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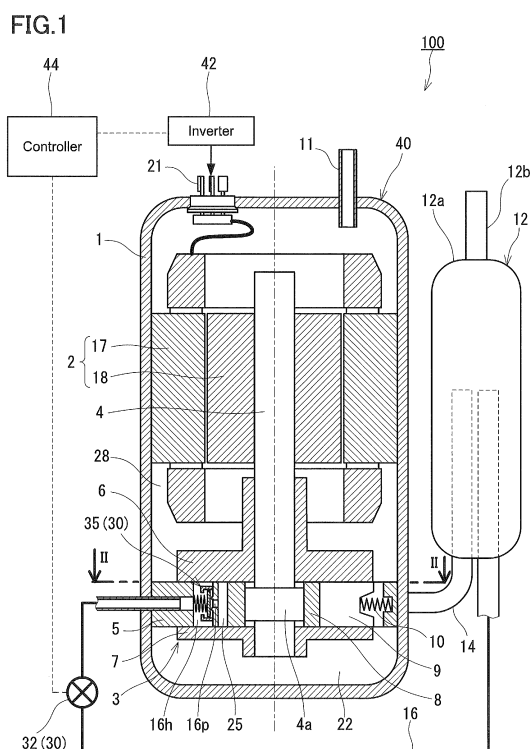
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(54) **POSITIVE DISPLACEMENT COMPRESSOR**

(57) A rotary compressor 100 includes a compression mechanism 3, a motor 2, a suction path 14, a return path 16, a volume varying mechanism 30, an inverter 42, and a controller 44. The return path 16 serves to return a working fluid from the working chamber 25 to the suction path 14. The volume varying mechanism 30 is provided in the return path 16, permits the working fluid to return from the working chamber 25 to the suction path 14 through the return path 16 when the suction volume of the compression mechanism 3 should be set to a relatively small value, and prohibits the working fluid from returning from the working chamber 25 to the suction path 14 through the return path 16 when the suction volume should be set to a relatively large value. The volume varying mechanism 30 and the inverter 42 are controlled so as to compensate for a decrease in the suction volume with an increase in the rotational speed of the motor 2.



Description

TECHNICAL FIELD

[0001] The present invention relates to positive displacement compressors.

BACKGROUND ART

[0002] A motor of a compressor is usually controlled by an inverter and a microcomputer. If the rotational speed of the motor is decreased, a refrigeration cycle apparatus in which the compressor is used can be operated with a power sufficiently lower than a rated value. In addition, Patent Literature 1 provides a technique for operating the refrigeration cycle apparatus with such a low power as cannot be realized by inverter control.

[0003] FIG. 19 is a configuration diagram of an air conditioner described in Patent Literature 1. A refrigeration cycle is constituted by a compressor 915, a four-way valve 917, an indoor heat exchanger 918, a pressure reducing device 919, and an outdoor heat exchanger 920. A cylinder of the compressor 915 is provided with an intermediate discharge port that opens from the start of a compression process to some point in the process. The intermediate discharge port is connected to a suction path of the compressor 915 via a bypass path 923. The bypass path 923 is provided with a flow rate control device 921 and a solenoid on-off valve 922. The solenoid on-off valve 922 is opened only in operation at a low set frequency. This allows operation with a lower power.

CITATION LIST

Patent Literature

[0004] PTL1: JP S61(1986)-184365 A

SUMMARY OF INVENTION

Technical Problem

[0005] An easy way to improve the efficiency of a refrigeration cycle apparatus is to improve the efficiency of a compressor. The efficiency of the compressor largely depends on the efficiency of a motor used in the compressor. Many motors are designed to exhibit the highest efficiency at a rotational speed close to a rated rotational speed (e.g., 60 Hz). Therefore, if the motor is driven at an extremely low rotational speed, improvement in the efficiency of the compressor cannot be expected.

[0006] In view of such circumstances, the present invention aims to provide a positive displacement compressor that can exhibit high efficiency even when a low power is required (even when a load is small).

Solution to Problem

[0007] That is, the present invention provides a positive displacement compressor including:

a compression mechanism having a working chamber;
a motor that moves the compression mechanism;
a suction path that guides a working fluid to be compressed to the working chamber;
a return path that returns the working fluid from the working chamber to the suction path;
a volume varying mechanism that is provided in the return path, permits the working fluid to return from the working chamber to the suction path through the return path when a suction volume of the compression mechanism should be set to a relatively small value, and prohibits the working fluid from returning from the working chamber to the suction path through the return path when the suction volume should be set to a relatively large value;
an inverter that drives the motor; and
a controller that controls the volume varying mechanism and the inverter so as to compensate for a decrease in the suction volume with an increase in a rotational speed of the motor.

