(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) **EP 4 542 132 A1**

(12) **EUROPEAN PATENT APPLICATION**
published in accordance with Art. 153(4) EPC

(43) Date of publication:
23.04.2025 Bulletin 2025/17

(21) Application number: **24780851.2**

(22) Date of filing: **29.03.2024**

(51) International Patent Classification (IPC):
F25B 1/053 *(2006.01)* ***F25B 1/00*** *(2006.01)*

(52) Cooperative Patent Classification (CPC):
F25B 1/00; F25B 1/053

(86) International application number:
PCT/JP2024/013147

(87) International publication number:
WO 2024/204756 (03.10.2024 Gazette 2024/40)

(84) Designated Contracting States:
AL AT BE BG CH CY CZ DE DK EE ES FI FR GB GR HR HU IE IS IT LI LT LU LV MC ME MK MT NL NO PL PT RO RS SE SI SK SM TR
Designated Extension States:
BA
Designated Validation States:
GE KH MA MD TN

(30) Priority: **31.03.2023 JP 2023058725**

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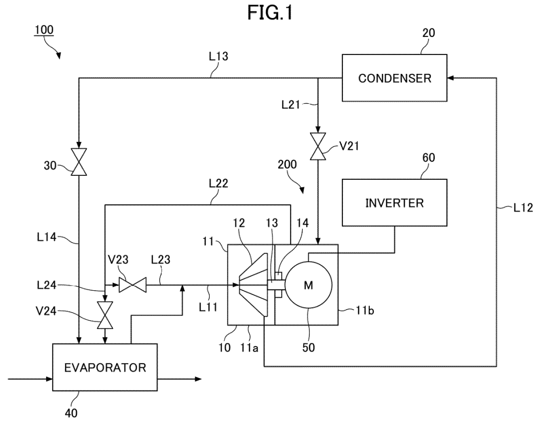
(54) REFRIGERATOR

(57) A refrigerator capable of preventing or substantially preventing occurrence of surging is provided. A refrigerator runs a refrigeration cycle and includes: a compressor configured to compress a refrigerant; a condenser configured to condense the compressed refrigerant; an expansion valve configured to expand the condensed refrigerant; an evaporator configured to evaporate the expanded refrigerant; a suction pipe configured to connect the evaporator and the compressor; a motor configured to drive the compressor; a first pipe configured to supply the refrigerant discharged from the condenser to a target part; a second pipe configured to supply the refrigerant having cooled down the target part to the suction pipe, the suction pipe serving as a first position; and a third pipe configured to supply the refrigerant having cooled down the target part to a second position, the second position being different from the first position and located inside the refrigerator.

FIG.1



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Description

Technical Field

[0001] The present disclosure relates to a refrigerator.

Related Art

[0002] A refrigerator includes a compressor, a condenser, an expansion valve, and an evaporator. The compressor here may be, in some cases, a turbo compressor. The turbo compressor here may include a rotary shaft. One end of the rotary shaft is provided with an impeller, and the other end of the rotary shaft is located in a motor chamber (see, for example, patent document 1). In the turbo compressor described in patent document 1, a low-temperature, low-pressure gas that is yet to be pressurized is introduced into the motor chamber.

Citation List

Patent Documents

[0003] Patent Document 1: Unexamined Japanese Patent Application No. H11-13697

Summary of the Invention

Problem to be Solved by the Invention

[0004] A compressor has various limitations or constraints; the compressor can operate only within a certain range that is determined based on these limitations. In the event these limitations or constraints are not satisfied, the operation of the refrigerator becomes unstable. The present disclosure therefore aims to provide a refrigerator that can expand the range in which the compressor can operate.

Means for Solving the Problem

[0005] According to one example of the present disclosure, a refrigerator runs a refrigeration cycle and includes: a compressor configured to compress a refrigerant; a condenser configured to condense the compressed refrigerant; an expansion valve configured to expand the condensed refrigerant; an evaporator configured to evaporate the expanded refrigerant; a suction pipe configured to connect the evaporator and the compressor; a motor configured to drive the compressor; a first pipe configured to supply the refrigerant discharged from the condenser to a target part; a second pipe configured to supply the refrigerant having cooled down the target part to the suction pipe, the suction pipe serving as a first position; and a third pipe configured to supply the refrigerant having cooled down the target part to a second position, the second position being different from the first position and located inside the refrigerator.

[0006] The refrigerator of this example may supply the refrigerant having cooled down the target part to the suction pipe that serves as the first position, or to the second position. The refrigerator may also supply the refrigerant having cooled down the target part to both the first position and the second position. With this refrigerator, the range in which the compressor can operate can be expanded by choosing where the refrigerant having cooled down the target part is supplied to.

[0007] With the refrigerator according to one example of the present disclosure, the compressor may be a turbo compressor. The refrigerator according to one example may thus choose where the refrigerant having cooled down the target part is supplied to, thereby changing the refrigerator's operating status and preventing or substantially preventing occurrence of surging.

[0008] With the refrigerator according to one example of the present disclosure, the target part may be a motor, a bearing that supports a rotary shaft of the compressor, or a controller for the motor.

[0009] The refrigerator of this example may cool down the motor, thereby preventing or substantially preventing the motor from producing heat. The refrigerator of this example may also cool down the bearing, thereby preventing or substantially preventing the bearing from burning out. The refrigerator of this example may furthermore cool down the controller, thereby preventing or substantially preventing the controller from malfunctioning.

[0010] With the refrigerator according to one example of the present disclosure, the evaporator may serve as the second position. The refrigerator of this example can bring back the refrigerant having cooled down the target part to the evaporator, so that the degree of superheat of the refrigerant gas that is supplied to the compressor can be made relatively low.

[0011] With the refrigerator according to one example of the present disclosure, the target part may be the motor, and the first pipe may be connected to a flow path that communicates with the inside of the casing of the motor. The refrigerator of this example may supply the refrigerant into the casing of the motor, thereby cooling down the motor directly. The refrigerator of this example may cool down the motor, thereby alleviating the motor's temperature-related limitations or constraints. The motor's temperature-related limitations or constraints may include, for example, temperature conditions for the motor's windings or magnets. The range in which the compressor can operate can be expanded by cooling down the motor's windings or magnets.

[0012] With the refrigerator according to one example of the present disclosure, the target part may be a bearing that supports the rotary shaft of the compressor, and the bearing may be an oil-less bearing. The refrigerator of this example may use a refrigerant as a gas to be supplied to the oil-less bearing. Consequently, the refrigerator of this example can alleviate the bearing's temperature-related limitations or constraints by cooling down the oil-less bearing. By this means, the range in which the compressor can operate can be expanded. The refrigerator of this example may employ an oil-less bearing, and therefore need not to use a bearing that uses a highly viscous fluid such as oil as a working fluid. By using an oil-less bearing, the bearing loss can be reduced even when the bearing rotates fast. Consequently, the limitations or constraints on the number of rotations of the bearing per unit time can be alleviated, so that the range in which the compressor can operate can be expanded.

[0013] With the refrigerator according to one example of the present disclosure, the evaporator may be an air-heat exchanger that transfers heat between the refrigerator and the air. The refrigerator of this example may exchange heat between the refrigerator and the air, thereby cooling down the air. An air-heat exchanger is likely to be affected by the outside air temperature, and so its temperature changes more significantly than that of a water-heat exchanger does. It then follows that the operating status of a compressor, which is provided downstream of the air-heat exchanger, is also likely to change often, and a refrigerator with an air-heat exchanger is required to have a wider range of operation than that of a refrigerator with a water-heat exchanger. The refrigerator of this example can expand its range of operation and therefore is suitable for use when the refrigerator has an air-heat exchanger that is likely to change its status of operation often.

[0014] A refrigerator according to one example of the present disclosure may include a first control valve provided in a second pipe, a second control valve provided in a third pipe, and a control part configured to control the first control valve and the second control valve. With this refrigerator according to one example, the control part may control the opening and closing operations of the first control valve and the second control valve, so that where the refrigerant having cooled down the target part is supplied to can be switched.

[0015] With the refrigerator according to one example of the present disclosure, the control part can execute a first operating mode in which the first control valve is open and the second control valve is closed, and execute a second operating mode in which the first control valve is closed and the second control valve is open.

[0016] The refrigerator according to this example can bring back the refrigerant having cooled down the target part to the suction pipe of the compressor by choosing the first operating mode. By this means, the degree of superheat of the refrigerant gas that is supplied to the compressor can be made relatively high. Also, the refrigerator of this example can bring back the refrigerant having cooled down the target part to the second position by choosing the second operating mode. The refrigerator according to this example may choose between the first operating mode and the second operating mode, thereby changing the degree of superheat of the refrigerant gas that is supplied to the compressor, and expanding the range in which the compressor can operate.

[0017] In the first operating mode, in which the refrigerant having cooled down the target part is brought back to the compressor's suction pipe, the compressing force required of the compressor increases as the degree of superheat of the refrigerant increases. Consequently, limitations or constraints related to the motor's outputs arise at operating points where the compressor is required to exert large force. In order to avoid having such limitations or constraints on the motor's outputs, the operating mode can be switched from the first operating mode to the second operating mode, thereby enabling the compressor to continue operating. Consequently, the range in which the compressor can operate can be expanded.

[0018] In the second operating mode, in which the refrigerant is brought back to the evaporator after cooling the target part, the refrigeration capacity decreases. In this second operating mode, a higher suction volumetric flow rate is required to ensure sufficient refrigeration capacity. Therefore, at operating points where high refrigeration capacity is required, limitations or constraints surface with regard to the number of rotations of the compressor per unit time. To avoid having such limitations or constraints on the number of rotations of the compressor per unit time, the second operating mode may be switched to the first operating mode, so that it is possible to ensure sufficient refrigeration capacity while maintaining the number of rotations of the compressor per unit time. With this example, it is possible to switch the first operating mode to the second operating mode or switch the second operating mode to the first operating mode, depending on the operating status of the compressor, thereby expanding the range in which the compressor can operate.

[0019] The refrigerator according to one example of the present disclosure may include a control part configured to control the flow rate of the refrigerant in the third pipe based on a surge line that indicates the boundary of the range in which the compressor has a surge. The refrigerator of this example can prevent or substantially prevent occurrence of surging in the compressor by controlling the flow rate of the refrigerant in the third pipe based on the surge line.

[0020] With the refrigerator according to one example of the present disclosure, the control part may be configured to control the flow rate of the refrigerant in the third pipe based on an operational range boundary, which marks the range of operation on the side, relative to the surge line, where the flow rate of the refrigerant is large, and which takes into account a margin with respect to the surge line. The refrigerator of this example may thus control the flow rate of the refrigerant in the third pipe based on an operational range boundary that takes into account a margin with respect to a surge line, thereby reliably preventing or substantially preventing the compressor from having a surge.

[0021] With the refrigerator according to one example of the present disclosure, the evaporator may be the second position, and the control part may be configured to predict an operating point of the compressor at a second point in time, which is a predetermined period of time after the current first point in time, and increase the flow rate of the refrigerant in the third pipe when the predicted operating point is in a range that is closer to the surge line than to the operational range boundary or is in the range in which surging occurs. The refrigerator of this example can thus predict the operating point of the compressor at a second point in time, which comes later in time than the current point in time, and predict occurrence of surging. The refrigerator of this example may be configured to increase the flow rate of the refrigerant in the third pipe in the event the compressor is likely to have a surge, thereby preventing or substantially preventing the compressor from having a surge.

