 20 to communicate with each other. The refrigerant supply path 22 is provided with the pressure pump 19 for pumping the liquid-phase refrigerant stored in the supply tank 20 to the buffer chamber 35h. The temperature of the liquid-phase refrigerant in the supply tank 20 is, for example, 35°C.

[0174] Examples of the supply tank 20 include a condenser, an evaporator, and other buffer tanks.

[0175] The pressure pump 19 is a pump for pressurizing the liquid-phase refrigerant in the supply tank 20 and supplying the liquid-phase refrigerant to the buffer chamber 35h. The supply pressure of the liquid-phase refrigerant is, for example, about 25 kPa to 100 kPa. The pressure pump 19 may be a positive displacement pump or a dynamic pump. A positive displacement pump is a pump that sucks and discharges a liquid-phase refrigerant by a change in volume and increases

the pressure of the refrigerant. Examples of a positive displacement pump include a rotary pump, a screw pump, a scroll pump, a vane pump, and a gear pump. A dynamic pump is a pump that gives momentum to a liquid-phase refrigerant and reduces the speed of the liquid-phase refrigerant to increase the pressure of the refrigerant. Examples of a dynamic pump (a turbo pump) includes a centrifugal pump, a mixed flow pump, and an axial flow pump. In addition, a cascade pump, a hydrocera pump, or the like may be used. The pressure pump 19 may be a mechanism including a motor driven by a pump controller, such as an inverter, and capable of changing the rotational speed. The supply pressure of the pressure pump 19 is adjusted in consideration of the pressure loss of the main flow path 21 and the refrigerant supply path 22. The liquid-phase refrigerant is pumped such that the pressure rises above the pressure caused by evaporation in the main flow path 21 for the flow rate of the liquid-phase refrigerant required for cooling in accordance with the operating conditions.

[0176] The liquid-phase refrigerant that cools the gas-phase refrigerant is the liquid-phase refrigerant stored in the supply tank 20, which is supplied from the inflow port 21a via the buffer chamber 35h and branches into the injection flow path 24 via the main flow path 21 in the rotating shaft 25. The liquid-phase refrigerant is centrifugally pressurized in the injection flow path 24 inside the rotating body 27 which rotates at high speed and is injected from the outflow port 24b into the refrigerant flow path 40, where the liquid-phase refrigerant is sucked along with the gas-phase refrigerant sucked into the compressor 3b. When the refrigeration power is 880 kW as a rated condition, the injection quantity of liquid-phase refrigerant required to remove the heat generated in the compression process is, for example, 0.034 kg/s. For example, if the diameter of the port of the injection flow path 24 is 0.13 mm and the number of ports is 16, the liquid-phase refrigerant is injected into the refrigerant flow path 40 from the outflow port 24b at a pressure of about 1.4 MPa through the injection flow path 24. The liquid-phase refrigerant is continuously supplied from the supply tank 20 and is pumped into the buffer chamber 35h by the pressure pump 19 while being sucked by the centrifugal pressurization.

[0177] As described above, the liquid-phase refrigerant is centrifugally pressurized in the injection flow path 24 inside the rotating body 27 rotating at high speed and is injected into the refrigerant flow path 40. Accordingly, the superheated gas-phase refrigerant is continuously cooled. The liquid-phase refrigerant is pressurized by the pressure pump 19 when passing through the refrigerant supply path 22. Thus, the pressure of the liquid-phase refrigerant rises, and the boiling point rises. As a result, the liquid-phase refrigerant is less likely to evaporate inside the main flow path 21, and clogging of a flow path by vapor can be prevented. At the same time, the efficiency of the motor 16 is also improved since the motor 16 can be cooled reliably.

[0178] More specifically, when the refrigeration power is 880 kW as the rated condition, the heat value of the motor 16 is about 0.8 kW, and the injection quantity of the liquid-phase refrigerant required to remove the heat generated in the compression process is, for example, 0.034 kg/s. Assuming that the temperature of the liquid-phase refrigerant in the supply tank 20 is 35°C. (4.25 kPa), the temperature after the liquid-phase refrigerant passes through the main flow path 21 is 40.46°C. (7.57 kPa). The compressor 3b is provided at a height of, for example, 1.5m with respect to the outflow port of the supply tank 20. In consideration of the pressure loss of the main flow path 21 and the refrigerant supply path 22 of the compressor 3b, the pressure of the liquid-phase refrigerant supplied from the supply tank 20 needs to be increased by, for example, 22.3 kPa or more to prevent the liquid-phase refrigerant from evaporating in the main flow path 21. Therefore, if the supply pressure of the pressure pump 19 is set to 22.3 kPa or higher, the liquid-phase refrigerant having an increased pressure higher than or equal to the evaporating pressure is supplied. In this manner, the liquid-phase refrigerant is less likely to evaporate in the main flow path 21, and clogging of the flow path by vapor can be prevented.

[0179] Fig. 16 illustrates a cross section of a compressor 3c according to another modification. In the compressor 3c, the buffer chamber 35h is connected to the refrigerant supply path 22. The refrigerant supply path 22 enables the buffer chamber 35h and the supply tank 20 to communicate with each other. The refrigerant supply path 22 is provided with a pressure pump 19 for pumping the liquid-phase refrigerant stored in the supply tank 20 to the buffer chamber 35h and a heat exchanger 23 for exchanging heat with an external heat source.

[0180] The compressor 3c differs from the compressor 3b illustrated in Fig. 15 in that it further includes a heat exchanger 23.

[0181] The refrigerant supply path 22 is a flow path connected to the buffer chamber 35h and the pressure pump 19. The heat exchanger 23 is provided in the refrigerant supply path 22 between the buffer chamber 35h and the pressure pump 19.

[0182] The temperature of the liquid-phase refrigerant in the supply tank 20 is, for example, 35°C. The inflow temperature of the heat exchanger 23 is, for example, 35°C, and the outflow temperature is, for example, 30°C.

[0183] As described above, since the liquid-phase refrigerant is cooled by the heat exchanger 23 provided in the refrigerant supply path 22, the supercooled liquid-phase refrigerant is supplied to the main flow path 21, and the liquid-phase refrigerant is less likely to evaporate inside the main flow path 21. As a result, clogging of the flow path due to vapor can be prevented even when the rotational speed of the compressor 3c increases and, thus, the amount of exhaust heat from the motor 16 increases, in particular, under high load operating conditions.

[0184] The structure of the heat exchanger 23 is not limited to a particular structure. For example, a fin tube heat

exchanger, a plate heat exchanger, or a double tube heat exchanger may be used as the heat exchanger 23. The external heat source for cooling the liquid-phase refrigerant by heat-exchanging the liquid-phase refrigerant in the heat exchanger 23 is not limited to a particular type. For example, air or cooling water may be used as the external heat source.

(Second Embodiment)

[0185] Fig. 17 is a configuration diagram of a refrigeration cycle apparatus according to a second embodiment of the present disclosure. The same reference symbols are used for elements common to the first embodiment and other embodiments, and description of the elements is not necessarily repeated. Descriptions of the embodiments can be interchangeably used as long as no technical conflicts occur. The embodiments may be combined with one another as long as no technical conflicts occur.

[0186] As illustrated in Fig. 17, in a refrigeration cycle apparatus 102 according to the second embodiment, a refrigerant supply path 11 connects a condenser 4 to a compressor 3. In the compressor 3, a liquid-phase refrigerant injected into a refrigerant flow path 40 through a main flow path 21 and an injection flow path 24 is a liquid-phase refrigerant stored in the condenser 4. In addition, even in the present modification, the effect of reducing the compression power can be obtained by the mechanism described in the first embodiment. That is, the liquid-phase refrigerant to be supplied to the main flow path 21 inside the compressor 3 is not limited to the liquid-phase refrigerant stored in the evaporator 2. As long as a refrigerant is present in the refrigerant circuit 10, the liquid-phase refrigerant can be supplied to the main flow path 21. For example, if there is a buffer tank that is connected to the evaporator 2 or the condenser 4 and that stores the liquid-phase refrigerant, the refrigerant supply path 11 may connect the buffer tank to the compressor 3 such that the liquid-phase refrigerant is supplied from the buffer tank to the main flow path 21. In addition, the refrigerant supply path 11 may branch off from the return path 9. In other words, the return path 9 may also serve as a part of the refrigerant supply path 11. In this case, the refrigerant supply path 11 leads the liquid-phase refrigerant from the condenser 4 to the main flow path 21.

[0187] According to the present modification, a liquid-phase refrigerant having a temperature higher than the temperature (the saturation temperature) of the gas-phase refrigerant sucked into the compressor 3 is sucked into the compressor 3. In this case, the mechanism described in the first embodiment provides the effect of reducing the compression power while preventing the gas-phase refrigerant from being too cooled and condensing inside the compressor 3.

[0188] The refrigeration cycle apparatus 102 may include a spare tank that stores a liquid-phase refrigerant. The spare tank is connected to, for example, the condenser 4. The liquid-phase refrigerant is transferred from the condenser 4 to the spare tank. The refrigerant supply path 11 connects the spare tank and the compressor 3 to each other so that the liquid-phase refrigerant is supplied from the spare tank to the compressor 3.

[0189] Instead of the compressor 3, other compressors 3a, 3b, 3c, 50, 60, 70, and 90 described above can be used.

(Third Embodiment)

[0190] Fig. 18 is a configuration diagram of a refrigeration cycle apparatus according to a third embodiment of the present disclosure. As illustrated in Fig. 18, a refrigeration cycle apparatus 104 includes an ejector 53, a buffer tank 52, and a heat exchanger 23 as an alternative to the condenser 4.

[0191] The operation and action performed by the refrigeration cycle apparatus 104 having the above-described configuration are described below.

[0192] The gas-phase refrigerant compressed and discharged by the compressor 3 is sucked into the ejector 53. In addition, a liquid-phase refrigerant is stored in the buffer tank 52, and the liquid-phase refrigerant in the buffer tank 52 dissipates heat in the heat exchanger 23 and is supplied to the ejector 53. In the ejector 53, the gas-phase refrigerant received from the compressor 3 and the liquid-phase refrigerant received from the heat exchanger 23 are mixed with each other. The refrigerant is compressed in a two-phase state and is supplied to the buffer tank 52 in the form of a high-temperature liquid-phase refrigerant or a gas-liquid two-phase refrigerant. That is, the gas-phase refrigerant is pressurized in the ejector 53 in a two-phase state and, thus, the gas-phase refrigerant is condensed. The liquid-phase refrigerant dissipates heat in the heat exchanger 23. In this manner, the ejector 53, the buffer tank 52, and the heat exchanger 23 function as a substitute for the condenser 4. The temperature of the liquid-phase refrigerant in the buffer tank 52 is, for example, 38.5°C. The inflow temperature of the heat exchanger 23 is, for example, 38.5°C, and the outflow temperature is, for example, 33.5°C.

[0193] The liquid-phase refrigerant in the buffer tank 52 is pumped to the heat exchanger 23 by the pressure pump 19. The flow path of the liquid-phase refrigerant on the discharge side of the pressure pump 19 branches into two paths. One of the paths communicates with the heat exchanger 23, and the other communicates with the buffer chamber 35h of the compressor 3. In other words, the flow path that enables a branch point of the flow path of the liquid-phase refrigerant on the discharge side of the pressure pump 19 to communicate with the buffer chamber 35h is the refrigerant supply path 22. The supply pressure of the pressure pump 19 is, for example, about 250 kPa.

[0194] As described above, the liquid-phase refrigerant is centrifugally pressurized in the injection flow path 24 inside the rotating body 27 rotating at high speed and is injected into the refrigerant flow path 40. Accordingly, the superheated gas-phase refrigerant is continuously cooled. The liquid-phase refrigerant is pressurized by the pressure pump 19 when passing through the refrigerant supply path 22, and the pressure of the liquid-phase refrigerant rises. Thus, the boiling point of the liquid-phase refrigerant rises. For this reason, the liquid-phase refrigerant is less likely to evaporate inside the main flow path 21, and clogging of the flow path by vapor can be prevented.

(Fourth Embodiment)

[0195] Fig. 19 is a configuration diagram of a refrigeration cycle apparatus according to a fourth embodiment of the present disclosure. As illustrated in Fig. 19, a refrigeration cycle apparatus 106 includes an ejector 53, a buffer tank 52, and a heat exchanger 23 as an alternative to the condenser 4.

[0196] The operation and action performed by the refrigeration cycle apparatus 106 having the above-described configuration are described below.