Advantageous Effects of Invention

[0008] According to the present invention, the positive displacement compressor can be operated with a relatively small suction volume by returning the working fluid from the working chamber to the suction path by means of the return path. On the other hand, if the working fluid is prohibited from returning from the working chamber to the suction path, the positive displacement compressor can be operated with a relatively large suction volume, that is, with a normal suction volume. Furthermore, according to the present invention, the volume varying mechanism and the inverter are controlled so as to compensate for a decrease in the suction volume with an increase in the rotational speed of the motor. That is, the motor is not driven at a low rotational speed, but instead, the suction volume is decreased. Therefore, it is possible to provide a positive displacement compressor that can exhibit high efficiency even when a load is small.

BRIEF DESCRIPTION OF DRAWINGS

[0009]

FIG. 1 is a longitudinal cross-sectional view of a rotary compressor according to a first embodiment of the present invention.

FIG. 2 is a transverse cross-sectional view of the rotary compressor of FIG. 1 taken along a II-II line. FIG. 3 is a diagram illustrating the operation principle of the rotary compressor of FIG. 1.

FIG. 4A is a graph showing the relationship between the rotational angle of a shaft and the volume of a suction chamber.

FIG. 4B is a graph showing the relationship between the rotational angle of the shaft and the volume of a compression-discharge chamber.

FIG. 5A is a flowchart illustrating control of a volume varying mechanism (on-off valve) and an inverter.

FIG. 5B is another flowchart illustrating control of the volume varying mechanism (on-off valve) and the inverter.

FIG. 6 is a graph showing the relationship among the power of the rotary compressor, the suction volume of a compression mechanism, the state of the on-off valve, and the rotational speed of a motor.

FIG. 7 is a graph showing the relationship between the power of the rotary compressor and the efficiency of the rotary compressor.

FIG. 8A is a graph showing the relationship between the rotational angle of the shaft and the flow velocity of a refrigerant in a suction path.

FIG. 8B is a graph showing the relationship between the rotational angle of the shaft and the flow velocity of the refrigerant in a return path.

FIG. 8C is a graph showing the relationship between the rotational angle of the shaft and the flow velocity of the refrigerant in an introduction pipe of an accumulator.

FIG. 9 is a longitudinal cross-sectional view of a rotary compressor according to a second embodiment.

FIG. 10 is a transverse cross-sectional view of the rotary compressor of FIG. 9 taken along a X-X line.

FIG. 11 is a transverse cross-sectional view showing an example of modification of a position at which a return path and a first working chamber are connected.

FIG. 12 is a longitudinal cross-sectional view of a rotary compressor according to a third embodiment.

FIG. 13 is a longitudinal cross-sectional view of a rotary compressor according to a fourth embodiment.

FIG. 14 is a longitudinal cross-sectional view of a rotary compressor according to a fifth embodiment.

FIG. 15A is a partially enlarged cross-sectional view of the rotary compressor of FIG. 14 in a low volume mode.

FIG. 15B is a partially enlarged cross-sectional view of the rotary compressor of FIG. 14 in a high volume mode.

FIG. 16 is a longitudinal cross-sectional view of a scroll compressor according to a sixth embodiment.

FIG. 17 is a longitudinal cross-sectional view of a scroll compressor according to a seventh embodiment.

FIG. 18 is a configuration diagram of a refrigeration cycle apparatus in which the rotary compressor of the present embodiment is used.

FIG. 19 is a configuration diagram of a conventional

air conditioner.

DESCRIPTION OF EMBODIMENTS

[0010] Hereinafter, several embodiments of the present invention will be described with reference to the drawings. The present invention is not limited by the embodiments described below. The type of the positive displacement compressor is not particularly limited. Examples of the positive displacement compressor include rotary compressors, scroll compressors, reciprocating compressors, screw compressors, and swash plate compressors. In the present specification, embodiments of a rotary compressor and a scroll compressor are described.

(First embodiment)

[0011] As shown in FIG. 1, a rotary compressor 100 of the present embodiment includes a compressor body 40, an accumulator 12, a discharge path 11, a suction path 14, a return path 16, a volume varying mechanism 30, an inverter 42, and a controller 44.