Brief Description of the Drawings

[0022]

[FIG. 1] FIG. 1 is a schematic diagram that shows a refrigerator according to one embodiment;

[FIG. 2] FIG. 2 is a block diagram that shows a control part of the refrigerator;

[FIG. 3] FIG. 3 is a functional block diagram that shows the control part of the refrigerator;

[FIG. 4] FIG. 4 is a Mollier diagram of the refrigerator's refrigeration cycle;

[FIG. 5] FIG. 5 is a graph that shows a compressor's performance curve;

[FIG. 6] FIG. 6 is a graph that shows the relationship between volumetric flow rate and adiabatic efficiency in the compressor.

[FIG. 7] FIG. 7 is a flowchart that shows an example control flow in the refrigerator.

Detailed Description of Preferred Embodiments

[0023] A non-limiting embodiment of the present disclosure will be described below with reference to the accompanying drawings. Note that in the accompanying drawings, members or parts that are the same as or similar to each other will be assigned the same or similar reference codes. Also, members or parts that are the same as or similar to each other will be described once and not be described in a redundant manner. Also, the members or parts in the drawings are not necessarily drawn to scale. Therefore, a person skilled in the art can determine any detailed dimensions based on his/her choice by referring to the following non-limiting embodiment. Furthermore, the following embodiment is simply illustrative and is not meant to limit the present disclosure in any way. Features described in the following embodiment and their combinations are not necessarily material or essential to the present disclosure.

[Overview of Refrigerator]

[0024] The refrigerator 100 shown in FIG. 1 is used in, for example, air conditioners, refrigerator machinery, and freezer machinery. The refrigerator 100 may be used in other appliances and machines as well. The refrigerator 100 runs a refrigeration cycle. The refrigeration cycle of the refrigerator 100 is a vapor compression refrigeration cycle. The refrigerator 100 includes a compressor 10, a condenser 20, an expansion valve 30, and an evaporator 40.

[0025] The working fluid of the refrigerator 100 serves as the refrigerant, but this is by no means a limitation. The compressor 10 compresses the refrigerant gas. The condenser 20 condenses the refrigerant gas compressed by the compressor 10. The expansion valve 30 expands the refrigerant condensed by the condenser 20. The evaporator 40 evaporates the refrigerant expanded by the expansion valve 30. The refrigerant gas evaporated by the evaporator 40 is sucked into the compressor 10.

[0026] The compressor 10 performs reversible adiabatic compression on the refrigerant gas. The refrigerant gas supplied to the condenser 20 releases heat under constant pressure and is liquefied. The liquefied refrigerant expands irreversibly at constant enthalpy in the expansion valve 30, and thereupon part of the refrigerant evaporates. The refrigerant absorbs heat under constant pressure in the evaporator 40.

[0027] The refrigerator 100 has pipes L11 to L14, in which the refrigerant flows. The pipe L11 is a suction pipe that connects the evaporator 40 and the compressor 10. The pipe L12 connects the compressor 10 and the condenser 20. The pipe L13 connects the condenser 20 and the expansion valve 30. The pipe L14 connects the expansion valve 30 and the evaporator 40.

[0028] The refrigerant gas flows in the pipe L11 and is sucked into the compressor 10. The refrigerant gas compressed in the compressor 10 flows in the pipe L12 and is supplied to the condenser 20. The refrigerant liquid liquefied in the condenser 20 flows in the pipe L13 and enters the expansion valve 30. The refrigerant expanded in the expansion valve 30 flows in the pipe L14 and is supplied to the evaporator 40. The refrigerant gas absorbs heat in the evaporator 40, flows in the pipe L11, and is supplied to the compressor 10.

[0029] The compressor 10 is, for example, a turbo compressor. The compressor 10 includes a casing 11, an impeller 12, a drive shaft 13, a bearing 14, and a motor 50. The casing 11 houses the impeller 12, the drive shaft 13, the bearing 14, and the motor 50.

[0030] The casing 11 has a compression chamber 11a that houses the impeller 12 and a motor chamber 11b that houses the motor 50.

[0031] The impeller 12 is provided at one end of the drive shaft 13. The other end of the drive shaft 13 is located inside the motor chamber 11b. The drive shaft 13 includes the rotary shaft of the motor 50.

[0032] The bearing 14 rotatably supports the drive shaft 13. The bearing 14 is fastened to the casing 11. The bearing 14 may be, for example, a radial bearing. The compressor 10 has multiple bearings 14. The bearing 14 may be, for example, an oil-less bearing. The bearing 14 may be a slide bearing or a ball bearing.

[0033] The motor 50 is the driving source of the compressor 10. The motor 50 has a rotor and a stator. The rotor is fastened to the drive shaft 13 and rotates together with the drive shaft 13. The stator is fastened to the casing 11 and placed around the rotor.

[0034] The refrigerator 100 includes inverter 60. The inverter 60 controls the number of rotations of the motor 50 per unit time. The inverter 60 is a controller that controls the operating frequency of the motor 50.

[0035] The impeller 12 of the compressor 10 rotates by receiving rotational driving force from the motor 50. The impeller 12 rotates, and the refrigerant gas is compressed.

[0036] The compressor 10 is not limited to a turbo compressor (centrifugal compressor) and may be a displacement compressor as well. The displacement compressor may be, for example, a rotary type, a scroll type, a reciprocating type, or a screw type.

[0037] The condenser 20 is a heat exchanger that cools down the high-temperature, high-pressure evaporated refrigerant compressed by the compressor 10. The condenser 20 condenses the refrigerant by exchanging heat between the refrigerant and, for example, water or air. The heat exchanger, serving also as the condenser 20, may be, for example, a water-cooled type, an air-cooled type, an evaporator type, or some other type. The heat exchanger may be a shell-and-tube type, a double-tube type, a plate-fin type, or some other type.

[0038] The expansion valve 30 throttles and expands the high-pressure refrigerant liquid coming out of the condenser 20. The refrigerant having passed the expansion valve 30 is in a low-pressure, low-temperature state. The refrigerant having passed the expansion valve 30 is partially evaporated, and in a wet vapor state in which saturated vapor and saturated liquid coexist.

[0039] The evaporator 40 is a heat exchanger that exchanges heat between the expanded refrigerant leaving the expansion valve 30 and the fluid to be cooled. The fluid to be cooled here refers to the fluid that is targeted for cooling. The fluid to be cooled is, for example, the air or water. The fluid to be cooled can also be other fluids. In the evaporator 40, the refrigerant, which is in a wet vapor state, absorbs heat from the fluid to be cooled and evaporates. The evaporator 40 may be, for example, an air-heat exchanger. An air-heat exchanger transfers heat between the refrigerant and the air.

[0040] The evaporator 40 may be, for example, a dry type, a flooded type, or a forced liquid circulation type. The evaporator 40 may be, for example, a plate fin coil type, or a shell and tube type.

[Cooling mechanism]

[0041] Next, the cooling mechanism 200 will be described. The refrigerator 100 includes the cooling mechanism 200 that cools a target part by using the refrigerant discharged from the condenser 20. The target part is, for example, the motor 50. The target part may be the bearing 14 or may be the inverter 60, which serves as a controller for the motor 50. The target part may be other parts as well. The cooling mechanism 200 may cool down multiple target parts. The cooling mechanism 200 may cool down the motor 50, the bearing 14, and the inverter 60. The cooling mechanism 200 may cool down the motor 50 housed in the casing 11 by cooling down the casing 11, for example. The casing 11 of the compressor 10 may be the target part to be cooled down by the cooling mechanism 200.

[0042] The target part may be located at a position away from the compressor 10. The inverter 60 may be mounted on the compressor 10 or may be located at a position away from the compressor 10.

[0043] The cooling mechanism 200 includes pipes L21 to L24, an expansion valve V21, and control valves V23 and V24. The pipe L21 connects the condenser 20 and the motor chamber 11b. The pipe L21 connects the pipe L13, which is connected to the outlet of the condenser 20, and the motor chamber 11b. The pipe L21 is a pipe for supplying the refrigerant leaving the condenser 20, to the target part.

[0044] The pipe L21 is provided with the expansion valve V21. The expansion valve V21 can adjust the flow rate and pressure of the refrigerant that flows in the pipe L21. The refrigerant having passed the expansion valve V21 is reduced in pressure and therefore is in a low-temperature state. The low-temperature refrigerant flows in the pipe L21 and is supplied to the target part. The pipe L21 is connected to, for example, the inside of the motor chamber 11b of the casing 11. The refrigerant having flown through the pipe L21 is supplied to the motor chamber 11b and cools the motor 50.

[0045] The pipe L22 is connected to the motor chamber 11b. After cooling the motor 50, the refrigerant flows in the pipe L22 and is discharged outside the motor chamber 11b. The outlet side of the pipe L22 is branched and connected to the pipe L23 and the pipe L24.

[0046] The pipe L23 connects the pipe L22 and the pipe L11. The pipe L24 connects the pipe L22 and the evaporator 40. The pipe L11 is a suction pipe, which marks and which hereinafter will be referred to as the "first position." The evaporator 40 is an example of a second position, which is different from the first position. The second position is included in the refrigerator 100. The second position is not limited to that of the evaporator 40 and may assume other positions as well. For example, if the compressor 10 is a multi-stage compressor, the second position may be an inlet of a subsequent compression mechanism (for example, an impeller). Also, if the compressor 10 is a multi-stage compressor, the first position may be a suction pipe to a subsequent compression mechanism. There may be multiple first positions in the refrigerator 100.

[0047] The pipe L21 is an example of a first pipe for supplying the refrigerant discharged from the condenser 20 to the target part. The pipe L23 is an example of a second pipe for supplying the refrigerant having cooled down the target part to a suction pipe (pipe L11), which marks and will be hereinafter referred to as the "first position." The pipe L24 is an example of a third pipe for supplying the refrigerant having cooled down the target part to a "second position," which is different from the first position.

[0048] The pipe L23 is provided with a control valve V23, and the pipe L24 is provided with a control valve V24. The control valve V23 is an example of a first control valve. The control valve V24 is an example of a second control valve. In the cooling mechanism 200, by controlling the opening and closing operations of the control valve V23 and the control valve V24, it is possible to choose the place to which the refrigerant having cooled down the motor 50 is supplied.

[0049] In the refrigerator 100, a first operating mode can be executed, in which the control valve V23 is open and the control valve V24 is closed. In the first operating mode, the refrigerator 100 can bring back the refrigerant having cooled down the motor 50 to the pipe L11, which is a suction pipe.

[0050] Also, in the refrigerator 100, a second operating mode can be executed, in which the control valve V23 is closed and the control valve V24 is open. In the second operating mode, the refrigerator 100 can bring back the refrigerant having cooled down the motor 50 to the evaporator 40. The pipe L24 may be connected to the evaporator 40, or may be connected to the pipe L14 that is connected to the evaporator 40. The refrigerant having cooled down the target part may be supplied directly to the evaporator 40, or may be supplied indirectly to the evaporator 40 via the pipe L14.