[0197] The liquid-phase refrigerant in the buffer tank 52 is pumped to the heat exchanger 23 by the pressure pump 19, dissipates heat in the heat exchanger 23, and is supplied to the ejector 53. The flow path of the liquid-phase refrigerant on the outflow side of the heat exchanger 23 branches into two paths. One of the paths communicates with the ejector 53, and the other communicates with the buffer chamber 35h of the compressor 3. In other words, the flow path that enables a branch point of the flow path of the liquid-phase refrigerant on the outflow side of the heat exchanger 23 to communicate with the buffer chamber 35h is the refrigerant supply path 22. The temperature of the liquid-phase refrigerant in the buffer tank 52 is, for example, 38.5°C. The inflow temperature of the heat exchanger 23 is, for example, 38.5°C, and the outflow temperature is, for example, 33.5°C.

[0198] As described above, since the liquid-phase refrigerant is cooled by the heat exchanger 23 provided in the refrigerant supply path 22, the supercooled liquid-phase refrigerant is supplied to the main flow path 21 and, thus, the liquid-phase refrigerant is less likely to evaporate inside the main flow path 21. As a result, clogging of the flow path due to vapor can be prevented even when the rotational speed of the compressor 3 increases and, thus, the amount of exhaust heat from the motor 16 increases, in particular, under high load operating conditions.

Industrial Applicability

[0199] The refrigeration cycle apparatus disclosed in this specification is useful for air conditioners, chillers, heat storage devices, and the like. Air conditioners are used, for example, for central air conditioning of a building. Chillers are used, for example, in process cooling applications.

Reference Signs List

[0200]

2	evaporator
3, 3a, 3b, 3c, 50, 60, 70, 90	compressor
4	condenser
6	suction pipe
8	discharge pipe
9	return path
10	refrigerant circuit
11, 22	refrigerant supply path
12	heat absorption circuit
14	heat radiation circuit
16	motor
18	bearing
19	pressure pump
20	supply tank
21	main flow path
21a	inflow port
23	heat exchanger
24	injection flow path (first injection flow path)
24b, 32b, 74b	outflow port
25, 45	rotating shaft

	25c	end face
	26	impeller (first impeller)
	26t	upper surface
	27, 47, 77	rotating body
5	28	connection port
	29	seal
	30, 33	hub
	30p	surface of hub
	31	blade
10	31t	upstream end of blade
	32	downstream injection flow path
	35	housing
	35h	buffer chamber
	36, 76	suction flow path
15	37	shroud
	38, 78	inter-blade flow path
	40, 80	refrigerant flow path
	41	diffuser (first diffuser)
	42	volute chamber
20	51	second diffuser
	52	buffer tank
	53	ejector
	71	second impeller
	74	second injection flow path
25	79	return channel
	100, 102, 104, 106	refrigeration cycle apparatus
	241	first portion
	241a	radial portion
	241b	groove
30	242	second portion
	311	first blade
	312	second blade

35 Claims

1. A dynamic compressor comprising:

40 a rotating body including a rotating shaft and at least one impeller;
a refrigerant flow path located around the rotating body, the refrigerant flow path enabling a gas-phase refrigerant to flow therethrough;
a main flow path extending in the axial direction of the rotating body inside the rotating body, the main flow path enabling a liquid-phase refrigerant to flow therethrough; and
45 an injection flow path located inside the rotating body, the injection flow path branching off from the main flow path and extending from the main flow path to the refrigerant flow path so as to lead a liquid-phase refrigerant from the main flow path to the refrigerant flow path.

2. The dynamic compressor according to Claim 1, wherein the impeller has a hub and a blade fixed to the hub, and the injection flow path has an outflow port facing the refrigerant flow path, and
50 wherein the outflow port is located upstream of an upstream end of the blade in a flow direction of the gas-phase refrigerant.

3. The dynamic compressor according to Claim 1 or 2, wherein the impeller has a hub and a blade fixed to the hub, and the injection flow path has an outflow port located on a surface of the hub, and
55 wherein the injection flow path penetrates the hub in a radial direction of the rotating shaft.

4. The dynamic compressor according to any one of Claims 1 to 3, wherein the injection flow path includes a first portion extending from the main flow path in the radial direction of the rotating shaft inside the rotating shaft and a

second portion located between the first portion and the refrigerant flow path.

5. The dynamic compressor according to Claim 4, wherein the number of the injection flow paths each including the first portion and the second portion is greater than or equal to two.

6. The dynamic compressor according to Claim 4 or 5, wherein the first portion includes a groove provided on a side surface of the rotating shaft and extending in a circumferential direction of the rotating shaft, and wherein the second portion is connected to the groove.

7. The dynamic compressor according to any one of Claims 1 to 6, wherein the main flow path has an inflow port located on an end face of the rotating shaft.

8. The dynamic compressor according to any one of Claims 1 to 7, further comprising:

a supply tank storing the liquid-phase refrigerant;
a buffer chamber in contact with the inflow port of the main flow path; and
a pressure pump that pumps the liquid-phase refrigerant from the supply tank to the buffer chamber via a refrigerant supply path connected to the buffer chamber.

9. The dynamic compressor according to Claim 8, further comprising:

a heat exchanger that exchanges heat with an external heat source,
wherein the refrigerant supply path is a flow path connected to the buffer chamber and the pressure pump, and
wherein the heat exchanger is provided in the refrigerant supply path between the buffer chamber and the pressure pump.

10. The dynamic compressor according to any one of Claims 1 to 9, wherein the impeller includes a hub and a plurality of blades fixed to the hub,

wherein the injection flow path has an outflow port facing the refrigerant flow path,

wherein one of the blades located closest to the outflow port in a rotational direction opposite to a rotational direction of the rotating body is defined as a first blade,

wherein in a projection view obtained by projecting a blade root line of the first blade onto a plane perpendicular to the rotating shaft, an outermost peripheral portion of the blade root line is defined as a first trailing edge portion,

wherein a line extending from a central axis of the rotating body in the radial direction through the outflow port is defined as an r-axis,

wherein the rotational direction of the rotating body is defined as a positive direction,

wherein an angle formed by a line extending between the first trailing edge portion and the central axis and the r-axis as measured from the r-axis in the rotational direction of the rotating body is greater than or equal to -40 degrees,

wherein a ratio of a distance between the central axis of the rotating body and the first trailing edge portion to a distance between the central axis of the rotating body and the outflow port is greater than or equal to three, and

wherein in a projection view obtained by projecting an outflow direction of the liquid-phase refrigerant injected from the outflow port onto a plane perpendicular to the rotating shaft, an angle formed by the outflow direction of the liquid-phase refrigerant and the r-axis as measured from the r-axis in the rotational direction of the rotating body is greater than or equal to -25 degrees.

11. The dynamic compressor according to any one of Claims 1 to 10, wherein the at least one impeller comprises a first impeller and a second impeller,

wherein each of the first impeller and the second impeller is provided with the injection flow path,

wherein an opening area of the outflow port of the injection flow path provided in the first impeller is denoted by S_1 ,

wherein an opening area of the outflow port of the injection flow path provided in the second impeller is denoted by S_2 ,

wherein a distance between the central axis of the rotating body and the outflow port provided in the first impeller is denoted by R_1 ,

wherein a distance between the central axis of the rotating body and the outflow port provided in the second impeller is denoted by R_2 , and

wherein a relationship $(R_2/R_1 \leq S_1/S_2)$ is satisfied.

12. The dynamic compressor according to any one of Claims 1 to 11, wherein the at least one impeller comprises the first impeller and the second impeller,

wherein the dynamic compressor further comprises a first diffuser facing the first impeller,
 wherein the first impeller is provided with a downstream injection flow path that is located inside the first impeller
 and that branches off from the main flow path and extends from the main flow path to the refrigerant flow path,
 wherein the downstream injection flow path is located downstream of the injection flow path in the flow direction of
 the gas-phase refrigerant, and
 wherein a central axis of the downstream injection flow path intersects with an inlet of the first diffuser.

13. The dynamic compressor according to Claim 12, further comprising:

a second diffuser facing the second impeller,
 wherein the second impeller is provided with a second injection flow path that is located inside the second
 impeller and that branches off from the main flow path and extends from the main flow path to the refrigerant
 flow path, and
 wherein a central axis of the second injection flow path intersects with an inlet of the second diffuser.

14. A refrigeration cycle apparatus comprising:

an evaporator;
 the dynamic compressor according to any one of Claims 1 to 13; and
 a condenser.

15. The refrigeration cycle apparatus according to Claim 14, wherein the evaporator has a liquid-phase refrigerant stored
 therein, and the condenser has a liquid-phase refrigerant stored therein,
 wherein the refrigeration cycle apparatus further comprises a refrigerant supply path that leads one of the liquid-
 phase refrigerant stored in the evaporator and the liquid-phase refrigerant stored in the condenser to the dynamic
 compressor.

16. A compression method for use of a dynamic compressor, the dynamic compressor having a rotating body including
 a rotating shaft and an impeller and a refrigerant flow path that is located around the rotating body and that enables
 a gas-phase refrigerant to flow therethrough from a suction port for the gas-phase refrigerant to a discharge port
 for the gas-phase refrigerant, the method comprising:

causing the dynamic compressor to suck the gas-phase refrigerant;
 accelerating and compressing the sucked gas-phase refrigerant in the dynamic compressor; and
 injecting, through a flow path that communicates with an outflow port disposed in a surface of the rotating body
 and that is located inside the rotating body, a liquid-phase refrigerant from the outflow port toward the gas-
 phase refrigerant present in the refrigerant flow path.

17. The compression method according to Claim 16, wherein the flow path located inside the rotating body includes a
 main flow path that extends in an axial direction of the rotating body inside the rotating body and that enables the
 liquid-phase refrigerant to flow therethrough and an injection flow path that is located inside the rotating body and
 that branches off from the main flow path and extends from the main flow path to the refrigerant flow path so as to
 lead the liquid-phase refrigerant from the main flow path to the refrigerant flow path, and
 wherein the liquid-phase refrigerant flowing in the main flow path flows in a direction opposite to a direction in which
 the gas-phase refrigerant is sucked and flows.

18. The compression method according to Claim 16 or 17, wherein the liquid-phase refrigerant is injected from the
 outflow port by centrifugal force generated by rotation of the rotating body, and
 wherein the injected liquid-phase refrigerant is sucked into an inter-blade flow path of the dynamic compressor.

19. The compression method according to any one of Claims 16 to 18, wherein the impeller has a hub and a blade fixed
 to the hub, and
 wherein the outflow port is located upstream of an upstream end of the blade in the flow direction of the gas-phase
 refrigerant.