[0012] The compressor body 40 includes a closed casing 1, a motor 2, a rotary compression mechanism 3, and a shaft 4. The compression mechanism 3 is located in a lower portion of the closed casing 1. The motor 2 is located above the compression mechanism 3 in the closed casing 1. The compression mechanism 3 and the motor 2 are coupled by the shaft 4. A terminal 21 for supplying electric power to the motor 2 is provided at the top of the closed casing 1. An oil reservoir 22 for retaining a lubricating oil is formed in a bottom portion of the closed casing 1. The compressor body 40 has a structure of a so-called hermetic compressor.

[0013] The discharge path 11, the suction path 14, and the return path 16 are each formed by a refrigerant pipe. The discharge path 11 penetrates through the top of the closed casing 1, and opens inside the closed casing 1. The discharge path 11 serves to guide a working fluid (typically, a refrigerant) having been compressed to the outside of the compressor body 40. The suction path 14 has one end connected to the compression mechanism 3 and the other end connected to the accumulator 12, and penetrates through a trunk portion of the closed casing 1. The suction path 14 serves to guide the refrigerant to be compressed from the accumulator 12 to a working chamber 25 of the compression mechanism 3. The return path 16 has one end connected to the compression mechanism 3 at a position different from a position at which the suction path 14 is connected to the compression mechanism 3, and also has the other end connected to the accumulator 12. The return path 16 penetrates through the trunk portion of the closed casing 1. The return path 16 serves to return, to the suction path 14, the refrigerant that has been drawn into the working chamber 25 of the compression mechanism 3 but has not been compressed yet.

[0014] The compression mechanism 3 is a positive displacement fluid mechanism, and is moved by the motor 2 so as to compress the refrigerant. As shown in FIG. 1 and FIG. 2, the compression mechanism 3 is composed of a cylinder 5, a piston 8, a vane 9, a spring 10, an upper bearing 6, and a lower bearing 7. The piston 8 fitted to an eccentric portion 4a of the shaft 4 is disposed inside the cylinder 5 so as to form the working chamber 25 between the outer circumferential surface of the piston 8 and the inner circumferential surface of the cylinder 5. A vane groove 24 is formed in the cylinder 5. The vane 9 having one end contacting the outer circumferential surface of the piston 8 is placed in the vane groove 24. The spring 10 is disposed in the vane groove 24 so as to push the vane 9 toward the piston 8. The upper bearing 6 and the lower bearing 7 are respectively provided on and under the cylinder 5 so as to close the cylinder 5. The working chamber 25 between the cylinder 5 and the piston 8 is divided by the vane 9, and thus a suction chamber 25a and a compression-discharge chamber 25b are formed. The refrigerant to be compressed is guided to the working chamber 25 (the suction chamber 25a) through the suction path 14 and a suction port 27. A discharge port 29 is formed in the upper bearing 6 so as to guide the compressed refrigerant from the working chamber 25 (the compression-discharge chamber 25b) to an internal space 28 of the closed casing 1. The discharge port 29 is provided with a discharge valve which is not shown in the drawings. The vane 9 may be integrated with the piston 8. That is, the piston 8 and the vane 9 may be formed as a swing piston.

[0015] The motor 2 is composed of a stator 17 and a rotor 18. The stator 17 is fixed to the inner circumferential surface of the closed casing 1. The rotor 18 is fixed to the shaft 4, and rotates together with the shaft 4. The piston 8 is moved inside the cylinder 5 by the motor 2. A motor whose rotational speed is variable, such as an IPMSM (Interior Permanent Magnet Synchronous Motor) and a SPMSM (Surface Permanent Magnet Synchronous Motor), can be used as the motor 2.

[0016] The controller 44 controls the inverter 42 to adjust the rotational speed of the motor 2, that is, the rotational speed of the rotary compressor 100. A DSP (Digital Signal Processor) including an A/D conversion circuit, an input/output circuit, an arithmetic circuit, a storage device, etc., can be used as the controller 44.

[0017] The accumulator 12 is composed of an accumulation container 12a and an introduction pipe 12b. The accumulation container 12a has an internal space capable of retaining the liquid refrigerant and the gaseous refrigerant. The introduction pipe 12b penetrates through the top of the accumulation container 12a, and opens toward the internal space of the accumulation container 12a. The suction path 14 and the return path 16 are each connected to the accumulator 12 in such a manner as to penetrate through the bottom of the accumulation container 12a. The suction path 14 and the return path 16 extend upward from the bottom of the accumulation con-

tainer 12a, and open toward the internal space of the accumulation container 12a at a certain height. That is, the return path 16 is connected to the suction path 14 via the internal space of the accumulator 12. Another member such as a baffle may be provided inside the accumulation container 12a in order to ensure that the liquid refrigerant is prevented from entering the suction path 14 directly from the introduction pipe 12b.