[0051] In the refrigerator 100, the opening and closing operations of the control valve V23 and the control valve V24 may be controlled to bring back the refrigerant having cooled down the target part to the pipe L11 alone, to the evaporator 40 alone, or to both the pipe L11 and the evaporator 40. Also, the refrigerator 100 may be provided with a control valve, which is a three-way valve, instead of the control valves V23 and V24. Also, the refrigerator 100 may be provided with the control valve V23 in the pipe L23, and the control valve V24 need not be provided in the pipe L24. Similarly, the refrigerator 100 may be provided with the control valve V24 in the pipe L24, and the control valve V23 need not be provided in the pipe L23.

[Surging]

[0052] Next, surging will be described. FIG. 5 is a graph that shows the compressor's performance curves. In FIG. 5, the horizontal axis is the flow rate, and the vertical axis is the adiabatic head. In FIG. 6, the horizontal axis is the volumetric flow rate, and the vertical axis is the adiabatic efficiency. FIG. 5 shows performance curves G1 to G3. The performance curves G1 to G3 show respective adiabatic heads at varying numbers of rotations per unit time.

[0053] Axial compressors, centrifugal compressors, and other compressors used for various purposes may experience surging during operation. For example, in a centrifugal compressor, if the pressure is increased while the flow rate is throttled during operation, rotating stall or surging will occur. Generally speaking, as the flow rate becomes lower, rotating stall will occur, and eventually surging will occur.

[0054] For example, assuming that a centrifugal compressor is operating, if the flow rate crosses a certain flow rate and keeps dropping, a backflow range will form in the circumferential direction inside the centrifugal compressor (turbo machine). The occurrence of this type of backflow range is referred to as "rotating stall." The phenomenon in which a backflow range forms is a localized phenomenon inside the compressor and varies in the circumferential direction.

[0055] When surging occurs in the compressor, the flow rate and pressure fluctuate significantly, throughout the piping connected to the compressor, due to backflow and pulsation. In this case, the equipment, including the piping, is placed under heavy strain, and therefore the compressor is unable to continue operating.

[0056] For example, it is preferable to conduct an operational test to predict the occurrence of surging and take a safety factor into account such that the compressor's operating point does not enter the range in which surging occurs. By operating the compressor in a range in which the flow rate is higher than the operating point that takes the safety factor into account, occurrence of surging can be prevented or substantially prevented.

[0057] For example, in operation in which the compressor's rotation speed is controlled and changed using an inverter or the like, the flow rate at which surging occurs is determined per rotation speed. By connecting these rotation speed-specific operating points at which surging occurs, a surge line SL1 (that is, the boundary of the compressor's operational range) is determined by the volumetric flow rate and the adiabatic head.

[0058] FIG. 5 shows a surge line SL1, a line SL2 that marks the operational range boundary, and a line SL3 at which rotating stall occurs. The operational range boundary SL2 is in the range of operation on the side, relative to the surge line, where the refrigerant flow rate is high. The operational range boundary SL2 is a line that takes into account a margin SR with respect to the surge line SL1.

[0059] In FIG. 5, the area to the left of the surge line SL1 is where surging occurs. The area to the right of the operational range boundary SL2 is where the compressor can operate.

[Issues with Related Art]

[0060] Issues with related art will be explained now. For example, in an air conditioner according to related art, when the flow rate is low (that is, the refrigeration capacity is low) and the head is high (that is, the pressure gap is wide), a centrifugal compressor's operating point enters a range beyond the surge line SL1, and the centrifugal compressor is no longer able to operate.

[Control Part]

[0061] Next, the control part 210 of the refrigerator 100 will be described with reference to FIG. 2. FIG. 2 is a block diagram that shows the control part 210 of the refrigerator 100. FIG. 2 shows a hardware structure of the control part 210. As shown in FIG. 2, the control part 210 is electrically connected to various sensors 220, the inverter 60, the expansion valve 30, the expansion valve V21, and the control valves V23 and V24.

[0062] The sensors 220 may include, for example, a temperature sensor for measuring the temperature (outside air temperature) outside the refrigerator 100 (outside air temperature sensor), a temperature sensor for measuring the temperature of the refrigerant, a pressure sensor for measuring the pressure of the refrigerant, a flow rate sensor for measuring the flow rate of the refrigerant, and so forth. The sensors 220 may also include other sensors for acquiring different pieces of information.

[0063] These sensors 220 may include, for example, a suction pressure sensor 221, a suction temperature sensor 222, a discharge pressure sensor 223, a discharge temperature sensor 224, and a water temperature sensor 225. The suction pressure sensor 221 detects the pressure of the refrigerant gas that is sucked into the compressor 10. The suction temperature sensor 222 detects the temperature of the refrigerant gas that is sucked into the compressor 10. The discharge pressure sensor 223 detects the pressure of the refrigerant gas discharged from the compressor 10. The discharge temperature sensor 224 detects the temperature of the refrigerant gas discharged from the compressor 10.

[0064] For example, if the evaporator is a water-heat exchanger, the water temperature sensor 225 detects the temperature of the water that flows into the evaporator 40. The water temperature sensor 226 detects the temperature of the water that flows out of the evaporator 40. The control part 210 receives the data detected by the sensors 220 as inputs. The control part 210 can calculate various data using the data acquired by the sensors 220. Using the data acquired from the water temperature sensors 225 and 226, the control part 210 can calculate the volumetric flow rate of the refrigerant gas that is sucked into the compressor 10. The control part 210 can calculate the volumetric flow rate of the refrigerant gas based on the amount of heat transfer in the evaporator 40. The control part 210 can calculate the volumetric flow rate of the refrigerant gas based on the entropy in the evaporator 40. In case a water-heat exchanger or the like is used, the volumetric flow rate of the refrigerant gas may be calculated using a flow rate meter for measuring the refrigerant's flow rate.

[0065] The control part 210 includes a CPU 211 and a storage part 212. The CPU (Center Processing Unit) 211 is responsible for overall processing in the refrigerator 100. The CPU 211 can control the number of rotations of the motor 50 per unit time via the inverter 60. The CPU 211 can control the opening and closing operations of the expansion valve 30. The CPU 211 can control the opening and closing operations of the expansion valve V21. The CPU 211 can control the opening and closing operations of the control valves V23 and V24.

[0066] The storage part 212 includes a ROM (Read Only Memory) 213 and a RAM (Random Access Memory) 214. The ROM 213 stores various programs for causing the CPU 211 to execute control processes, as well as various data necessary for the operation of the refrigerator 100. The RAM 214 temporarily stores the data obtained from the sensor 220.

[Functional Blocks]

[0067] Next, referring to FIG. 3, the functional blocks in the control part 210 will be described. FIG. 3 is a functional block diagram that shows the control part 210 of the refrigerator 100. In the control part 210, functional blocks that implement the functions of the refrigerator 100 are configured by combining hardware and software control parts.

[0068] In the refrigerator 100, each function of the embodiment can be implemented by one or more processing circuits. In this specification, "processing circuit" refers to a processor that is programmed to implement the above-described functions by software, such as a processor implemented by an electronic circuit, as well as devices such as an ASIC (Application-Specific Integrated Circuit), DSP (digital signal processor), FPGA (field programmable gate array), and other circuit modules designed to implement the functions described hereinabove.

[0069] The CPU 211 of the control part 210 executes programs stored in a storage part 212 such as a ROM 213 to implement the functions of the motor control part 231, the expansion valve control part 232, and the valve control part 233 shown in FIG. 3. Note that external devices and sensors may be connected to the control part 210 and implement some of these functions. Also, the functions of the control part 210 are by no means limited to these.

[0070] The motor control part 231 can control the number of rotations of the motor 50 per unit time via the inverter 60. By controlling the number of rotations of the motor 50 per unit time, the control part 210 can control the number of rotations of the impeller 12 per unit time in the compressor 10. The expansion valve control part 232 may control the expansion valve 30 to control the pressure of the refrigerant having passed the expansion valve 30.

[0071] The valve control part 233 can control the opening and closing operations of the expansion valve V21. The valve control part 233 can control the opening and closing operations of the control valve V23. The valve control part 233 can control the opening and closing operations of the control valve V24. The valve control part 233 may control the opening and closing operations of the expansion valve V21 to adjust the pressure of the refrigerant that is supplied to the motor 50.

[0072] The valve control part 233 can switch the place to which the refrigerant having cooled down the target part is supplied, by controlling the opening and closing operations of the control valve V23 and the control valve V24. In the refrigerator 100, the first operating mode can be executed by supplying the refrigerant having cooled down the target part to the pipe L11. In the refrigerator 100, the second operating mode can be executed by supplying the refrigerant having cooled down the target part to the evaporator 40.

[0073] As described above, the motor control part 231, the expansion valve control part 232, and the valve control part 233 can be implemented by software, based on the programs stored in the storage part 212. All or part of the motor control part 231, the expansion valve control part 232, and the valve control part 233 may be implemented by hardware such as an IC (Integrated Circuit).

[Ph diagrams]

[0074] Next, referring to FIG. 4, Mollier diagrams of the refrigeration cycle of the refrigerator 100 will be explained. In FIG. 4 the horizontal axis is enthalpy h, and the vertical axis is pressure P. Ph1 is a Ph diagram of the first operating mode. Ph2 is a Ph diagram of the second operating mode.

[0075] The change of status "1→2" is caused by reversible adiabatic compression by the compressor 10, and is an isentropic change. An enthalpy change that occurs when an isentropic change occurs is referred to as an "adiabatic head." The change of status "2→3" is an isobaric change in the condenser 20, and the refrigerant is compressed and cooled down. The change of status "3→4" is caused by throttling expansion by the expansion valve 30, and is a geometric enthalpy change. The change of status "4→1" is an isobaric change in the evaporator 40, and the refrigerant is evaporated and heated. Although the actual compression step "1→2" deviates from reversible adiabatic compression, past knowledge has shown that, as long as a turbo compressor maintains the same number of rotations per unit time and volumetric flow rate, the compressor's operating point shows the same adiabatic head characteristics. Therefore, the description in this specification will focus on the reversible adiabatic compression step and the adiabatic head.

[Ph diagram in the first operating mode]

[0076] Ph1 shows a case in which the refrigerant having cooled down the target part is brought back to the pipe L11, which is a suction gas line, in the first operating mode. In the first operating mode, the degree of superheat of the refrigerant gas that is sucked into the compressor 10 is relatively high. The refrigerant having cooled down the target part flows from the pipe L23 into the pipe L11, so that the temperature of the refrigerant gas compressed in and discharged from the compressor 10 becomes higher than in the second operating mode.

[0077] Also, in the first operating mode, the pressure increase ΔP1 through the compressor 10 is smaller than the pressure increase ΔP2 in the second operating mode. Note that, in comparison between the first operating mode and the second operating mode, the number of rotations of the compressor 10 per unit time and the volumetric flow rate are the same. Similarly, in the comparison with related art, the number of rotations of the compressor 10 per unit time and the volumetric flow rate are the same. The pressure increase ΔP1 is the difference between the pressure P21 and the pressure P1. The pressure increase ΔP2 is the difference between the pressure P22 and the pressure P1. The pressure P21 after the discharge from the compressor 10 in the first operating mode is lower than the pressure P22 after the discharge from the compressor 10 in the second operating mode.