FIG. 1

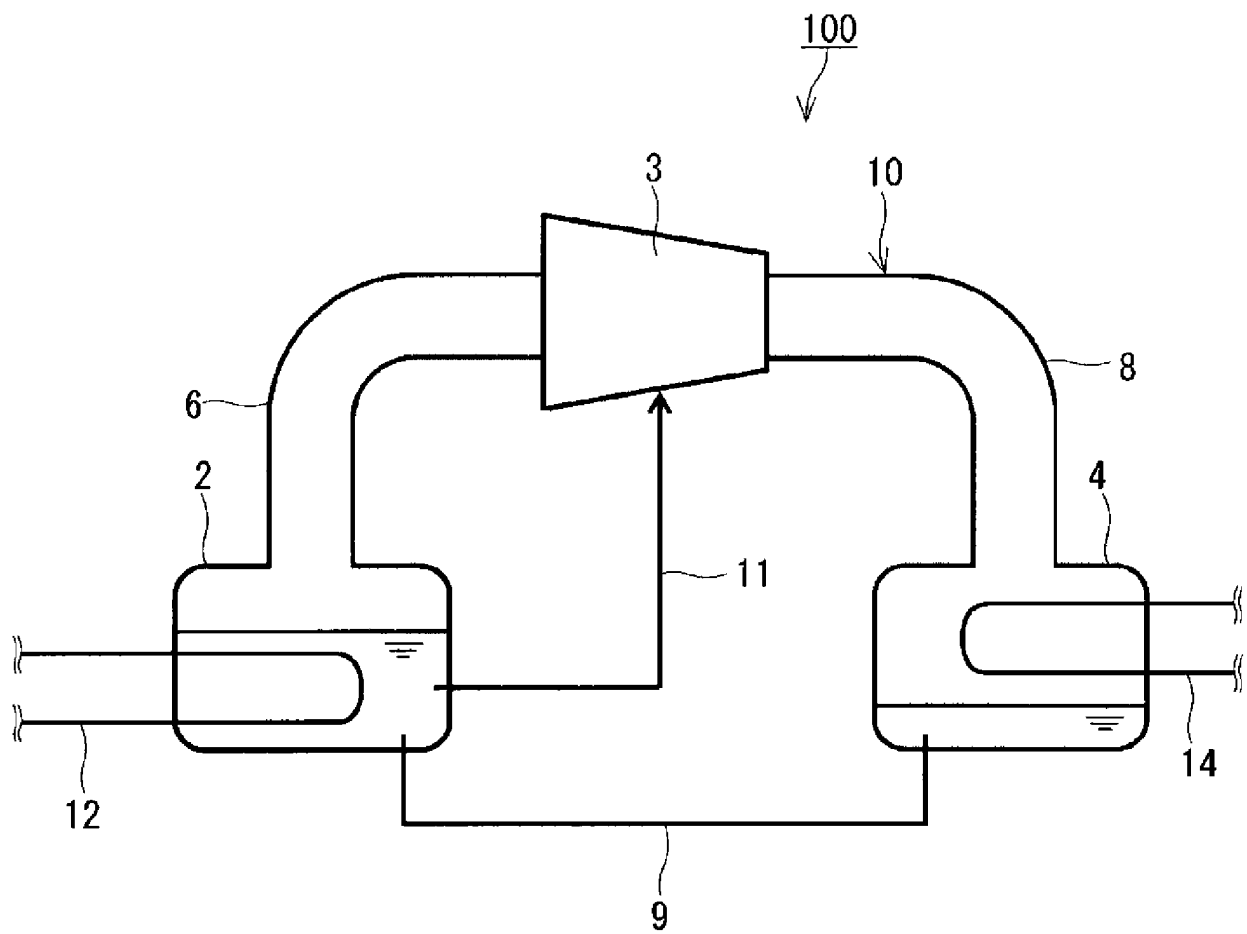


FIG. 2

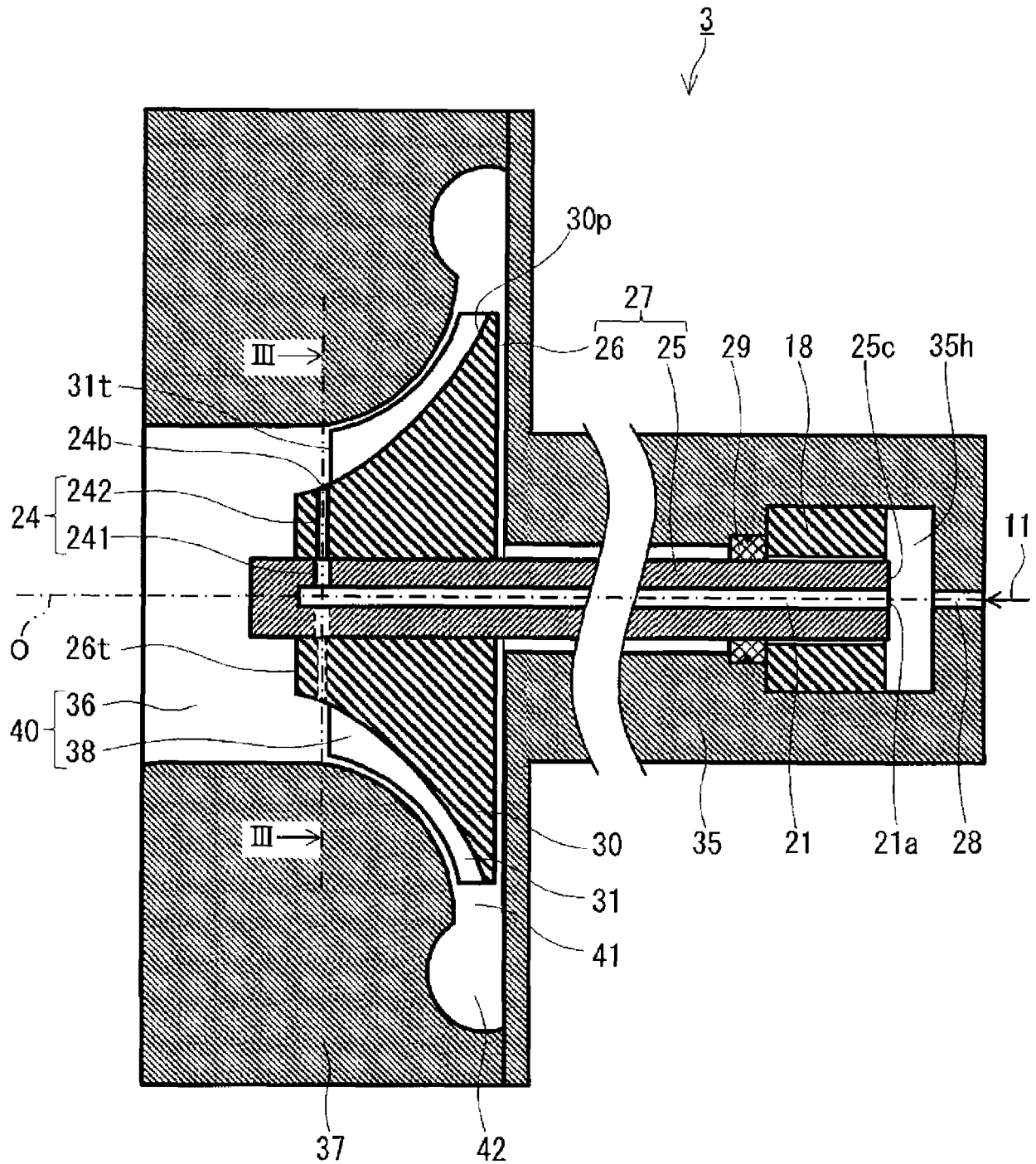


FIG. 3

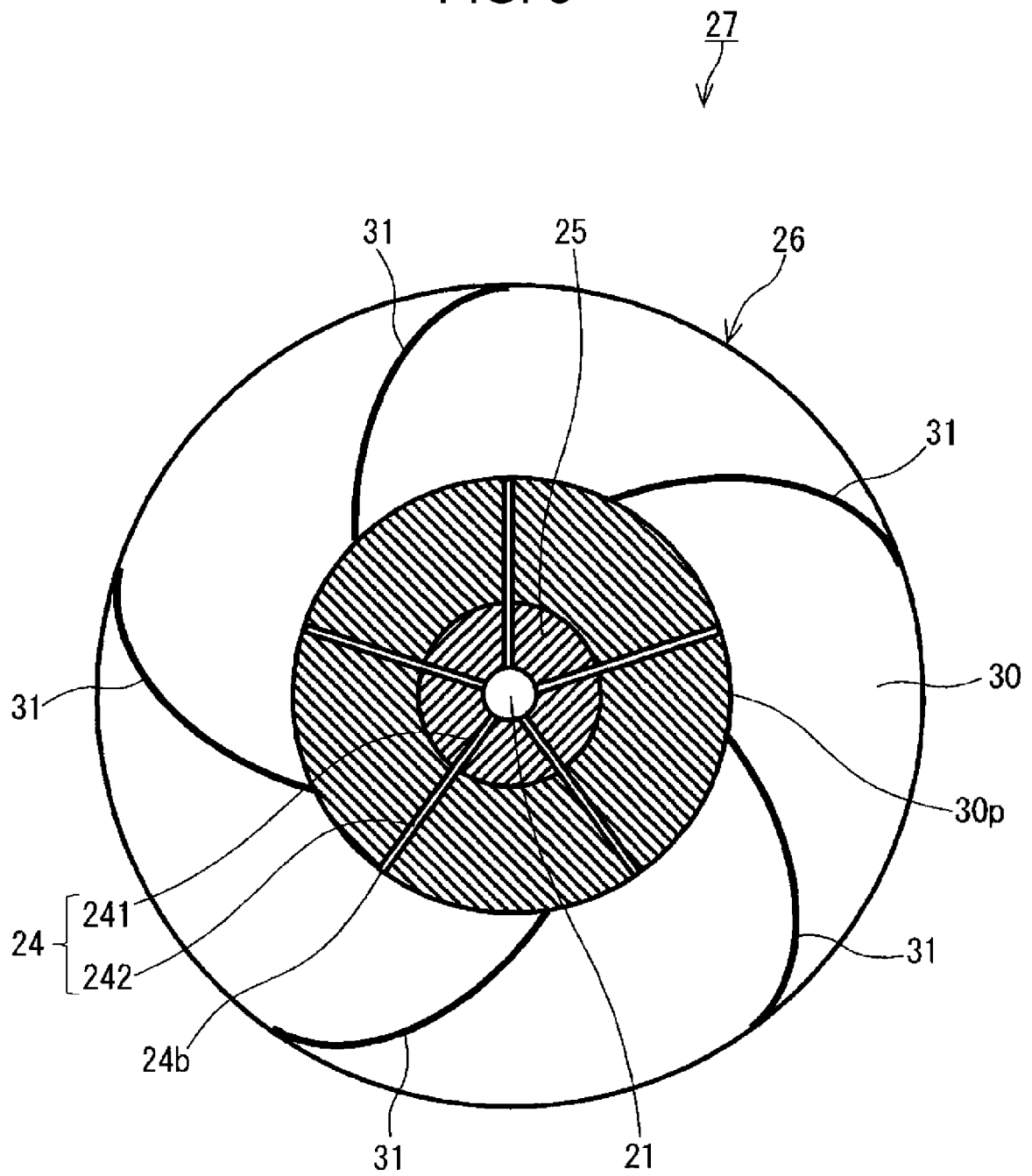


FIG. 4A

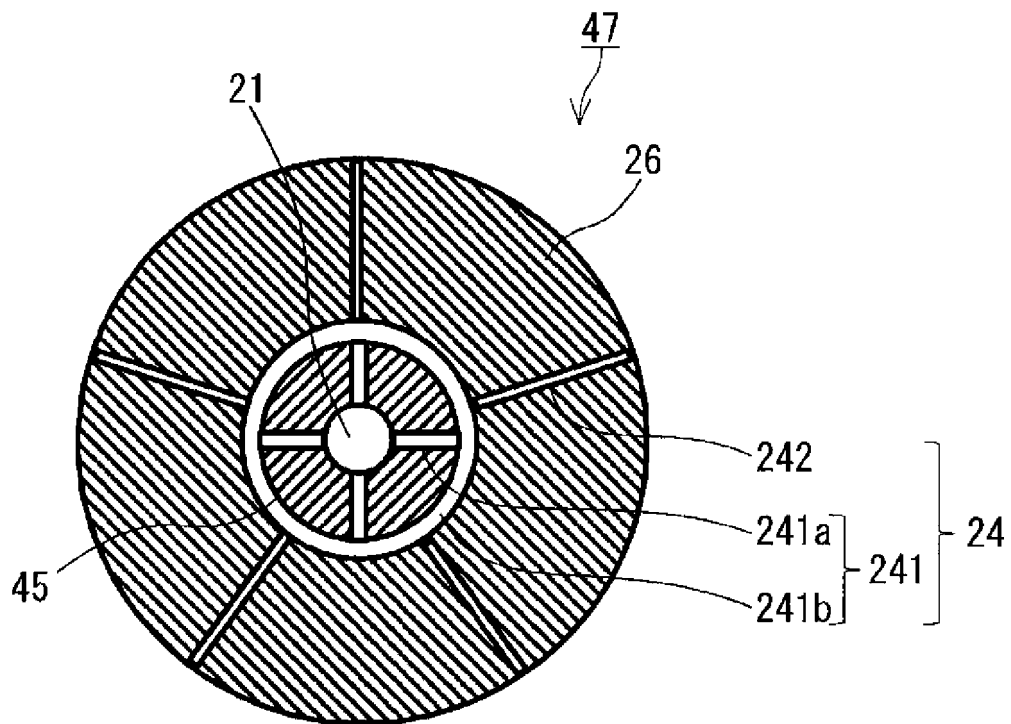


FIG. 4B

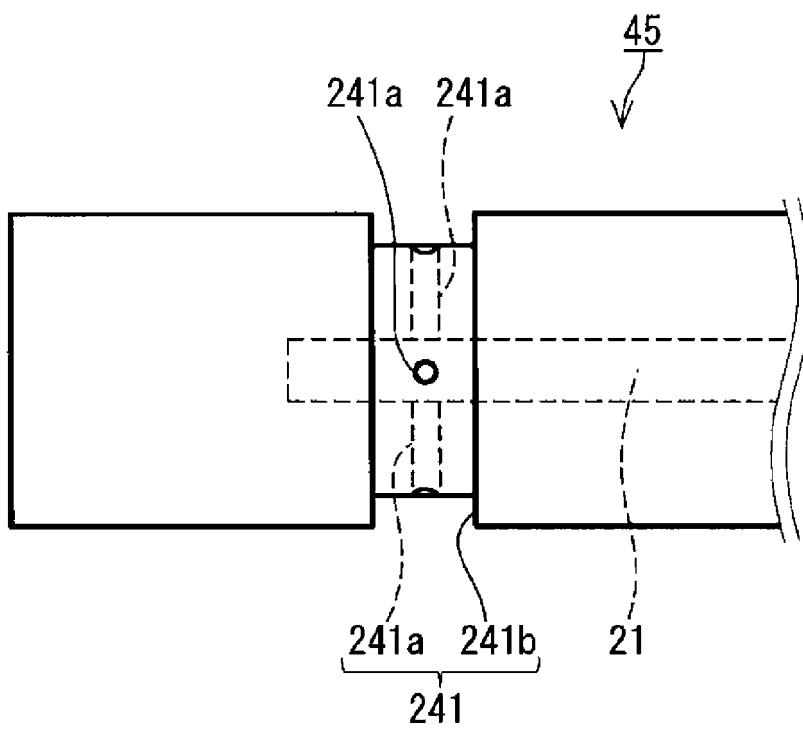


FIG. 5

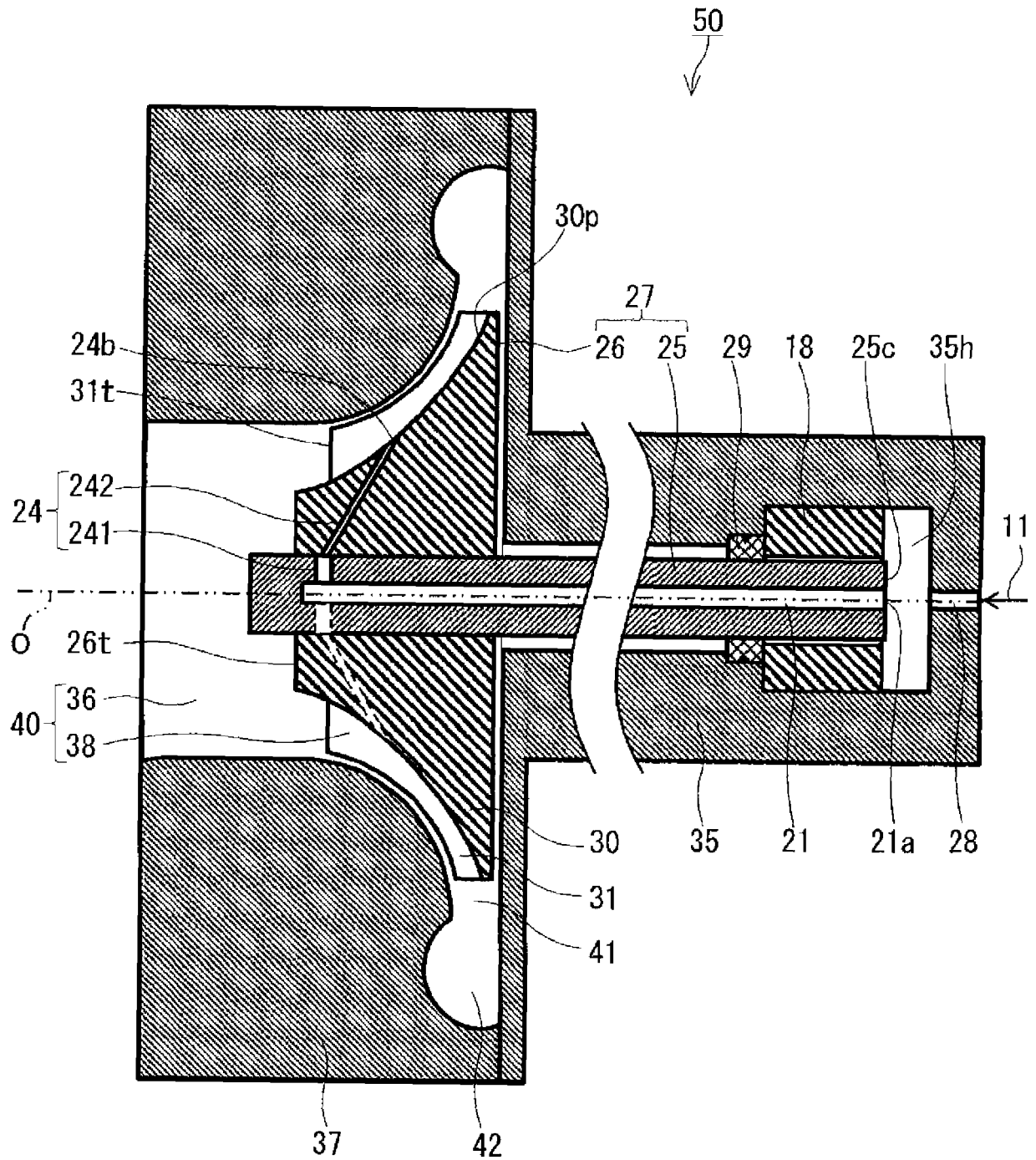


FIG. 6

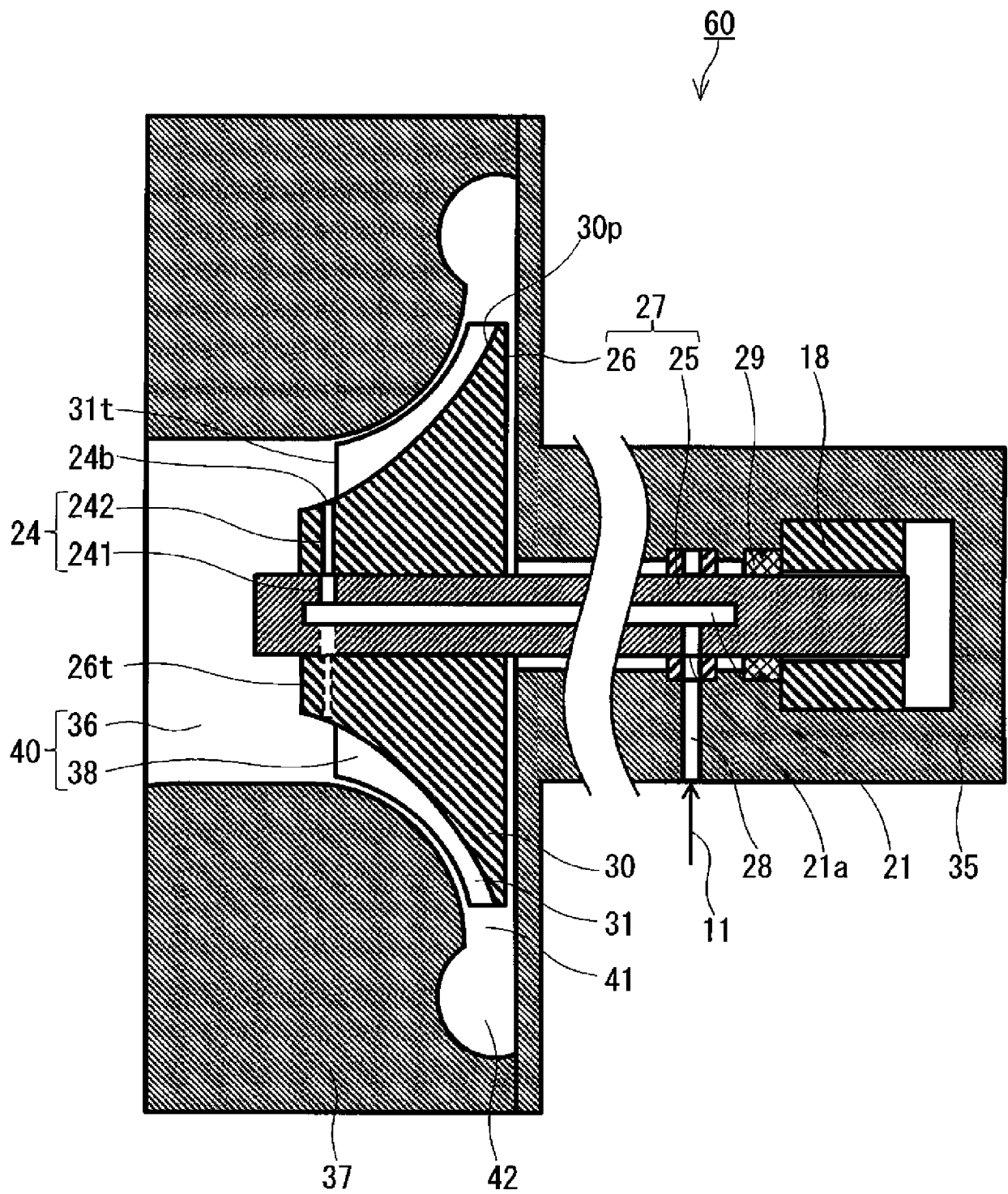


FIG. 7

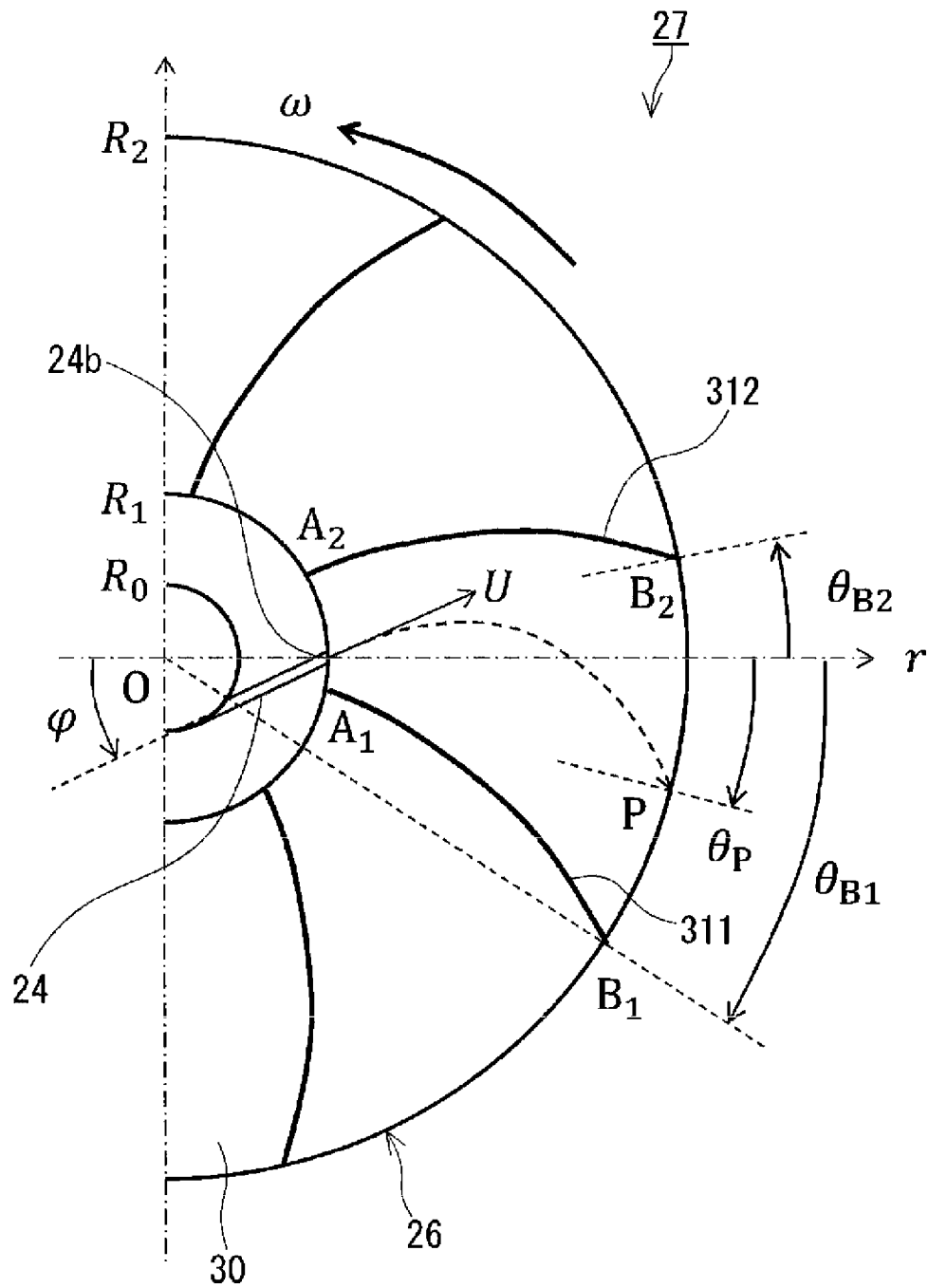


FIG. 8