[0018] The volume varying mechanism 30 is provided in the return path 16. In the present embodiment, the volume varying mechanism 30 is composed of an on-off valve 32 and a check valve 35. That is, in the present embodiment, the volume varying mechanism 30 has no ability to reduce the pressure of the refrigerant. In addition, the refrigerant having been drawn into the suction chamber 25a can be returned to the suction path 14 through the return path 16, substantially without being compressed in the compression-discharge chamber 25b. Therefore, reduction of the efficiency due to pressure loss is very small. However, the volume varying mechanism 30 may have the ability to reduce the pressure of the refrigerant to the extent that large influence is not exerted on the efficiency of the rotary compressor 100. For the same reason, the refrigerant having been compressed in the compression-discharge chamber 25b may be returned to the suction path 14 through the return path 16.

[0019] The on-off valve 32 is located outside the compressor body 40, and provided in the return path 16. On the other hand, the check valve 35 is provided inside the compressor body 40. As shown in FIG. 1 and FIG. 2, the return path 16 includes an upstream portion 16h formed inside the compression mechanism 3 (in particular, inside the cylinder 5), and a return port 16p that allows communication between the working chamber 25 and the upstream portion 16h. The check valve 35 is provided in the upstream portion 16h. The check valve 35 blocks a flow of the refrigerant from the return path 16 to the working chamber 25. By using the check valve 35, it is possible to block a flow of the refrigerant from the return path 16 to the working chamber 25 with a relatively simple structure, without resorting to electric control.

[0020] As shown in FIG. 2, the check valve 35 is composed of a valve body 36, a guide 37, and a spring 38. The valve body 36 is made of a thin metal plate having two surfaces, and is located inwardly of the guide 37 so as to be able to reciprocate between a first position for closing the return port 16p and a second position for opening the return port 16p. One of the surfaces of the valve body 36 faces the return port 16p, and the other surface faces the spring 38. The spring 38 pushes the valve body 36 toward the return port 16p. A gap with an appropriate width is formed between the valve body 36 and the guide 37. When the valve body 36 is moved away from the return port 16p, in other words, when the valve body 36 occupies the second position, the working chamber 25 communicates with the upstream portion 16h of the return path 16. When the valve body 36 comes into

contact with the return port 16p, in other words, when the valve body 36 occupies the first position, the working chamber 25 is disconnected from the upstream portion 16h of the return path 16.

[0021] The volume varying mechanism 30 serves to vary the suction volume (confined volume) of the rotary compressor 100. When the suction volume of the rotary compressor 100 should be set to a relatively small value, the refrigerant that has not been compressed yet is permitted to return from the working chamber 25 (in particular, the compression-discharge chamber 25b) to the suction path 14 through the return path 16. Specifically, the on-off valve 32 is caused to open. On the other hand, when the suction volume of the rotary compressor 100 should be set to a relatively large value, the refrigerant that has not been compressed yet is prohibited from returning from the working chamber 25 to the suction path 14 through the return path 16. Specifically, the on-off valve 32 is caused to close. While the on-off valve 32 is open, the rotary compressor 100 is operated in a low volume mode. While the on-off valve 32 is closed, the rotary compressor 100 is operated in a high volume mode.

[0022] When the operation mode of the rotary compressor 100 is switched from the high volume mode to the low volume mode by controlling the volume varying mechanism 30, the inverter 42 is controlled so as to compensate for a decrease in the suction volume with an increase in the rotational speed of the motor 2. This can prevent extreme decrease in the rotational speed of the motor 2 even when a low power is required (even when a load is small). That is, even when a low power is required, the motor 2 can be driven at a rotational speed that allows high efficiency to be exhibited. Consequently, the efficiency of the rotary compressor 100 is also improved.

[0023] As shown in FIG. 2, the upstream portion 16h and the return port 16p of the return path 16 are formed at a position corresponding to 180 degrees in terms of the rotation angle of the shaft 4. In the present specification, the position of the vane 9 and the vane groove 24 is defined as a reference position located at "0 degrees" in the rotational direction of the shaft 4. In other words, the rotational angle of the shaft 4 at the moment when the vane 9 is maximally pushed into the vane groove 24 by the piston 8 is defined as "0 degrees".