[0078] In the first operating mode, when the desired temperature difference ΔT in the refrigerator 100 is small, the refrigerator 100 can continue operating at a point where the adiabatic efficiency is high, rather than operating at an operating point where the adiabatic efficiency is low. By this means, the range of operation in the refrigerator 100 can be expanded compared to related art. Thus, the range of operation can be expanded while preventing or substantially preventing occurrence of surging in the refrigerator 100.

[0079] FIG. 5 is a graph that shows the compressor's performance curves. In FIG. 5, the horizontal axis is the volumetric flow rate, and the vertical axis is the adiabatic efficiency. FIG. 5 shows performance curves G1 and G2. The performance curves G1 and G2 show respective adiabatic heads at varying numbers of rotations per unit time. FIG. 5 shows operating points P21 and P22.

[0080] Also, FIG. 5 shows a surge line SL1. The surge line SL1 shows where surging occurs. When the volumetric flow rate hits or falls below the surge line SL1, surging occurs, which makes the flow of the refrigerant gas unstable.

[0081] The operating point P21 is an operating point where the volumetric flow rate is higher and the adiabatic head is lower than at the operating point P22. The operating point P22 is an operating point where the volumetric flow rate is lower and the adiabatic head is higher than at the operating points P21. The operating point P23 is an operating point where the volumetric flow rate is lower and the adiabatic head is higher than at the operating points P21 and P22. The volumetric flow rate at the operating point P23 hits or falls below the surge line SL. The operating point P23 is a virtual operating point in the range in which the compressor cannot operate.

[0082] FIG. 6 is a graph that shows the relationship between the volumetric flow rate and the adiabatic efficiency. In FIG. 6, the horizontal axis is the volumetric flow rate, and the vertical axis is the adiabatic efficiency. FIG. 6 shows a graph G3. The adiabatic efficiency at the operating point P21 is lower than the adiabatic efficiency at the operating point P22. The adiabatic efficiency at the operating point P22 is higher than the adiabatic efficiency at the operating point P21. The adiabatic efficiency at the operating point P22 is close to the peak of the compressor 10's adiabatic efficiency.

[0083] The refrigerator 100 can switch the mode of operation, and also change the operating points P21 and P22. In the refrigerator 100, the risk of surging can be reduced by switching to the operating point P21, which is the farther one from the surge line SL1. Also, in the refrigerator 100, by switching the mode of operation, it is possible to choose the operating point P22 where the adiabatic efficiency is higher or choose the operating point P21 where the adiabatic efficiency is lower.

[Ph diagram in the second operating mode]

[0084] Ph2 shows a case in which the refrigerant having cooled down the target part is brought back to the evaporator 40 in the second operating mode. In the second operating mode, the degree of superheat of the refrigerant gas sucked in the compressor 10 is relatively low. In the second operating mode, the degree of superheat of the refrigerant gas sucked in the compressor 10 is lower than the degree of superheat of the refrigerant gas sucked in the compressor 10 in the first operating mode.

[0085] Also, in the second operating mode, the pressure increase $\Delta P2$ through the compressor 10 is larger than the pressure increase $\Delta P1$ in the first operating mode. Therefore, in the second operating mode, the pressure can be increased to a pressure P22, which otherwise cannot be reached because the number of rotations of the drive shaft 13 of the compressor 10 per unit time is limited. In some cases, the limit on the number of rotations of the compressor 10 per unit time may depend on the strength of rotating bodies, such as the impeller 12, the drive shaft 13, and the rotor of the motor 50. Also, in other cases, the limit on the number of rotations of the compressor 10 per unit time may depend on, for example, the axial resonance of rotating bodies. Also, in yet other cases, the limit on the number of rotations of the compressor 10 per unit time may depend on, for example, the strength of the bearing 14 that supports the drive shaft 13. The refrigerator 100 executes the second operating mode, the pressure of the refrigerant gas discharged from the compressor 10 can be increased, compared to any conventional method, at the same number of rotations per unit time and the same volumetric flow rate. By this means, the range of operation for the refrigerator 100 can be expanded compared to any conventional method. Thus, the range of operation for the refrigerator 100 can be expanded while preventing or substantially preventing occurrence of surging.

[0086] In the second operating mode, the refrigerant's circulation volumetric flow rate in the refrigerator 100 is low, so that the range of operation for the refrigerator 100 can be expanded to a range in which the refrigerator 100 cannot otherwise operate because it enters a surging range.

[0087] Also, in the second operating mode, the refrigeration capacity of the evaporator 40 is lower than in the first operating mode, and the refrigeration capacity of the evaporator 40 can be adjusted at the same desired temperature difference ΔT. For example, in heating operation of the refrigerator 100, the heating capacity is not affected and high pressure can be maintained while operating.

[Comparison between the first operating mode and the second operating mode]

[0088] The adiabatic head $\Delta h1$ of the compressor 10 in the first operating mode is the same as the adiabatic head $\Delta h2$ of the compressor 10 in the second operating mode.

[0089] The inclination of the isentropic line in the first operating mode is different from the inclination of the isentropic line in the second operating mode. The isentropic line in the second operating mode is inclined more than the isentropic line in the first operating mode is. In the second operating mode, the pressure P22 of the refrigerant gas compressed by the compressor 10 is higher than the pressure P21 of the refrigerant gas compressed by the compressor in the first operating mode.

[0090] The enthalpy h11 in the state "1" as of when the refrigerant gas is sucked into the compressor 10 in the first operating mode is higher than the enthalpy h12 in the state "1" as of when the refrigerant gas is sucked into the compressor 10 in the second operating mode. Furthermore, the degree of suction superheat in the first operating mode is higher than the degree of suction superheat in the second operating mode ($\Delta h3=h11-h12$).

[0091] As for the properties of the refrigerant, when the degree of suction superheat increases, the inclination of the isentropic line ($1\rightarrow2$) becomes smaller. That is, as long as the compressor 10 maintains the same number of rotations per unit time and the same volumetric flow rate, the adiabatic heads $\Delta h1$ and $\Delta h2$ are the same, but the pressure increases $\Delta P1$ and $\Delta P2$ vary. The reaching pressure P22 in the second operating mode is higher than the reaching pressure P21 in the first operating mode. In other words, when the degree of suction superheat is large, the reaching pressure P21 of the refrigerant after compression by the compressor 10 is lower ($P21<P22$) than when the degree of suction superheat

is low.

[Control procedures in the refrigerator 100]

[0092] Next, the control procedures in the refrigerator 100 will be explained with reference to FIG. 7. FIG. 7 is a flowchart that shows an example control flow in the refrigerator 100.

[0093] First, the control part 210 of the refrigerator 100 receives various data as inputs. The control part 210 receives various data from the sensors 220 as inputs. The control part 210 can also receive various signals from other input parts and switches. The control part 210 may also receive various data from external processing devices and terminals as inputs. The control part 210 can receive, for example, data about the outside air temperature, data about the heat load, and data about the target water temperature (room temperature), as inputs. The control part 210 may calculate various data from the input data.

[0094] The control part 210 determines whether or not an end signal has been received (step S12). For example, when the user wants to end the operation of the refrigerator 100, the user operates a switch. By operating the switch, an end signal is output. When the control part 210 receives the end signal, the process is terminated here ("YES" in step S12). In the event the control part 210 does not receive the end signal, the process of step S12 is executed ("NO" in step S12).

[0095] Next, the control part 210 checks the operating point information of the compressor 10 after the next control is executed (step S13).

[0096] Next, the control part 210 calculates the adiabatic head and volumetric flow rate at operating points (step S14). The control part 210 can calculate the adiabatic head and volumetric flow rate at operating points using various input data. Furthermore, based on the adiabatic heads and volumetric flow rates calculated thus, the control part 210 can calculate a future operating point after a predetermined period of time has elapsed. The control part 210 may calculate a future operating point by using, for example, the map shown in FIG. 5.

[0097] Next, the control part 210 determines whether the future operating point exceeds the operational range boundary SL2 (step S15). The control part 210, for example, references the map shown in FIG. 5 and determines whether or not the future operating point is in the area to the left of the operational range boundary SL2. If the control part 210 predicts that the future operating point will be in the area to the left of the operational range boundary SL2, the control part 210 determines that the future point will exceed the operational range boundary SL2. If the volumetric flow rate is lower than the state at the operational range boundary SL2, the operating point is located to the left of the operational range boundary SL2. If the volumetric flow rate is higher than the state at the operational range boundary SL2, the operating point is located to the right of the operational range boundary SL2.

[0098] If the future operating point exceeds the operational range boundary SL2 ("YES" in step S15), the control part 210 executes the process of step S18. If the future operating point does not exceed the operational range boundary SL2 ("NO" in step S15), the control part 210 executes the process of step S16.

[0099] In step S16, the control part 210 executes control. "Control" here refers to, for example, controlling the number of rotations of the compressor 10 per unit time and controlling the actuation of valves. "Valves" here refers to, for example, the expansion valve 30, expansion valve V21, the control valve V23, and the control valve V24. The control part 210 may execute other types of controls as well. The control part 210 executes various controls and runs the refrigerator 100.

[0100] Next, the control part 210 checks the operating point information of the refrigerator 100 after the current control is executed (step S17). The control part 210 acquires various data and checks the operating point information of the refrigerator 100 after the current control is executed. As for the operating point information, the control part 210 checks, for example, the suction pressure and suction temperature of the refrigerant gas that is sucked into the compressor 10, the discharge pressure and discharge temperature of the refrigerant gas discharged from the compressor 10, and the volumetric flow rate of the refrigerant gas that is sucked into the compressor 10.

[0101] After executing the process of step S17, the control part 210 repeats steps S13 to S15.

[0102] In step S18, the control part 210 determines whether or not the place to which the intake gas is returned can be switched to the evaporator 40. The "place to which the intake gas is returned" refers to the place where the refrigerant that flows in the pipe L22 shown in FIG. 1 is brought back. The intake gas is brought back, for example, to the evaporator 40 or to the pipe L11.

[0103] If at present the intake gas is brought back to the pipe L11 ("YES" in step S18), the control part 210 determines that the place to which the intake gas is returned can be switched to the evaporator 40, and executes the process of step S19.

[0104] If at present the intake gas is brought back to the evaporator 40 ("NO" in step S18), the control part 210 does not determine that the place to which the intake gas is returned can be switched to the evaporator 40, and executes step S20 instead.

[0105] In step S19, the place to which the intake gas is returned is switched from the pipe L11 to the evaporator 40. In the refrigerator 100, the refrigerant that flows in the pipe L22 is supplied to the evaporator 40. After executing the process of step S19, the control part 210 repeats the processes of steps S17, S13, S14, and S15.

[0106] In step S20, the control part 210 issues an error signal and terminates the process here. After executing the process of step S20, the control part 210 may execute a process for stopping the operation of the refrigerator

100.

[Functions and effects of the refrigerator 100]

[0107] The refrigerator 100 according to this embodiment includes a pipe L23 for supplying the refrigerant having cooled down the motor 50, which is the target part, to the pipe L11 (the suction pipe of the first position), and a pipe L24 for supplying the refrigerant having cooled down the motor 50 to the evaporator 40 (the second position).