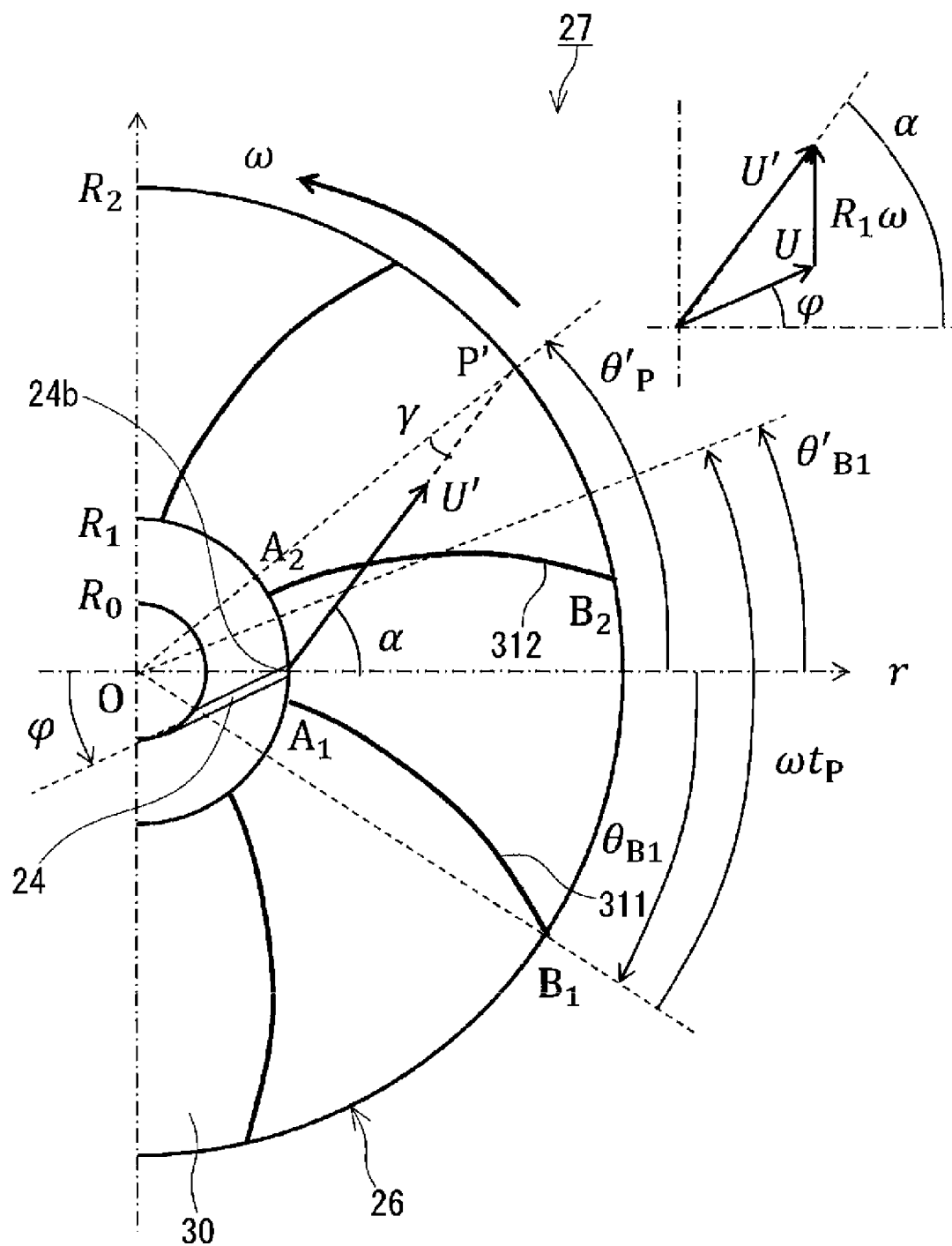


FIG. 9

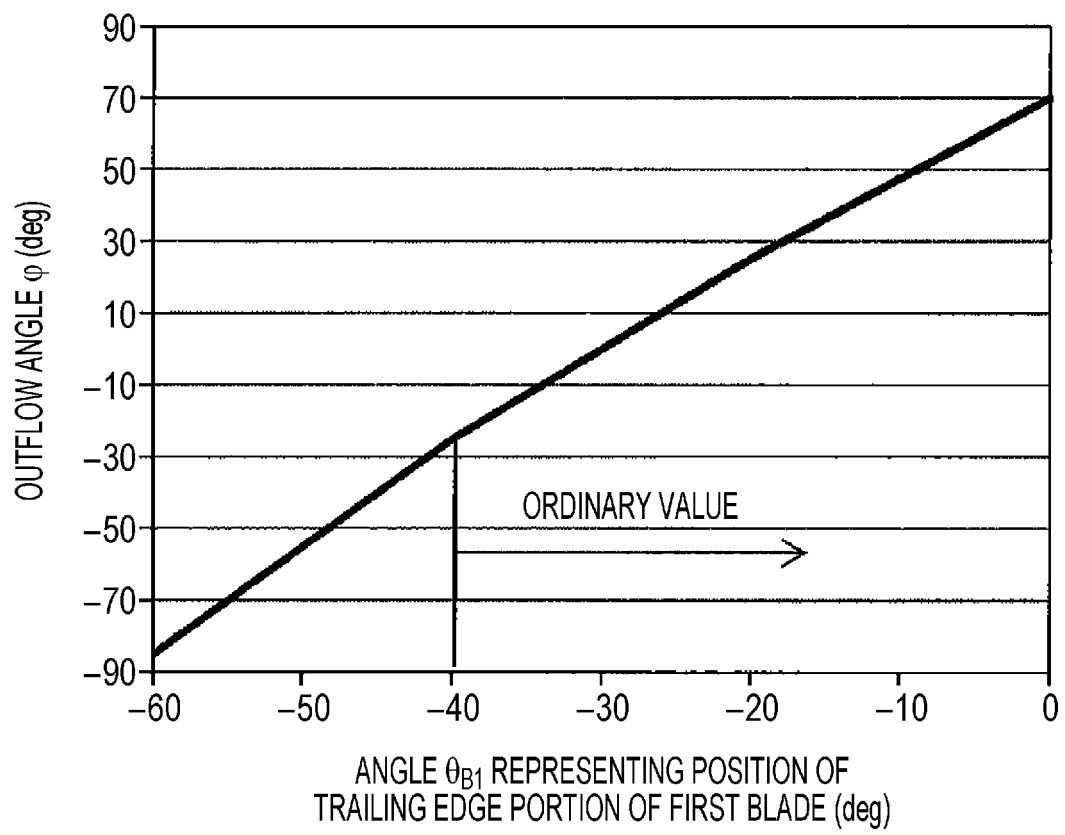


FIG. 10

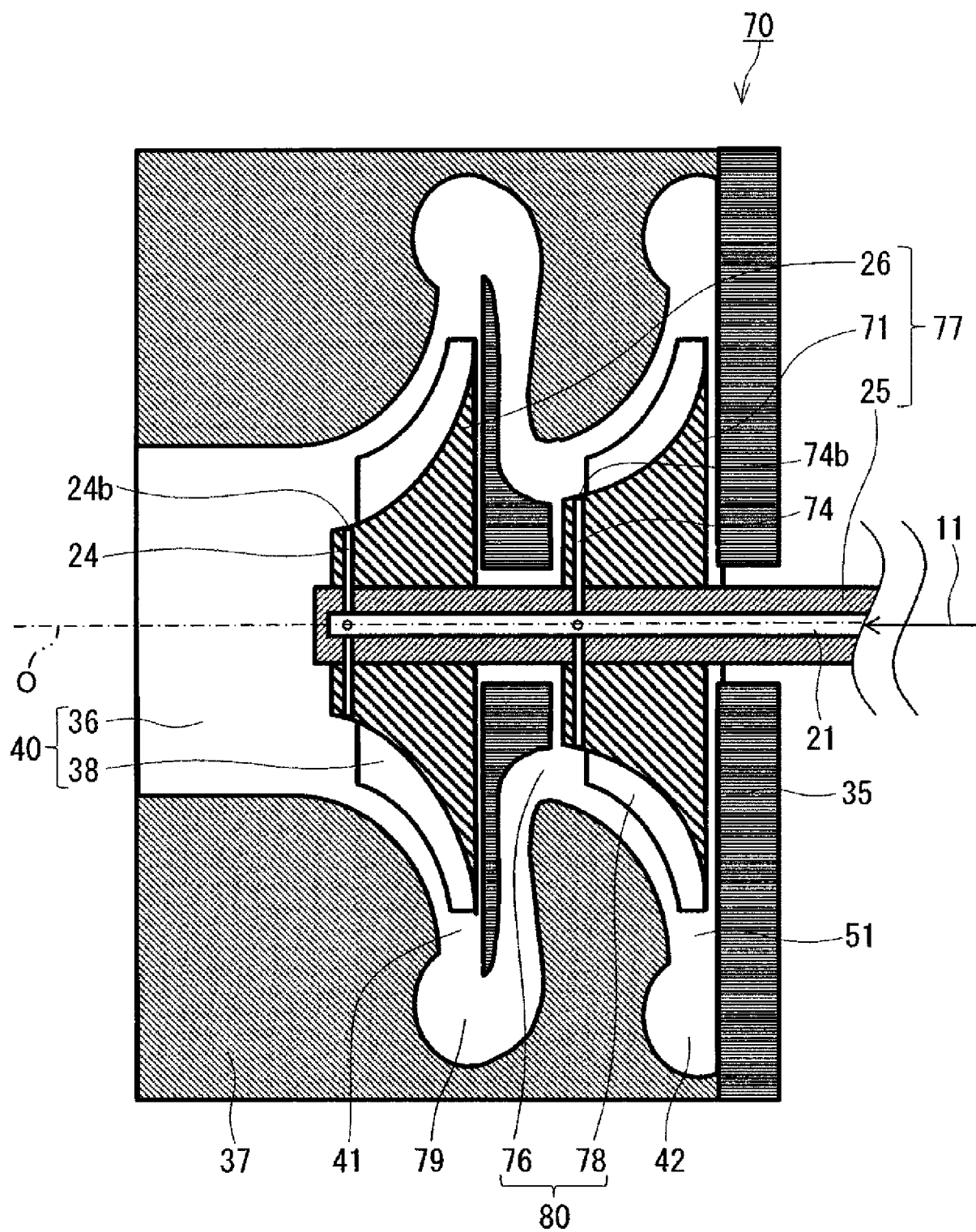


FIG. 11

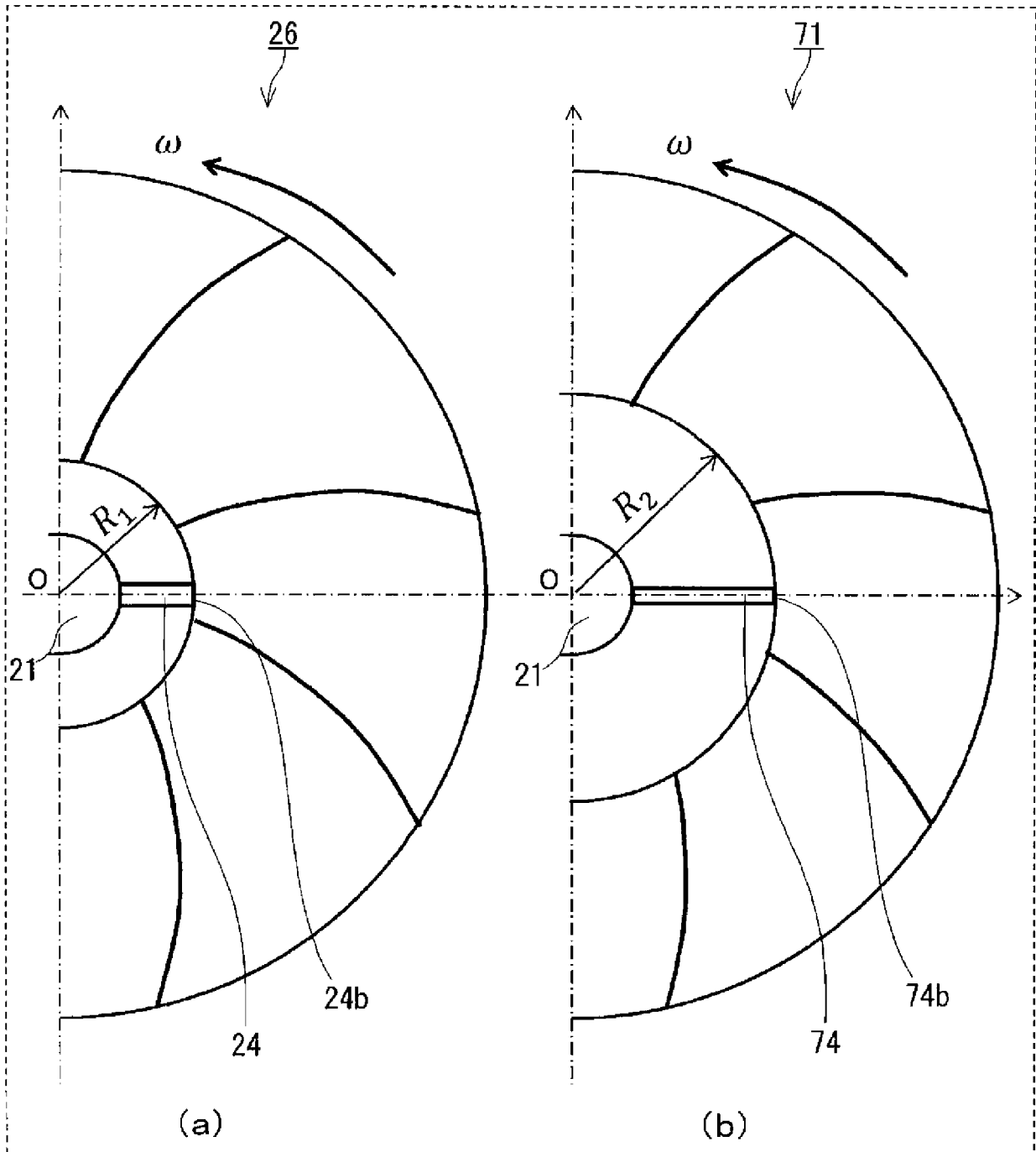


FIG. 12

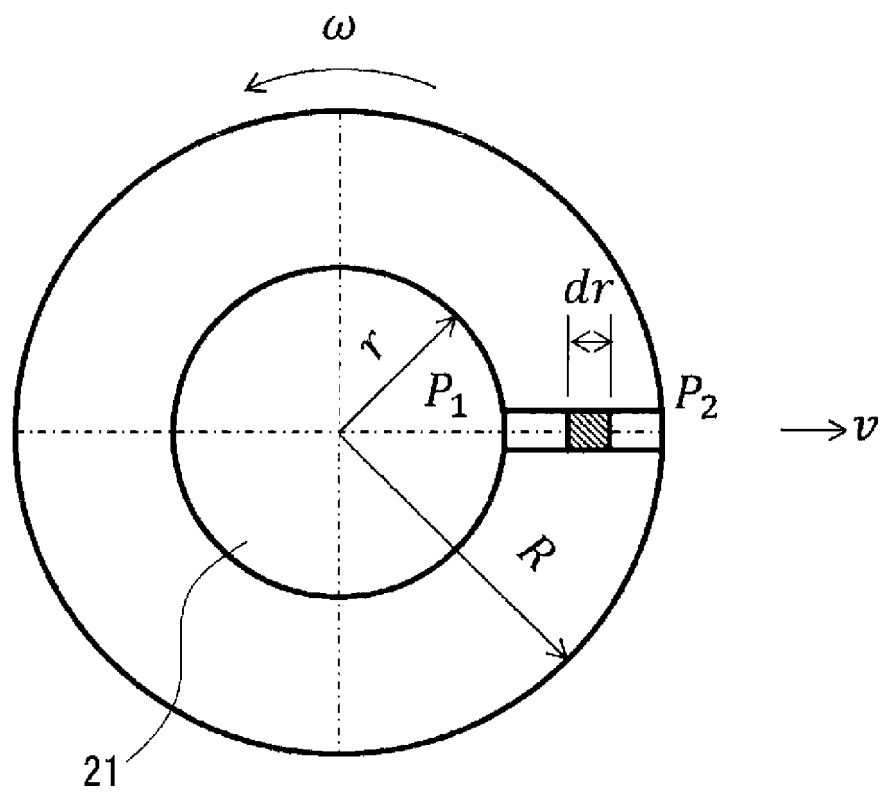


FIG. 13

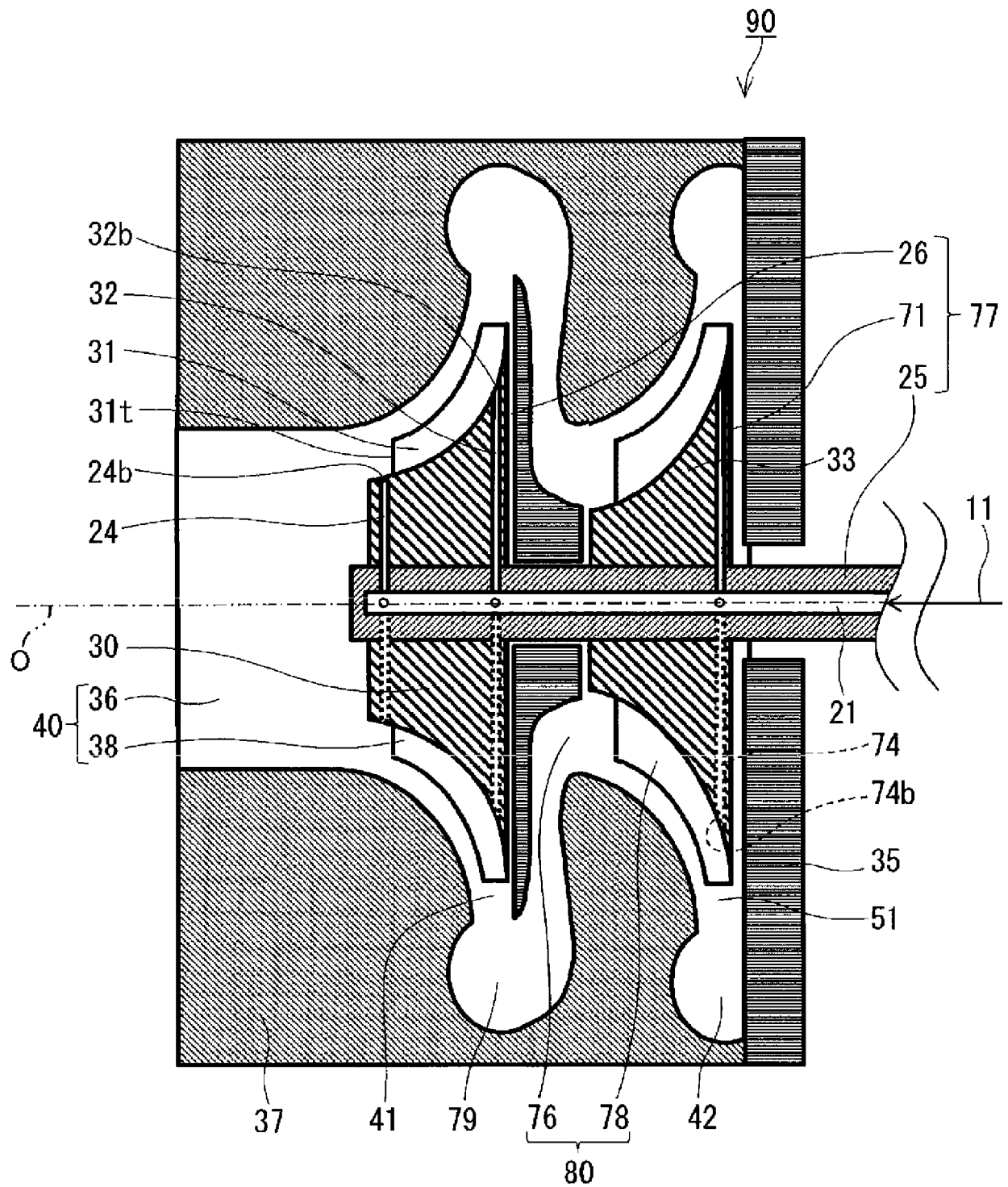


FIG. 14

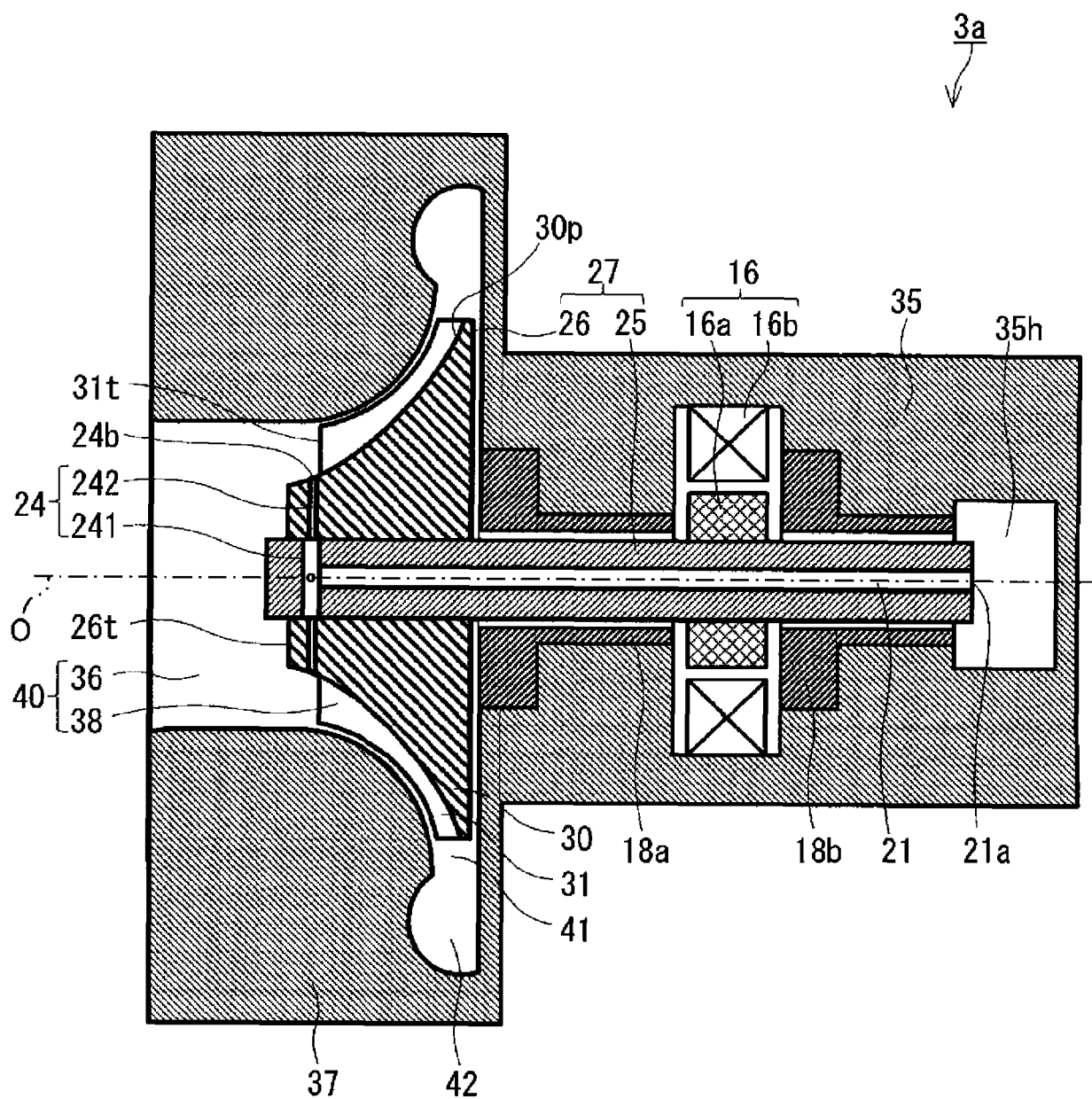


FIG. 15

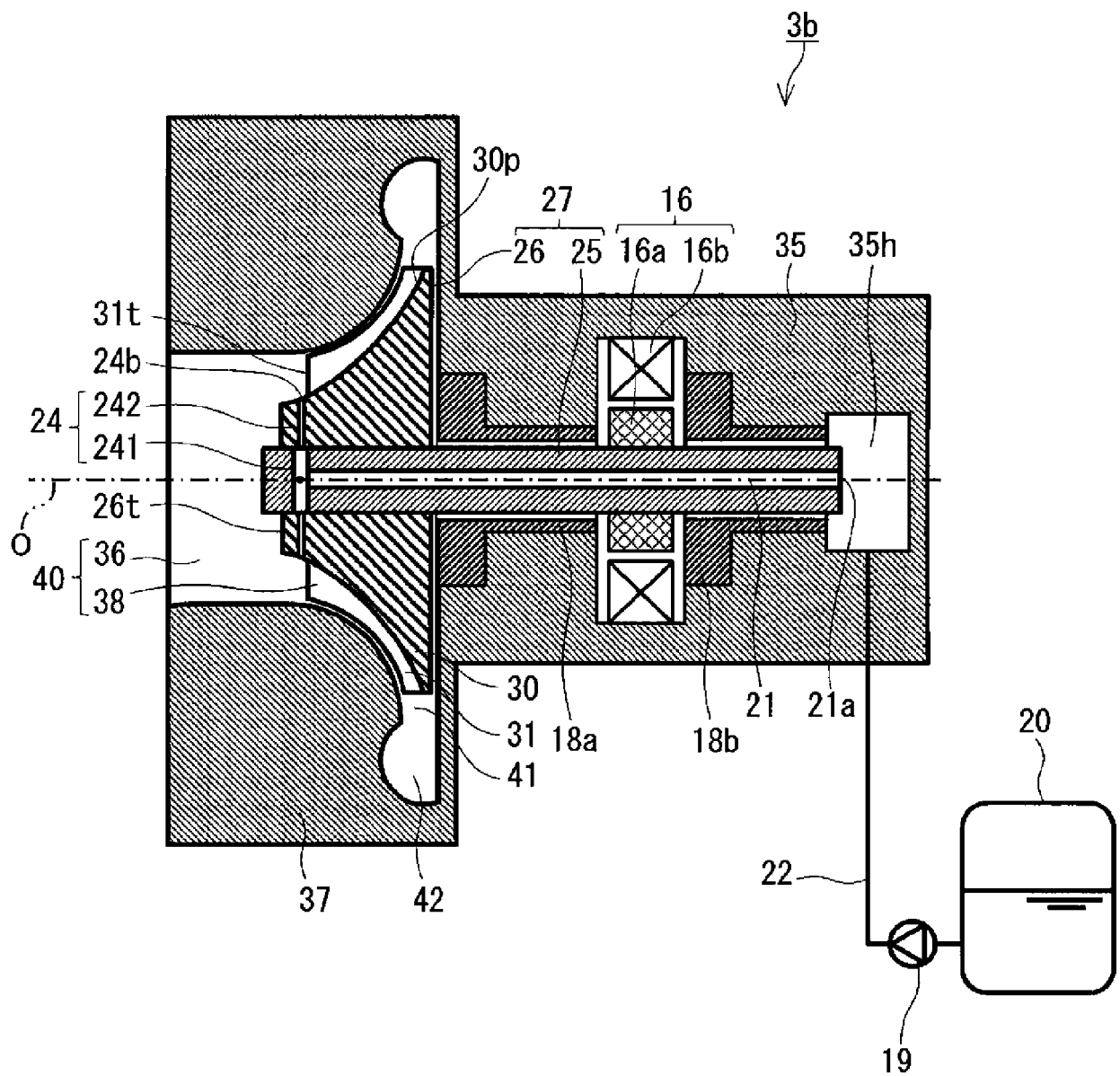


FIG. 16

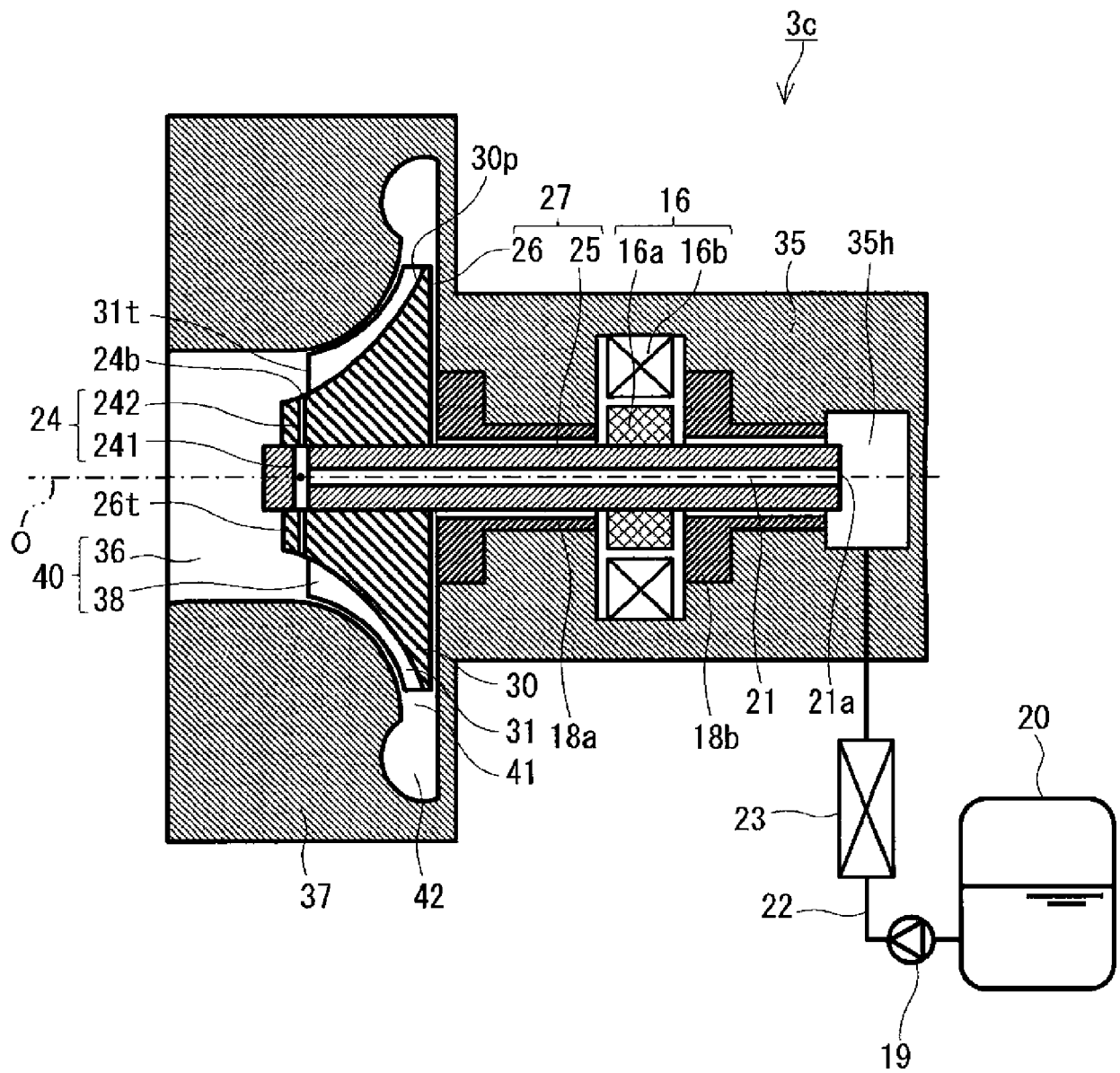