[0024] In the high volume mode, a process for compressing the refrigerant confined in the compression-discharge chamber 25b (a compression process) starts from the rotational angle of 0 degrees. On the other hand, in the low volume mode, a process for discharging the refrigerant confined in the compression-discharge chamber 25b from the return port 16p is carried out during the period from 0 degrees to 180 degrees, and the compression process starts from the rotational angle of 180 degrees. Therefore, assuming that the suction volume in the high volume mode is V, the suction volume in the low volume mode is V/2. It should be understood that the

position of the return port 16p or the like can be changed as appropriate depending on the rate of change of the suction volume. For example, in the case where the return port 16p is formed at a position corresponding to 90 degrees, the suction volume in the low volume mode is $\{1 + (1/2)^{1/2}\}V/2$.

[0025] Next, the behavior of the compression mechanism 3 will be described with reference to FIG. 3.

[0026] FIG. 3 shows the shaft 4 and the piston 8 which are rotating counterclockwise. The volume of the suction chamber 25a increases with the rotation of the shaft 4. As shown in the upper left of FIG. 3, when the shaft 4 completes one rotation, the volume of the suction chamber 25a becomes maximum. Thereafter, the suction chamber 25a is converted to the compression-discharge chamber 25b. The volume of the compression-discharge chamber 25b decreases with the rotation of the shaft 4. As shown in FIG. 4A and FIG. 4B, while the volume of the suction chamber 25a increases through points A, B, and C, the volume of the compression-discharge chamber 25b decreases through points D, E, and F.

[0027] As shown in the upper right of FIG. 3, in the case where the on-off valve 32 is open, the check valve 35 opens along with decrease in the volume of the compression-discharge chamber 25b, and the refrigerant is discharged to the outside of the compression-discharge chamber 25b through the return port 16p. The discharged refrigerant is returned to the suction path 14 through the return path 16. Therefore, the pressure of the compression-discharge chamber 25b is not increased. As shown in the lower right of FIG. 3, when the rotational angle of the shaft 4 reaches 180 degrees, the compression-discharge chamber 25b is disconnected from the return path 16, and the refrigerant begins to be compressed in the compression-discharge chamber 25b. That is, the suction volume of the compression mechanism 3 is "V/2". The compression process continues until the pressure of the compression-discharge chamber 25b reaches the pressure of the internal space 28 of the closed casing 1. After the pressure of the compression-discharge chamber 25b has reached the pressure of the internal space 28, the discharge process is performed until the rotational angle of the shaft 4 reaches 360 degrees (0 degrees). As shown in the lower left and the upper left of FIG. 3, when the shaft 4 completes one rotation, the volume of the compression-discharge chamber 25b becomes zero.

[0028] In the case where the on-off valve 32 is closed, the refrigerant cannot return from the working chamber 25 to the suction path 14 through the return path 16. Therefore, the suction volume of the compression mechanism 3 is "V", and the compression process starts immediately after the end of the suction process. At this time, the portion of the return path 16 from the return port 16p to the on-off valve 32, that is, the upstream portion 16h of the return path 16, has a relatively high pressure. This is because when the on-off valve 32 is closed, the refrigerant compressed up to an intermediate pressure is gradually accumulated in the upstream portion 16h. If

the pressure of the compression-discharge chamber 25b is lower than the pressure of the upstream portion 16h, the check valve 35 prevents the refrigerant from flowing back to the compression-discharge chamber 25b from the return path 16. That is, since the check valve 35 is provided on the working chamber 25 side with respect to the on-off valve 32, it is possible to avoid a situation where the return path 16 causes dead volume. In the present embodiment, since the check valve 35 is provided in the upstream portion 16h formed inside the cylinder 5, the dead volume due to the return path 16 is substantially zero.

[0029] Next, with reference to FIG. 5A, a description will be given of the steps of control performed by the controller 44 for the volume varying mechanism 30 (the on-off valve 32) and the inverter 42.