[0108] In this example, the refrigerator 100 can supply the refrigerant having cooled down the target part to the pipe L11 of the first position or to the evaporator 40 of the second position. The refrigerator 100 can supply the refrigerant having cooled down the target part to both the pipe L11 and the evaporator 40. The refrigerator 100 determines where the refrigerant having cooled down the target part is supplied to, so that the refrigerator 100's operating status can be changed to prevent or substantially prevent occurrence of surging. The refrigerator 100 can choose where the refrigerant having cooled down the target part is supplied to, so that the range in which the compressor 10 can operate can be expanded.

[0109] According to the refrigerator 100, even if the number of rotations of the compressor 10 per unit time is not changed, the refrigeration capacity of the evaporator 40 can be adjusted by changing the place to which the refrigerant having cooled down the target part is supplied. Also, according to the refrigerator 100, the range in which the compressor 10 can operate can be expanded by changing the place to which the refrigerant having cooled down the target part is supplied, so that the refrigerator 100 can operate even in a pressure range in which the refrigerator 100 cannot otherwise operate because it enters a surging range. The refrigerator 100 chooses the place to which the refrigerant having cooled down the target part is supplied, so that the refrigerator 100's operating status can be changed to prevent or substantially prevent occurrence of surging.

[0110] In the refrigerator 100, the target part is the motor 50. The refrigerator 100 according to this example cools down the motor 50, thereby preventing or substantially preventing the motor 50 from producing heat, and improving the reliability of the motor 50. With the refrigerator 100 according to this example, the limitations or constraints on the temperature of the motor 50 can be alleviated by cooling down the motor 50. The limitations or constraints on the temperature of the motor 50 include temperature conditions of the windings or magnets of the motor 50. The refrigerator 100 can cool down the windings and magnets of the motor 50, so that the range in which the compressor 10 can operate can be expanded.

[0111] In the refrigerator 100, the target part is not limited to the motor 50. The target part in the refrigerator 100 may be the bearing 14 that supports the drive shaft (rotary shaft) 13 of the compressor 10. In the refrigerator 100 structured this way, the bearing 14 can be cooled down, so that the bearing 14 can be prevented or substantially prevented from burning out. The refrigerator 100 according to this example can alleviate the limitations and constraints on the temperature of the bearing 14. The refrigerator 100 cools down the bearing 14, thereby preventing or substantially preventing the temperature of the bearing 14 from rising, and alleviating thus the limitations and constraints on the bearing 14. Therefore, the range in which the compressor 10 can operate can be expanded.

[0112] The target part in the refrigerator 100 may be the inverter 60, which is the controller for the motor 50. In the refrigerator 100 structured this way, by cooling down the inverter 60, it is possible to prevent or substantially prevent the inverter 60 from malfunctioning, thereby improving the reliability of the inverter 60.

[0113] In the refrigerator 100, the evaporator 40 marks the second position. With the refrigerator 100 according to this example, the refrigerant having cooled down the target part is supplied to the evaporator 40, so that the degree of superheat of the refrigerant gas that is supplied to the compressor 10 can be made relatively low.

[0114] In the second operating mode, in which the refrigerant is brought back to the evaporator 40 after cooling down the target part, the refrigeration capacity is reduced. In this second operating mode, it is necessary to increase the suction volumetric flow rate that is needed to ensure sufficient refrigeration capacity. Therefore, at operating points where high refrigeration capacity is required, the number of rotations of the compressor 10 per unit time is subject to limitations and constraints. To prevent the number of rotations of the compressor 10 per unit time from being limited or constrained this way, the second operating mode is switched to the first operating mode, so that it is possible to ensure sufficient refrigeration capacity while maintaining the number of rotations of the compressor 10 per unit time.

[0115] In the refrigerator 100, the motor 50 is the target part, and the pipe L21 is connected to a flow path that communicates with the inside of the motor chamber 11b. The pipe L21 may include a flow path that communicates with the inside of the motor chamber 11b. The flow path that communicates with the motor chamber 11b includes a flow path that penetrates the casing 11. The refrigerator 100 according to this example can supply the refrigerant to the inside of the motor chamber 11b, so that the rotary shaft, rotor, and stator of the motor 50 inside the casing 11 can be directly cooled down. With the refrigerator 100 according to this example, as described above, the limitations and constraints on the temperature of the motor 50 can be alleviated, so that the range in which the compressor 10 can operate can be expanded.

[0116] In the refrigerator 100, the bearing 14 that supports the drive shaft 13 of the compressor 10 may be the target part, and this bearing 14 may be an oil-less bearing. With the refrigerator 100 according to this example, a refrigerant can be used as a gas to be supplied to the oil-less bearing. By employing an oil-less bearing, the wear resistance and seizure resistance of the bearing 14 can be improved. Also, by employing an oil-less bearing, the maintenance of the bearing 14 can be reduced.

[0117] The refrigerator 100 of this example can thus alleviate the limitations and constraints on the temperature of the bearing 14 by cooling down the oil-less bearing. By this means, the range in which the compressor 10 can operate can be expanded. The refrigerator 100 according to this example employs an oil-less bearing, so that it is not necessary to use a bearing that uses a highly viscous fluid such as oil as the working fluid. When an oil-less bearing is used, the bearing loss can be reduced even when the speed of rotation is high. Consequently, the limitations or constraints on the number of rotations of the bearing 14 per unit time can be alleviated, so that the range in which the compressor 10 can operate can be expanded.

[0118] In the refrigerator 100, the evaporator 40 is an air-heat exchanger that transfers heat between the refrigerator and the air. The refrigerator 100 according to this example can cool down the air by exchanging heat between the refrigerator and the air. The refrigerator 100 can be applied to air conditioners, and can control the temperature of the air in a house, for example.

[0119] An air-heat exchanger is affected by the outside air temperature, so its temperature changes more significantly than that of a water-heat exchanger. Consequently, the operating status of the compressor 10 provided downstream of the air-heat exchanger is more prone to change, and therefore the refrigerator 100 with an air-heat exchanger requires a wider range of operation than that of a refrigerator with a water-heat exchanger. The refrigerator 100 according to this example can expand the range in which the compressor 10 can operate, which is effective for a refrigerator with an air-heat exchanger whose operating status is prone to change.

[0120] The refrigerator 100 includes a control valve V23 provided in the pipe L23, a control valve V24 provided in the pipe L24, and a control part 210 that controls the control valve V23 and the control valve V24. With the refrigerator 100 according to this example, the control part 210 controls the opening and closing operations of the control valve V23 and the control valve V24, so that the place to which the refrigerant having cooled down the target part is supplied can be switched.

[0121] In the refrigerator 100, the control part 210 can execute a first operating mode in which the control valve V23 is open and the control valve V24 is closed, and execute a second operating mode in which the control valve V23 is closed and the control valve V24 is open.

[0122] With the refrigerator 100 according to this example, by choosing the first operating mode, the refrigerant having cooled down the target part can be brought back to the pipe L11, which is a suction pipe for the compressor 10. By this means, the degree of superheat of the refrigerant gas that is supplied to the compressor 10 can be made relatively high. Also, with the refrigerator 100 according to this example, by choosing the second operating mode, the refrigerant having cooled down the target part can be supplied to the evaporator 40 of the second position. With the refrigerator 100 according to this example, the degree of superheat of the refrigerant gas that is supplied to the compressor 10 can be changed by choosing the first operating mode or the second operating mode, so that occurrence of surging can be prevented or substantially prevented. The refrigerator 100 can thus the expand the adjustable range of operation.

[0123] In the first operating mode, in which the refrigerant having cooled down the target part is brought back to the pipe L11, which is a suction pipe for the compressor 10, the compressing force required by the compressor 10 increases as the degree of superheat of the refrigerant increases. Consequently, limitations or constraints arise with regard to the motor's outputs at operating points where the compressor is required to exert large force. In order to avoid having such limitations or constraints on the motor's outputs, the compressor 10 is enabled to continue operating by switching from the first operating mode to the second operating mode. As a result of this, the range in which the compressor 10 can operate can be expanded.

[0124] In this example, the compressor 10 switches from the second operating mode to the first operating mode, thereby avoiding having the number of rotations of the compressor 10 per unit time limited or constrained and ensuring sufficient refrigeration capacity while maintaining the number of rotations of the compressor 10 per unit time. The refrigerator 100 can switch from the first operating mode to the second operating mode or from the second operating mode to the first operating mode, depending on the operating status, so that the range in which the compressor 10 can operate can be expanded.

[0125] The refrigerator 100 according to the embodiment may include a control part 210 that controls the flow rate of the refrigerant in a third pipe, namely a pipe L24, based on a surge line SL1 that indicates the boundary of the range in which the compressor 10 has a surge. The refrigerator 100 according to this example controls the flow rate of the refrigerant that flows in the pipe L24 based on the surge line SL1, thereby preventing or substantially preventing the compressor 10 from having a surge.

[0126] In the refrigerator 100 according to the embodiment, the control part 210 controls the flow rate of the refrigerant that flows in the pipe L24, based on an operational range boundary SL2 that indicates the range of operation on the side where the refrigerant flow rate is high with respect to the surge line SL1, and that takes into account a margin SR for the surge line SL1. With the refrigerator 100 according to this example, the flow rate of the refrigerant that flows in the pipe L24 is controlled based on the operational range boundary SL2, which takes into account a margin SR relative to the surge line SL1, thereby reliably preventing or substantially preventing the compressor 10 from having a surge.

[0127] In the refrigerator 100 according to the embodiment, the second position is an evaporator, and the control part 210 predicts the operating point of the compressor 10 at a second point in time that is a predetermined period of time after the current first point in time, and, if the predicted operating point is in a range closer to the surge line SL1 than to the operational range boundary SL2, or in a range in which surging occurs, the flow rate of the refrigerant that flows in the pipe L24 can be increased. With the refrigerator 100 according to this example, the operating point of the compressor 10 at a second point in time that comes later than the present point in time can be predicted, and occurrence of surging can be predicted. With the refrigerator of this example, when a surge is likely to occur, the flow rate of the refrigerant that flows in the pipe L24 can be increased to prevent or substantially prevent occurrence of surging.

[0128] With the refrigerator 100 of this embodiment, the refrigerant gas is returned to the pipe L11 on the intake side of the compressor 10, but this can be changed such that the refrigerant gas is returned to the evaporator 40. Given this type of the refrigerator 100, the increase of pressure by the compressor 10 can be increased even when the same number of rotations per unit time and the same volumetric flow rate are maintained. Also, with the refrigerator 100, the inclination of the isentropic line becomes steeper due to the decrease in the degree of superheat, and the ultimate pressure can be increased even when the same adiabatic head is maintained.

[0129] Also, with refrigerator 100, the refrigeration capacity is reduced by switching the place to which the refrigerant gas is returned to the evaporator 40. With this refrigerator 100, the refrigeration capacity is reduced because part of the enthalpy change inside the evaporator 40 cannot be used as refrigeration capacity. Therefore, with the refrigerator 100, the refrigeration capacity can be reduced without reducing the flow rate of the compressor 10.

[0130] A preferred embodiment of the present disclosure has been described above in detail. However, the present disclosure is by no means limited to the embodiment described hereinabove. The embodiment may be modified, substituted, and so forth, without departing from the scope of the present disclosure. Also, features of the present invention described above separately may be combined unless a technical contradiction arises.