FIG. 17

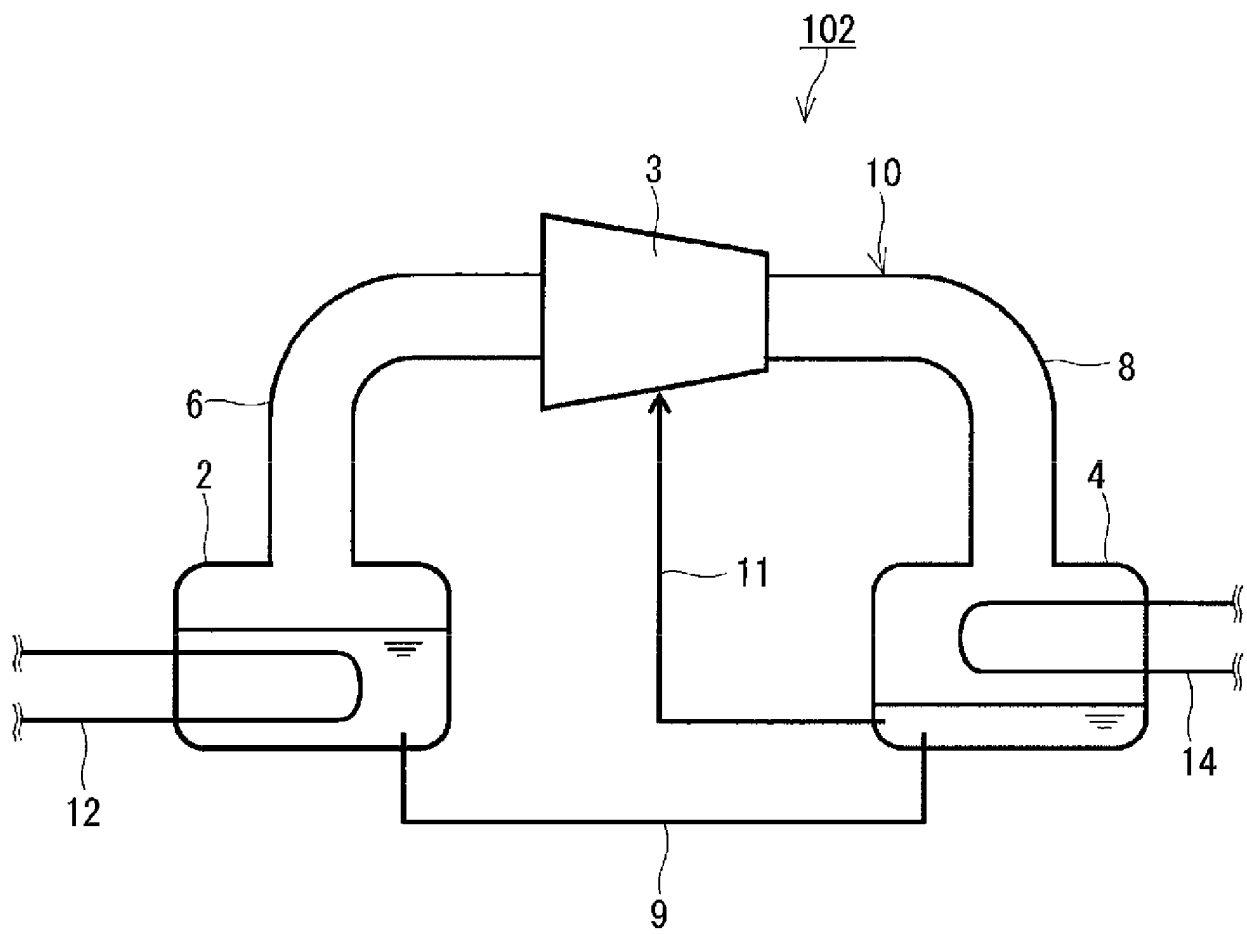


FIG. 18

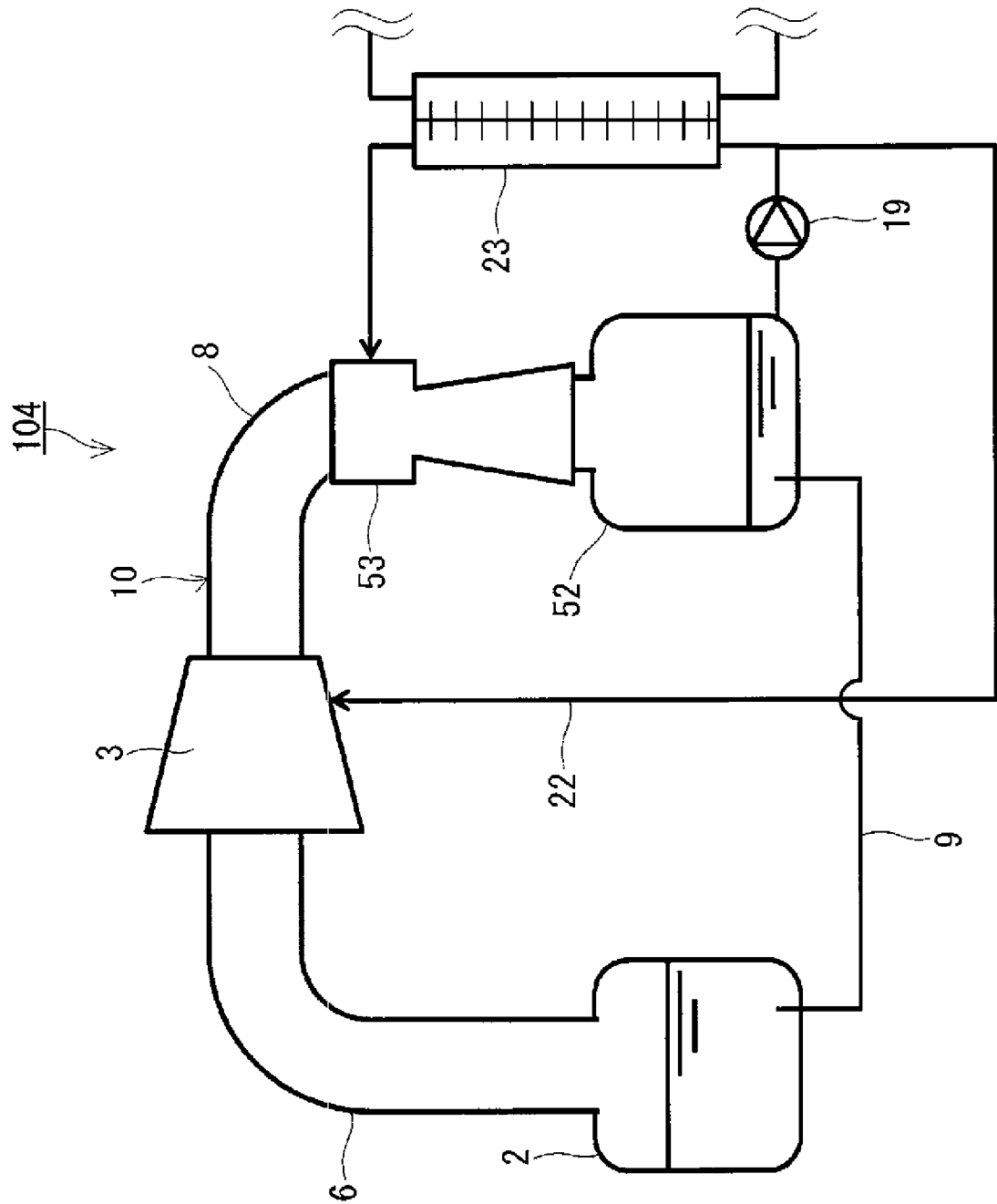


FIG. 19

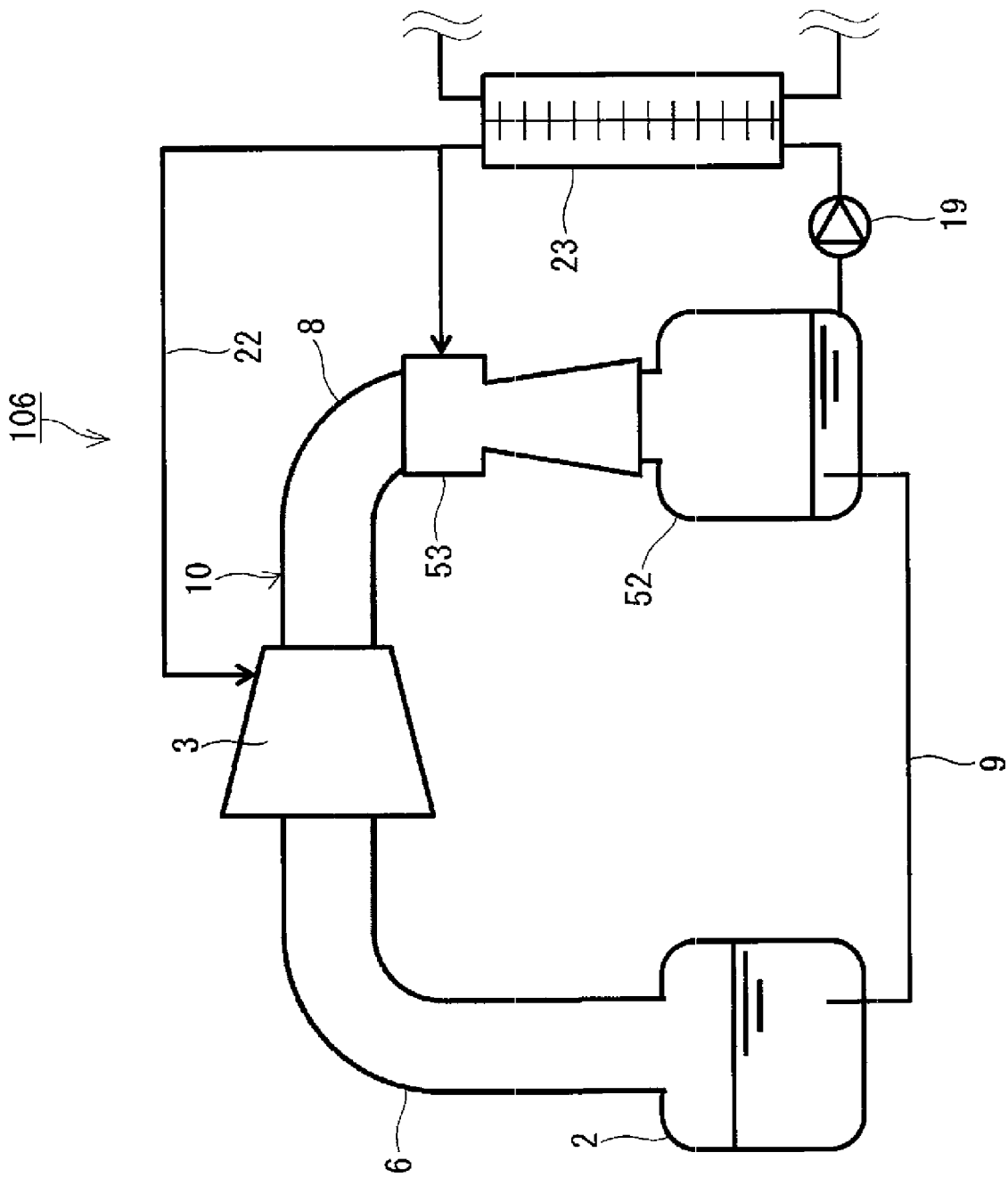


FIG. 20

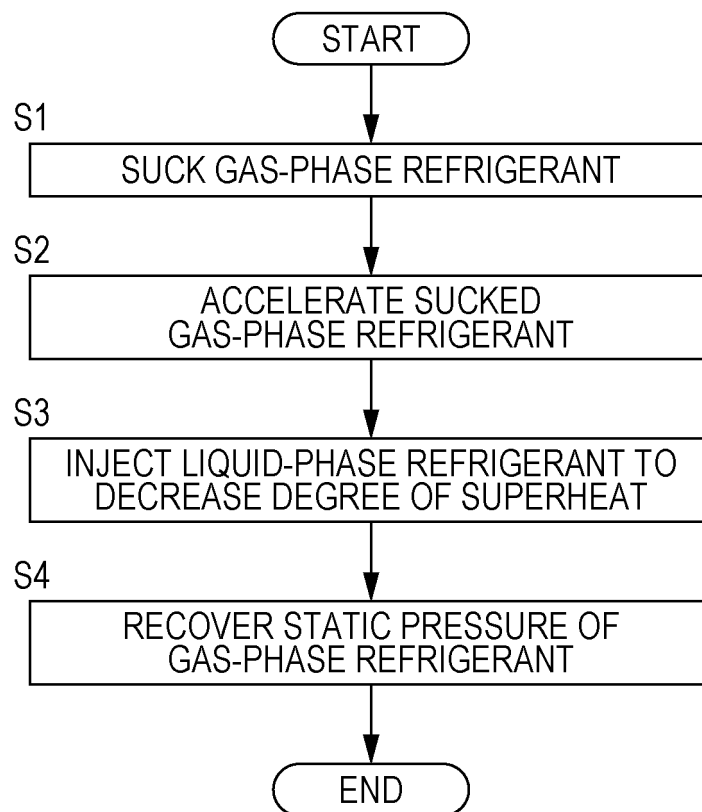
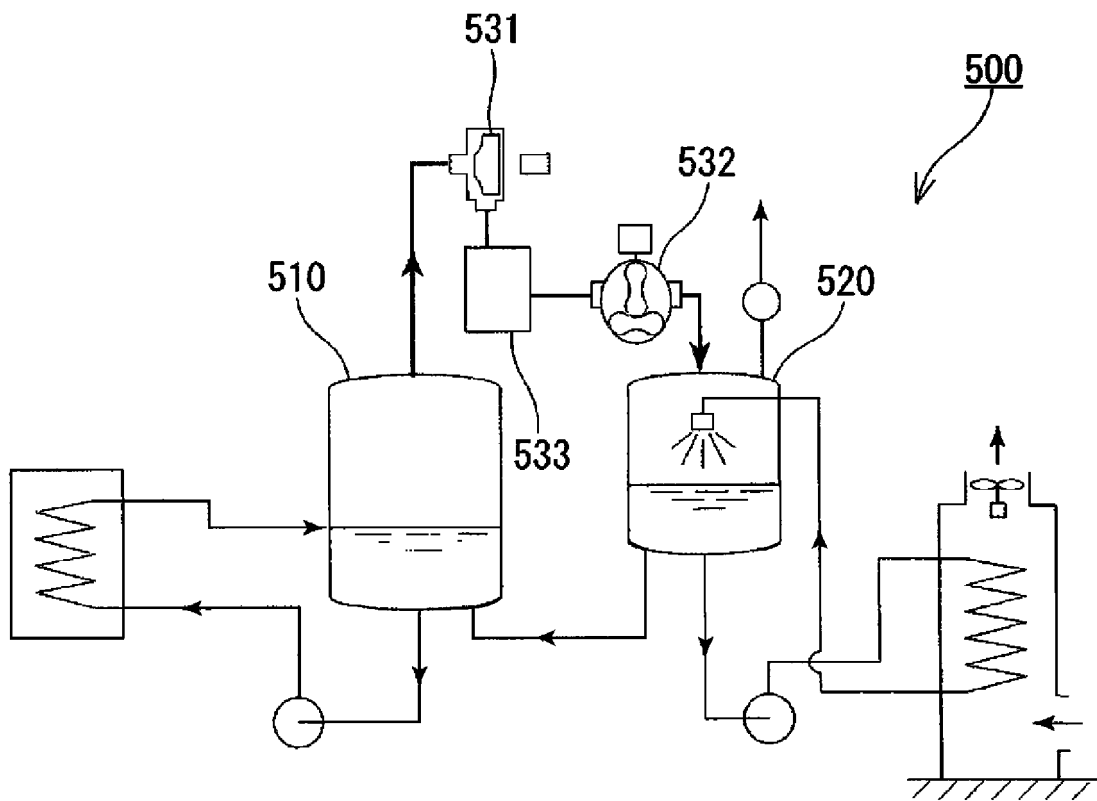


FIG. 21



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2019/000136

A. CLASSIFICATION OF SUBJECT MATTER

Int.Cl. F04D29/58(2006.01)i, F25B1/00(2006.01)i, F25B1/053(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)

Int.Cl. F04D29/58, F25B1/00, F25B1/053

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

Published examined utility model applications of Japan 1922-1996

Published unexamined utility model applications of Japan 1971-2019

Registered utility model specifications of Japan 1996-2019

Published registered utility model applications of Japan 1994-2019

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	CN 101210574 A (INDUSTRIAL TECHNOLOGY RESEARCH INSTITUTE) 02 July 2008, page 7, line 31 to page 8, line 13 (Family: none)	1-19
A	JP 2017-194042 A (MITSUBISHI HEAVY INDUSTRIES THERMAL SYSTEMS, LTD.) 26 October 2017, fig. 1 & WO 2017/183644 A1	1-19

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

* Special categories of cited documents:

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

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

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Date of the actual completion of the international search
05 April 2019 (05.04.2019)Date of mailing of the international search report
16 April 2019 (16.04.2019)Name and mailing address of the ISA/
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Patent documents cited in the description

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