[0030] In step S1, the rotational speed of the motor 2 is adjusted in accordance with a required power. Specifically, the rotational speed of the motor 2 is adjusted so as to obtain a required refrigerant flow rate. Next, in step S2 and step S6, it is determined whether the rotational speed of the motor 2 has been increased or decreased. When the process of decreasing the rotational speed has been performed in step S1, the control proceeds to step S3, and it is determined whether the current rotational speed is lower than or equal to 30 Hz. If the current rotational speed is lower than or equal to 30 Hz, it is determined in step S4 whether the on-off valve 32 is closed. If the on-off valve 32 is closed, the process of opening the on-off valve 32 and the process of increasing the rotational speed of the motor 2 to a rotational speed which is twice the current rotational speed, are executed in step S5. The order of the processes in step S5 is not particularly limited. The rotational speed of the motor 2 can be increased almost at the same time as the on-off valve 32 is caused to open.

[0031] On the other hand, when the process of increasing the rotational speed has been performed in step S1, the control proceeds to step S7, and it is determined whether the current rotational speed is higher than or equal to 70 Hz. If the current rotational speed is higher than or equal to 70 Hz, it is determined in step S8 whether the on-off valve 32 is open. If the on-off valve 32 is open, the process of closing the on-off valve 32 and the process of decreasing the rotational speed of the motor 2 to a rotational speed which is 1/2 times the current rotational speed, are executed in step S9. The order of the processes in step S9 is not particularly limited. The rotational speed of the motor 2 can be decreased almost at the same time as the on-off valve 32 is caused to close.

[0032] If the control is performed in accordance with the flowchart of FIG. 5A, the relationship between the state of the on-off valve 32 and the rotational speed of the motor 2 has a hysteresis as shown in FIG. 6. Such control allows prevention of hunting of the compression mechanism 3.

[0033] In the state where the on-off valve 32 is closed, that is, in the high volume mode in which the refrigerant

is prohibited from returning from the working chamber 25 to the suction path 14 through the return path 16, the suction volume of the compression mechanism 3 is "V". If the rotational speed of the motor 2 has been decreased from a high rotational speed to a first rotational speed (e.g., 30 Hz) or lower during the operation in the high volume mode, the controller 44 executes a process for the on-off valve 32 so as to decrease the suction volume, and also executes a process for the inverter 42 so as to increase the rotational speed of the motor 2. The process for the on-off valve 32 executed to decrease the suction volume is the process of opening the on-off valve 32. The process for the inverter 42 executed to increase the rotational speed of the motor 2 is the process of setting the target rotational speed of the motor 2 to a rotational speed which is twice the latest rotational speed.

[0034] In addition, the controller 44 controls the on-off valve 32 and the inverter 42 so as to compensate for an increase in the suction volume with a decrease in the rotational speed of the motor 2. In the state where the on-off valve 32 is open, that is, in the low volume mode in which the refrigerant is permitted to return from the working chamber 25 to the suction path 14 through the return path 16, the suction volume of the compression mechanism 3 is "V/2". If the rotational speed of the motor 2 has been increased to a second rotational speed (e.g., 70 Hz) or higher during the operation in the low volume mode, the controller 44 executes a process for the on-off valve 32 so as to increase the suction volume, and also executes a process for the inverter 42 so as to decrease the rotational speed of the motor 2. The process for the on-off valve 32 executed to increase the suction volume is the process of closing the on-off valve 32. The process for the inverter 42 executed to decrease the rotational speed of the motor 2 is the process of setting the target rotational speed of the motor 2 to a rotational speed which is 1/2 times the latest rotational speed.

[0035] As shown in FIG. 6, if the rotational speed of the motor 2 decreases to 30 Hz while the on-off valve 32 is closed, the on-off valve 32 is caused to open, and the rotational speed of the motor 2 is increased to 60 Hz. If the rotational speed of the motor 2 increases to 70 Hz while the on-off valve 32 is open, the on-off valve 32 is caused to close, and the rotational speed of the motor 2 is decreased to 35 Hz. Assuming that the rotational speed at the time when the on-off valve 32 is caused to open and the rotational speed of the motor 2 is increased is defined as a third rotational speed, and that the rotational speed at the time when the on-off valve 32 is caused to close and the rotational speed of the motor 2 is decreased is defined as a fourth rotational speed, the following relationships are satisfied: (the first rotational speed) < (the fourth rotational speed); and (the third rotational speed) < (the second rotational speed). For example, if the first rotational speed is set to a rotational speed lower than or equal to 30 Hz, the rotary compressor 100 can be operated with a broader range of power. The lower limit of the first rotational speed is not particularly limited, and

is, for example, 20 Hz.