[0131] According to the above embodiment, the compressor 10 may be a single-stage compressor or a multistage compressor with multiple impellers 12. For example, in a multi-stage compressor, the refrigerant having cooled down the target part may be supplied to a subsequent impeller (compression mechanism).

[0132] According to the above embodiment, the refrigerant having cooled down the target part is supplied to the evaporator 40 of the second position, but the refrigerant may be supplied to any position in the evaporator 40, and the refrigerant having cooled down the target part may be supplied to the upstream pipe L14 connected to the evaporator 40. Also, the second position may be multiple locations. For example, in the refrigerator 100, the refrigerant having cooled down the target part may be supplied to multiple locations that are determined with respect to the evaporator 40.

[0133] Examples of the present disclosure may include the following:

[1] A refrigerator for running a refrigeration cycle, the refrigerator including:

a compressor configured to compress a refrigerant;
a condenser configured to condense the compressed refrigerant;
an expansion valve configured to expand the condensed refrigerant;
an evaporator configured to evaporate the expanded refrigerant;
a suction pipe configured to connect the evaporator and the compressor;
a motor configured to drive the compressor;
a first pipe configured to supply the refrigerant discharged from the condenser to a target part;
a second pipe configured to supply the refrigerant having cooled down the target part to the suction pipe, the suction pipe serving as a first position; and
a third pipe configured to supply the refrigerant having cooled down the target part to a second position, the second position being different from the first position and located inside the refrigerator.

[2] The refrigerator according to [1], in which the compressor is a turbo compressor.

[3] The refrigerator according to [1] or [2], in which the target part is the motor, a bearing configured to support a rotary shaft of the compressor, or a controller for the motor.

[4] The refrigerator according to any one of [1] to [3], in which the evaporator serves as the second position.

[5] The refrigerator according to any one of [1] to [4],

in which the target part is the motor, and
in which the first pipe is connected to a flow path that communicates with inside of a casing of the motor.

[6] The refrigerator according to any one of [1] to [5],

in which the target part is a bearing configured to support a rotary shaft of the compressor, and
in which the bearing is an oil-less bearing.

[7] The refrigerator according to any one of [1] to [6], in which the evaporator is an air-heat exchanger configured to transfer heat between the refrigerant and air.

[8] The refrigerator according to any one of [1] to [7], further including:

a first control valve provided in the second pipe;
a second control valve provided in the third pipe; and
a control part configured to control the first control valve and the second control valve.

[9] The refrigerator according to [8],

in which the control part is capable of executing a first operating mode in which the first control valve is open and the second control valve is closed, and
in which the control part is capable of executing a second operating mode in which the first control valve is closed and the second control valve is open.

[10] The refrigerator according to any one of [1] to [9], further including a control part configured to control a flow rate of the refrigerant in the third pipe based on a surge line that indicates a boundary of a range in which the compressor has a surge.
[11] The refrigerator according to [10], in which the control part is configured to control the flow rate of the refrigerant in the third pipe based on an operational range boundary, the operational range boundary marking an operating range on a side of the surge line where the flow rate of the refrigerant is larger and taking into account a margin with respect to the surge line.
[12] The refrigerator according to [11],

in which the evaporator is the second position,
in which the control part is configured to predict an operating point of the compressor at a second point in time that is a predetermined period of time after a current first point in time, and
in which the control part is configured to increase the flow rate of the refrigerant in the third pipe when the predicted operating point is positioned closer to the surge line than to the operational range boundary or positioned in a range in which the surge occurs.

[0134] This international application claims priority based on Japanese Patent Application No. 2023-058725, filed on March 31, 2023, the entire contents of which are incorporated herein by reference.

Reference Signs List

[0135]

100	refrigerator
10	compressor
13	drive shaft (rotary shaft)
14	bearing (target part, oil-less bearing)
20	condenser
30	expansion valve
40	evaporator (second position, air-heat exchanger)
50	motor (target part)
60	inverter (motor's controller)
210	control part
L11	pipe (suction pipe, first position)
L21	pipe (first pipe)
L23	pipe (second pipe)
L24	pipe (third pipe)
V23	control valve (first control valve)
V24	control valve (second control valve)

Claims

1. A refrigerator for running a refrigeration cycle, the refrigerator comprising:

a compressor configured to compress a refrigerant;
a condenser configured to condense the compressed refrigerant;
an expansion valve configured to expand the condensed refrigerant;
an evaporator configured to evaporate the expanded refrigerant;
a suction pipe configured to connect the evaporator and the compressor;
a motor configured to drive the compressor;
a first pipe configured to supply the refrigerant discharged from the condenser to a target part;
a second pipe configured to supply the refrigerant having cooled down the target part to the suction pipe, the suction pipe serving as a first position; and
a third pipe configured to supply the refrigerant having cooled down the target part to a second position, the second position being different from the first position and located inside the refrigerator.

2. The refrigerator according to claim 1, wherein the compressor is a turbo compressor.

3. The refrigerator according to claim 1 or 2, wherein the target part is the motor, a bearing configured to support a rotary shaft of the compressor, or a controller for the motor.

4. The refrigerator according to any one of claims 1 to 3, wherein the evaporator serves as the second position.

5. The refrigerator according to any one of claims 1 to 4,

wherein the target part is the motor, and
wherein the first pipe is connected to a flow path that communicates with inside of a casing of the motor.

6. The refrigerator according to any one of claims 1 to 5,

wherein the target part is a bearing configured to support a rotary shaft of the compressor, and
wherein the bearing is an oil-less bearing.

7. The refrigerator according to any one of claims 1 to 6, wherein the evaporator is an air-heat exchanger configured to transfer heat between the refrigerant and air.

8. The refrigerator according to any one of claims 1 to 7, further comprising:

a first control valve provided in the second pipe;
a second control valve provided in the third pipe; and
a control part configured to control the first control valve and the second control valve.

9. The refrigerator according to claim 8,

wherein the control part is capable of executing a first operating mode in which the first control valve is open and the second control valve is closed, and
wherein the control part is capable of executing a second operating mode in which the first control valve is closed and the second control valve is open.

10. The refrigerator according to any one of claims 1 to 9, further comprising a control part configured to control a flow rate of the refrigerant in the third pipe based on a surge line that indicates a boundary of a range in which the compressor has a surge.

11. The refrigerator according to claim 10, wherein the control part is configured to control the flow rate of the refrigerant in the third pipe based on an operational range boundary, the operational range boundary marking an operating range on a side of the surge line where the flow rate of the refrigerant is larger and taking into account a margin with respect to the surge line.

12. The refrigerator according to claim 11,

wherein the evaporator is the second position,
wherein the control part is configured to predict an operating point of the compressor at a second point in time that is a predetermined period of time after a current first point in time, and
wherein the control part is configured to increase the flow rate of the refrigerant in the third pipe when the predicted operating point is positioned closer to the surge line than to the operational range boundary or positioned in a range in which the surge occurs.

FIG.1

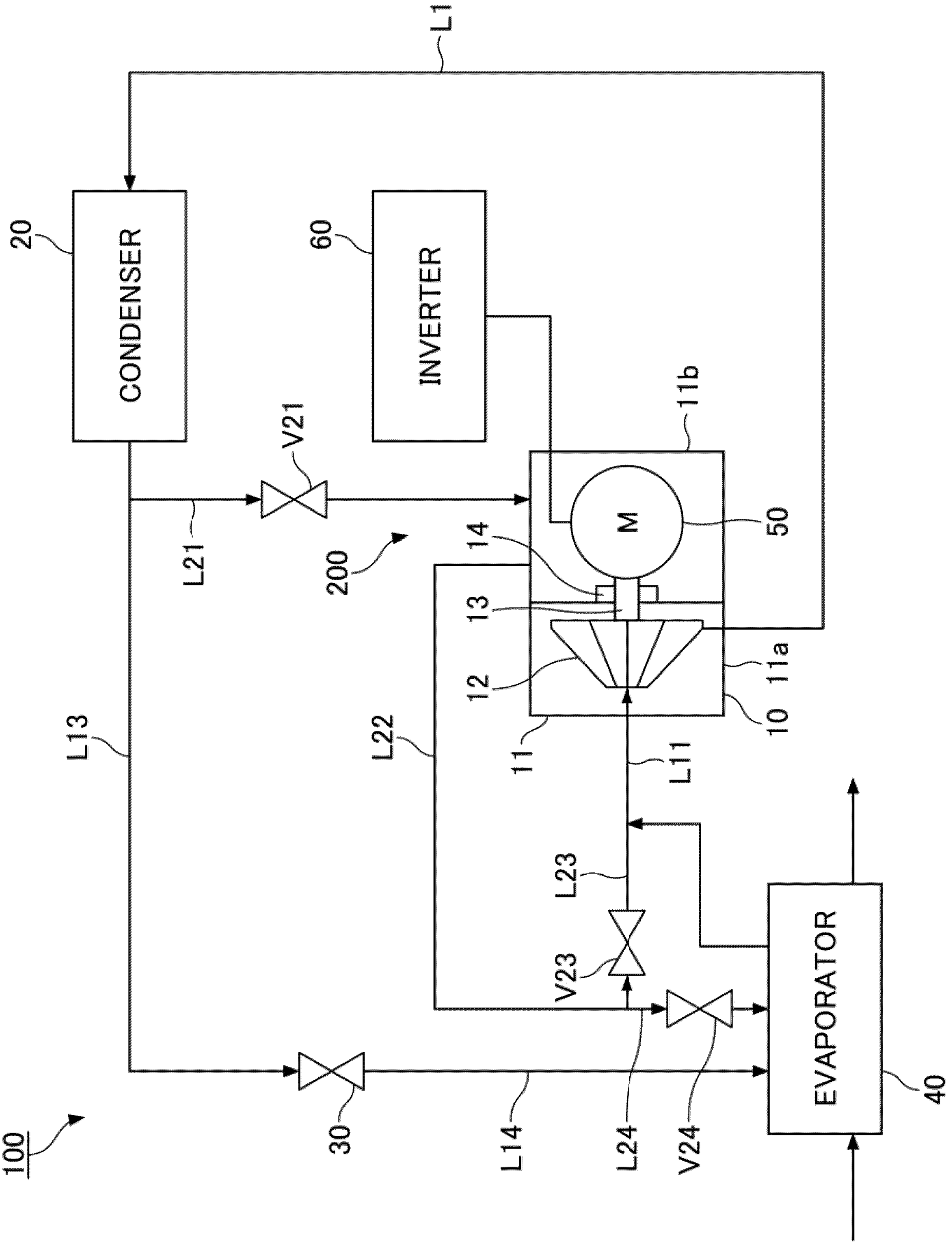

L12
20
CONDENSER
60
INVERTER
V21
11b
L21
200
M
50
14
13
12
11a
10
11
L13
L22
L11
L23
V23
EVAPORATOR
40
30
L14
L24
V24
100