[0036] When the operation mode is switched, the rotational speed of the motor 2 can be adjusted in accordance with (VL/VH) which is the ratio of a suction volume VL in the low volume mode to a suction volume VH in the high volume mode. When the operation mode is switched from the high volume mode to the low volume mode, the rotational speed (target rotational speed) of the motor 2 is set to a rotational speed that results from dividing the rotational speed of the motor 2 immediately before the mode switching by the ratio (VL/VH). Similarly, when the operation mode is switched from the low volume mode to the high volume mode, the rotational speed of the motor 2 is set to a rotational speed that results from multiplying the rotational speed of the motor 2 immediately before the mode switching by the ratio (VL/VH). This allows smooth switching of the operation mode between the high volume mode and the low volume mode.

[0037] It is not essential that 100% of a decrease in the power of the rotary compressor 100 caused by a decrease in the suction volume should be compensated for with an increase in the power of the rotary compressor 100 achieved by an increase in the rotational speed of the motor 2. In the example shown in FIG. 6, when the suction volume is decreased by 1/2 by opening the on-off valve 32, the rotational speed of the motor 2 is increased by twice. Therefore, the power of the rotary compressor 100 is not changed by the mode switching. However, no problem arises even if the power of the rotary compressor 100 is increased or decreased because of the mode switching.

[0038] The rotary compressor 100 may be stopped while the on-off valve 32 is open, in order to prevent the refrigerant from being confined in the upstream portion 16h of the return path 16 during the stoppage of the rotary compressor 100. A normally open valve can be used as the on-off valve 32. The check valve 35 may be a commonly-known lead valve including a stopper and a lead made of a thin metal plate.

[0039] Next, another example of the steps of control of the on-off valve 32 and the inverter 42 will be described.

[0040] The controller 44 may be configured to execute the process for the on-off valve 32 so as to decrease the suction volume, and the process for the inverter 42 so as to increase the rotational speed of the motor 2 when the flow rate of the refrigerant is excessive even if the rotational speed of the motor 2 is decreased to the first rotational speed (e.g., 30 Hz) in the high volume mode. That is, the controller 44 may be configured to determine the need for mode switching before the rotational speed of the motor 2 is actually decreased to the first rotational speed. Similarly, the controller 44 may be configured to execute the process for the on-off valve 32 so as to increase the suction volume, and the process for the inverter 42 so as to decrease the rotational speed of the motor 2 when the flow rate of the refrigerant is insufficient even if the rotational speed of the motor 2 is increased to the second rotational speed (e.g., 70 Hz) in the low

volume mode. That is, the controller 44 may be configured to determine the need for mode switching before the rotational speed of the motor 2 is actually increased to the second rotational speed. An example of such control will be described with reference to FIG. 5B.

[0041] As shown in FIG. 5B, a required rotational speed of the motor 2 is calculated in step S11 first. The "required rotational speed" means, for example, a rotational speed for obtaining a required refrigerant flow rate. Next, in step S12, it is determined whether the required rotational speed is lower than or equal to the first rotational speed (e.g., 30 Hz). If the required rotational speed is lower than or equal to the first rotational speed, it is determined in step S 13 whether the on-off valve 32 is closed. If the on-off valve 32 is closed, in step S15, the on-off valve 32 is caused to open, and the rotational speed of the motor 2 is adjusted to a rotational speed that allows the required refrigerant flow rate. If the on-off valve 32 is open, only the rotational speed of the motor 2 is adjusted in step S14.

[0042] On the other hand, if the required rotational speed is higher than the first rotational speed, it is determined in step S 16 whether the required rotational speed is higher than or equal to the second rotational speed (e.g., 70 Hz). If the required rotational speed is higher than or equal to the second rotational speed, it is determined in step S 17 whether the on-off valve 32 is open. If the on-off valve 32 is open, in step S18, the on-off valve 32 is caused to close, and the rotational speed of the motor 2 is adjusted to a rotational speed that allows the required refrigerant flow rate. If the on-off valve 32 is closed, only the rotational speed of the motor 2 is adjusted in step S 19.

[0043] Performing the control described with reference to FIG. 5A or FIG. 5B allows the rotary compressor 100 to exhibit high efficiency even when a low power is required (even when a load is small), as shown by a solid line in FIG. 7. In FIG. 7, the rated power of the rotary compressor 100 is "100%". When the rated power is defined as a reference, the efficiency of the rotary compressor 100 decreases with reduction of the power to be exerted, that is, with reduction of the rotational speed of the motor 2. As shown by a dashed line, the reduction of the efficiency is significant if the motor 2 is driven at a rotational speed which is 50% or less of the rated rotational speed. In the present embodiment, when a relatively low power is required, the rotary compressor 100 is operated in the low volume mode in which the suction volume is V/2. This allows the motor 2 to be driven at a rotational speed which is as close to the rated rotational speed as possible. Accordingly, the rotary compressor 100 can exhibit excellent efficiency even when the required power is 50% or less of the rated power.