FIG.2

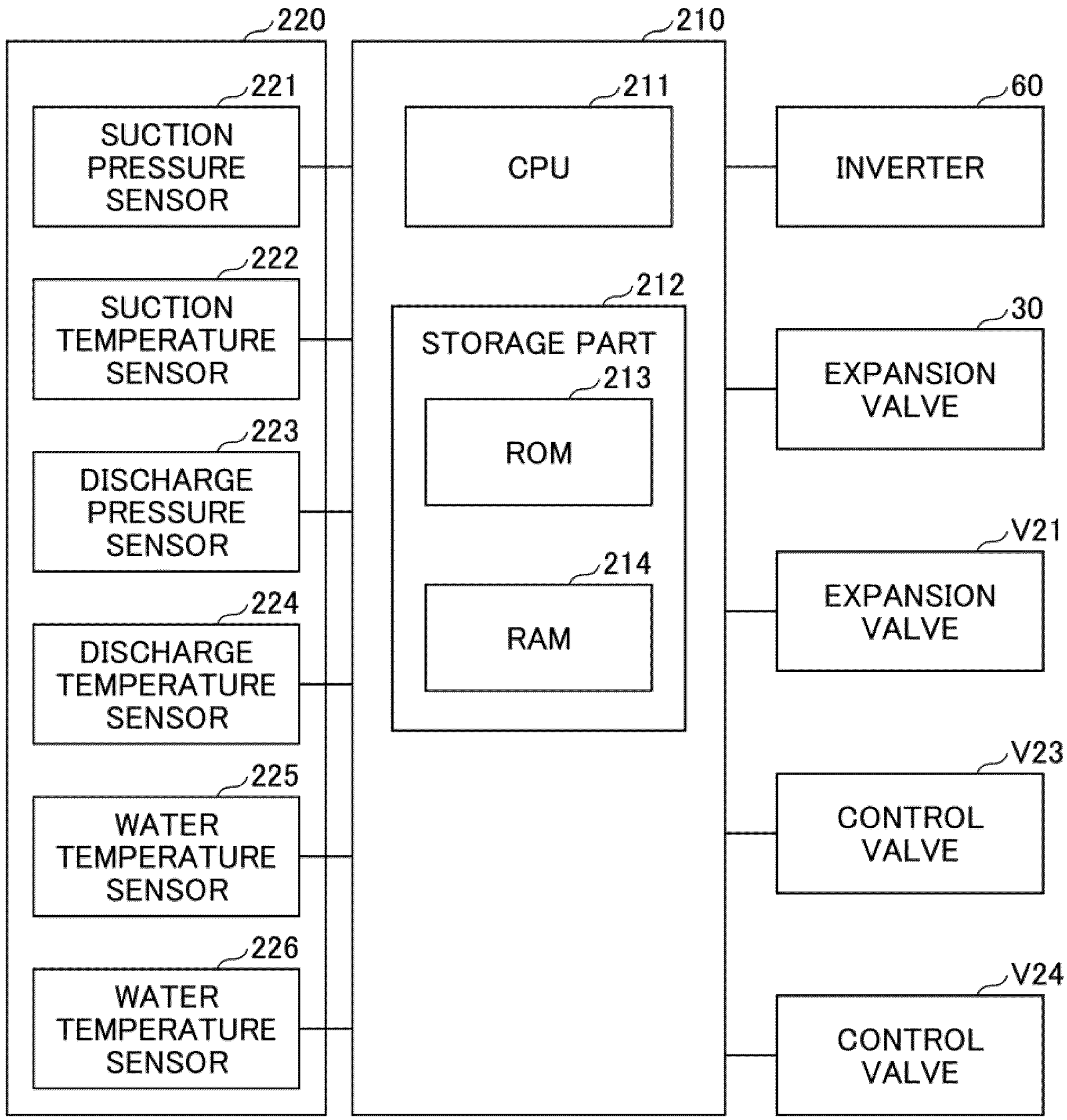

220
221
SUCTION PRESSURE SENSOR
222
SUCTION TEMPERATURE SENSOR
223
DISCHARGE PRESSURE SENSOR
224
DISCHARGE TEMPERATURE SENSOR
225
WATER TEMPERATURE SENSOR
226
WATER TEMPERATURE SENSOR
210
211
CPU
212
STORAGE PART
213
ROM
214
RAM
60
INVERTER
30
EXPANSION VALVE
V21
EXPANSION VALVE
V23
CONTROL VALVE
V24
CONTROL VALVE

FIG.3

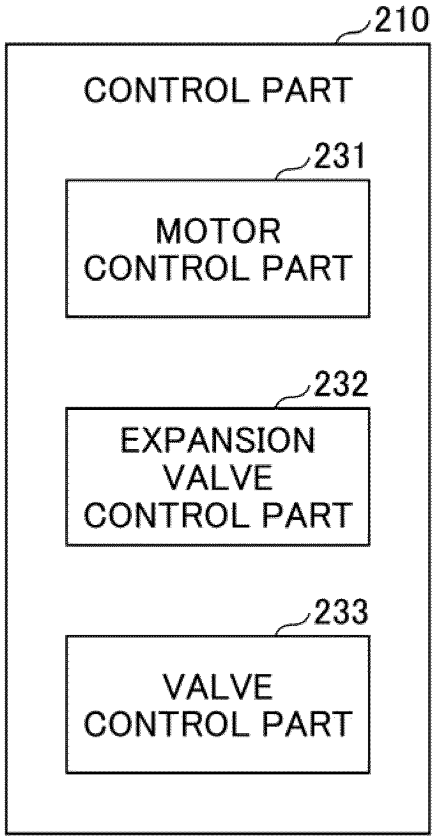

210
CONTROL PART
231
MOTOR
CONTROL PART
232
EXPANSION
VALVE
CONTROL PART
233
VALVE
CONTROL PART

FIG.4

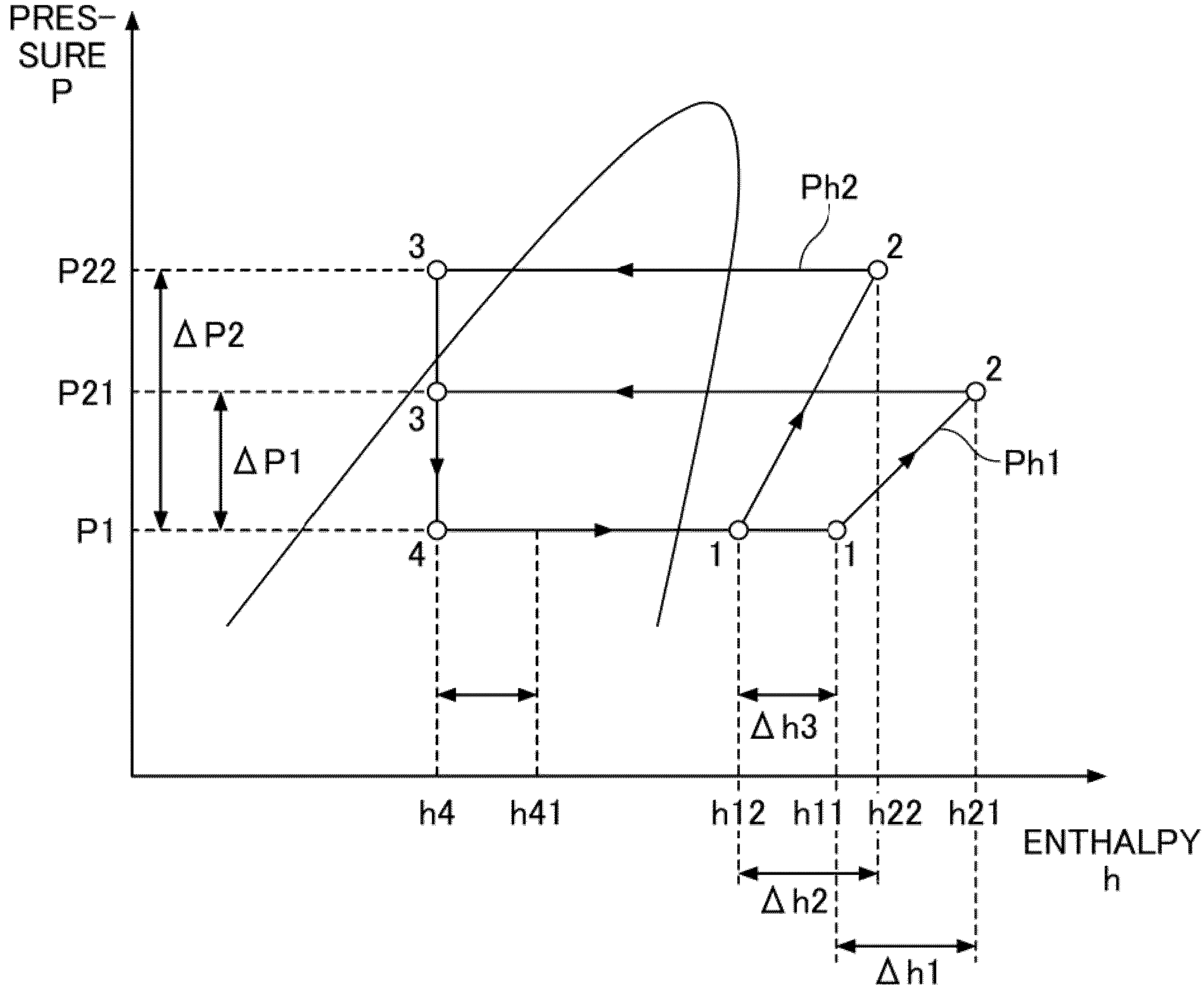

PRES-
SURE
P
P22
P21
P1
ΔP2
ΔP1
3
3
4
2
2
1
1
Ph2
Ph1
Δh3
Δh2
Δh1
h4
h41
h12
h11
h22
h21
ENTHALPY
h

FIG.5

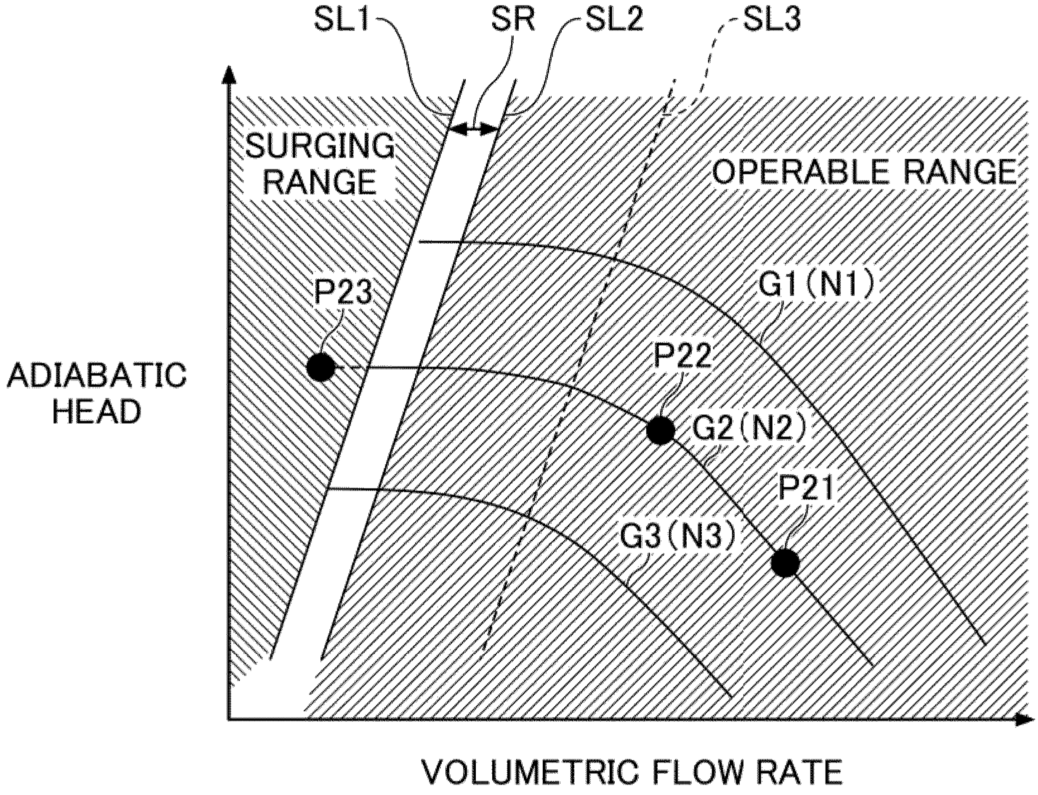

SL1
SR
SL2
SL3
SURGING RANGE
OPERABLE RANGE
P23
G1 (N1)
P22
ADIABATIC HEAD
G2 (N2)
P21
G3 (N3)
VOLUMETRIC FLOW RATE

FIG.6

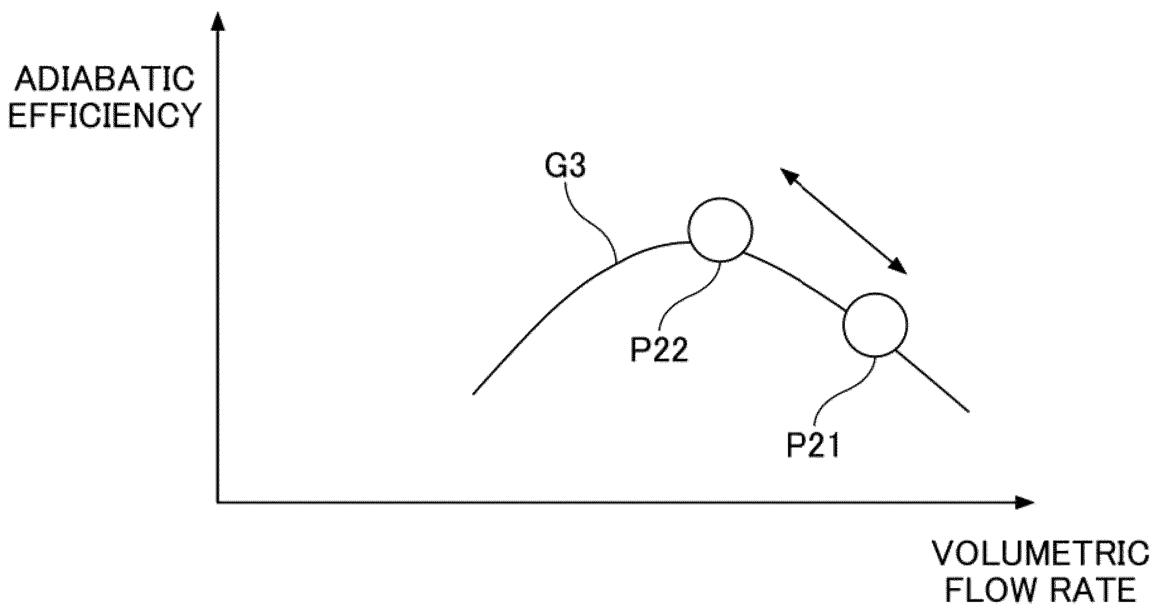
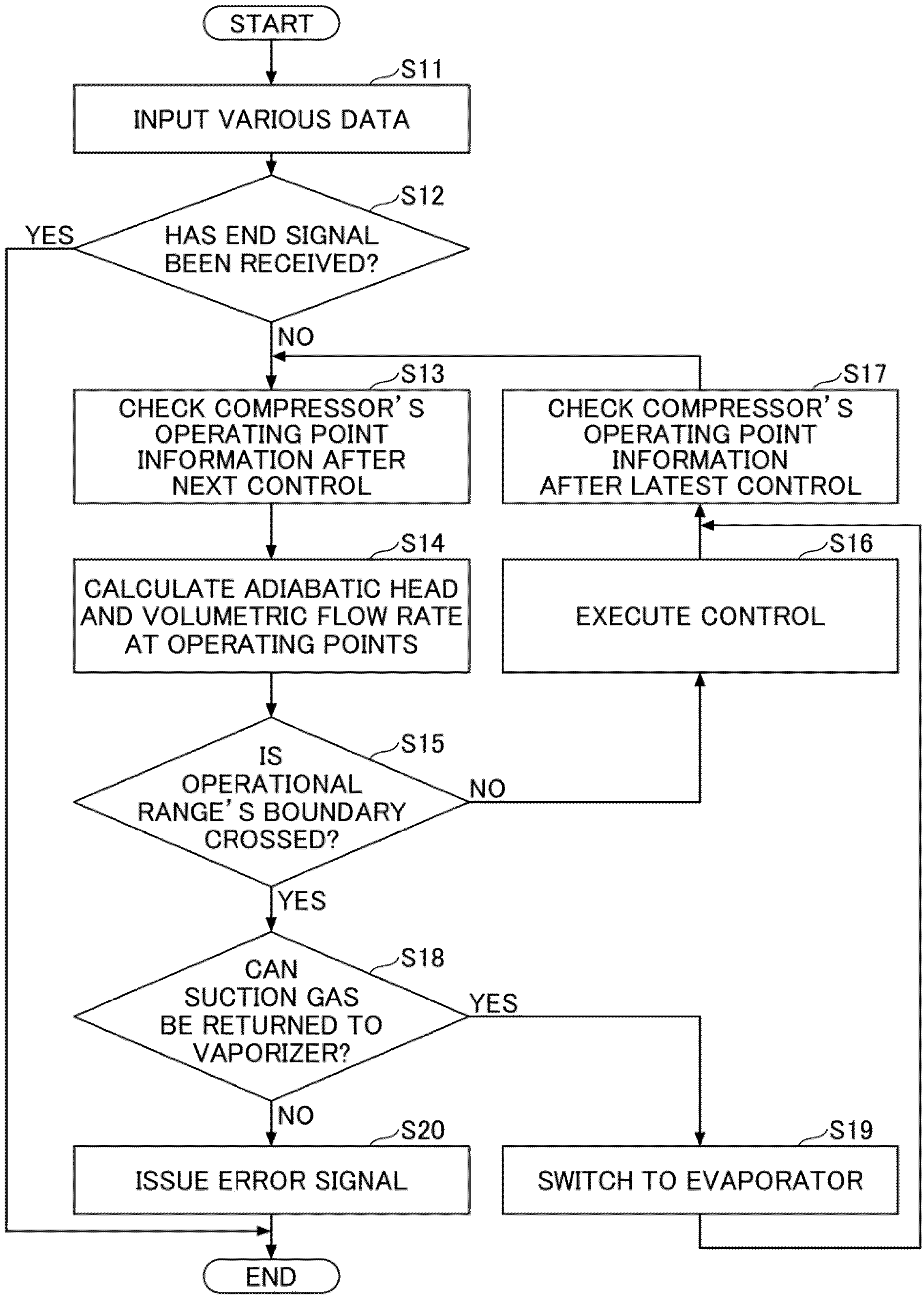

ADIABATIC EFFICIENCY
G3
P22
P21
VOLUMETRIC FLOW RATE

FIG.7


START
S11
INPUT VARIOUS DATA
S12
HAS END SIGNAL BEEN RECEIVED?
YES
NO
S13
CHECK COMPRESSOR'S OPERATING POINT INFORMATION AFTER NEXT CONTROL
S17
CHECK COMPRESSOR'S OPERATING POINT INFORMATION AFTER LATEST CONTROL
S14
CALCULATE ADIABATIC HEAD AND VOLUMETRIC FLOW RATE AT OPERATING POINTS
S16
EXECUTE CONTROL
S15
IS OPERATIONAL RANGE'S BOUNDARY CROSSED?
NO
YES
S18
CAN SUCTION GAS BE RETURNED TO VAPORIZER?
YES
NO
S20
ISSUE ERROR SIGNAL
S19
SWITCH TO EVAPORATOR
END

INTERNATIONAL SEARCH REPORT

International application No. **PCT/JP2024/013147**

A. CLASSIFICATION OF SUBJECT MATTER

F25B 1/053(2006.01)i; ***F25B 1/00***(2006.01)i
FI: F25B1/053 D; F25B1/00 321N; F25B1/00 321Z

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

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

F25B1/053; F25B1/00

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 1971-2024
Registered utility model specifications of Japan 1996-2024
Published registered utility model applications of Japan 1994-2024

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	US 2015/0226467 A1 (JOHNSON CONTROLS TECHNOLGY COMPANY) 13 August 2015 (2015-08-13) paragraphs [0009]-[0034], fig. 2-5	1-7
A	paragraphs [0009]-[0034], fig. 2-5	8-12
Y	Microfilm of the specification and drawings annexed to the request of Japanese Utility Model Application No. 079522/1989 (Laid-open No. 019499/1991) (THE TOKYO ELECTRIC POWER CO., INC.) 26 February 1991 (1991-02-26), specification, page 5, line 12 to page 6, line 13, fig. 1	1-7
Y	JP 2006-194579 A (SAMSUNG ELECTRONICS CO., LTD.) 27 July 2006 (2006-07-27) paragraph [0025], fig. 1-2	1-7
A	JP 49-043104 A (LITTON INDUSTRIES, INCORPORATED.) 23 April 1974 (1974-04-23) entire text, all drawings	1-12

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

* Special categories of cited documents:
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"E" earlier application or patent but published on or after the international filing date
"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)
"O" document referring to an oral disclosure, use, exhibition or other means
"P" document published prior to the international filing date but later than the priority date claimed
"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
10 June 2024	**18 June 2024**

Name and mailing address of the ISA/JP	Authorized officer
Japan Patent Office (ISA/JP) **3-4-3 Kasumigaseki, Chiyoda-ku, Tokyo 100-8915** **Japan**	Telephone No.

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INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2024/013147

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	JP 2021-156220 A (MITSUBISHI HEAVY INDUSTRIES THERMAL SYSTEMS, LTD.) 07 October 2021 (2021-10-07) entire text, all drawings	1-12
A	JP 2023-013514 A (DAIKIN INDUSTRIES, LTD.) 26 January 2023 (2023-01-26) entire text, all drawings	1-12

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INTERNATIONAL SEARCH REPORT
Information on patent family members

International application No.
PCT/JP2024/013147

Patent document cited in search report	Publication date (day/month/year)	Patent family member(s)	Publication date (day/month/year)
US 2015/0226467 A1	13 August 2015	WO 2014/084989 A2 EP 3022833 A2	
JP 03-019499 U1	26 February 1991	(Family: none)	
JP 2006-194579 A	27 July 2006	US 2006/0150668 A1 paragraph [0031], fig. 1-2 KR 10-2006-0081791 A	
JP 49-043104 A	23 April 1974	US 3805101 A GB 1432402 A DE 2331493 A1	
JP 2021-156220 A	07 October 2021	(Family: none)	
JP 2023-013514 A	26 January 2023	(Family: none)	

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REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP H1113697 B **[0003]**
- JP 2023058725 A **[0134]**