[0044] Next, a description will be given of the effect that is based on the fact that the return path 16 communicates with the suction path 14 via the internal space of the accumulator 12.

[0045] Basically, all of the refrigerant present in the

suction path 14 is drawn into the suction chamber 25a. Therefore, as shown in FIG. 8A, the flow velocity of the refrigerant in the suction path 14 varies in proportion to the change rate of the volume of the suction chamber 25a (see FIG. 4A). Specifically, the flow velocity of the refrigerant in the suction path 14 shows a sine wave profile with respect to the rotational angle of the shaft 4 in theory.

[0046] In the case where the on-off valve 32 is open, the refrigerant in the compression-discharge chamber 25b is discharged to the return path 16 through the return port 16p during the period in which the rotational angle of the shaft 4 varies from 0 to 180 degrees. The amount of the refrigerant discharged to the return path 16 from the compression-discharge chamber 25b is equal to the amount of decrease in the volume of the compression-discharge chamber 25b in the period from 0 to 180 degrees. As shown in FIG. 8B, the flow velocity of the refrigerant in the return path 16 varies in proportion to the change rate of the volume of the compression-discharge chamber 25b (see FIG. 4B) only during the period in which the rotational angle of the shaft 4 varies from 0 to 180 degrees. Specifically, in theory, the flow velocity of the refrigerant in the return path 16 shows a sine wave profile in the period from 0 to 180 degrees, and is zero in the period from 180 to 360 degrees.

[0047] The refrigerant flows into the accumulator 12 from both the introduction pipe 12b and the return path 16. The refrigerant having flowed into the accumulator 12 can advance only to the suction path 14. Therefore, the flow velocity of the refrigerant in the introduction pipe 12b of the accumulator 12 is approximately equal to the difference between the flow velocity of the refrigerant in the suction path 14 and the flow velocity of the refrigerant in the return path 16. Specifically, in theory, the flow velocity of the refrigerant in the introduction pipe 12b shows a sine wave profile in the period from 180 to 360 degrees, and is zero in the period from 0 to 180 degrees, as shown in FIG. 8C.

[0048] When the rotational angle of the shaft 4 is 180 degrees, the flow velocity of the refrigerant in the return path 16 rapidly drops from the maximum flow velocity v to zero. In addition, when the rotational angle of the shaft 4 is 180 degrees, the flow velocity of the refrigerant in the introduction pipe 12b rapidly increases from zero to the maximum flow velocity v . Such rapid change of the flow velocity may foster occurrence of water hammering, leading to problems such as reduction of reliability and occurrence of noise which are caused by vibration of pipes constituting the suction path 14 and the return path 16. Furthermore, a pressure wave transmitted to the suction path 14 may reduce the volume efficiency of the suction chamber 25a, thus resulting in reduction of the efficiency of the rotary compressor 100. However, in the present embodiment, the return path 16 is connected to the suction path 14 via the internal space of the accumulator 12. This configuration can prevent occurrence of water hammering, thereby making it possible to efficiently

suppress vibration, noise, and efficiency reduction.

(Second embodiment)

[0049] As shown in FIG. 9, a rotary compressor 200 of the present embodiment includes a second compression mechanism 33, in addition to the compression mechanism 3 described in the first embodiment. Hereinafter, the components of the compression mechanism 3 described in the first embodiment will be represented by adding "first". For example, the cylinder 5 is represented as a first cylinder 5, the piston 8 is represented as a first piston 8, the vane 9 is represented as a first vane 9, the working chamber 25 is represented as a first working chamber 25, and the compression mechanism 3 is represented as a first compression mechanism 3. Hereinafter, the same components as those described above are denoted by the same reference characters, and the description thereof is omitted.

[0050] As shown in FIG. 9 and FIG. 10, the second compression mechanism 33 is composed of a second cylinder 55, a second piston 58, a second vane 59, and a second spring 60. The second cylinder 55 is disposed concentrically with the first cylinder 5. The second piston 58 is fitted to a second eccentric portion 4b of the shaft 4 and is disposed inside the second cylinder 55 so as to form a second working chamber 75 between the outer circumferential surface of the second piston 58 and the inner circumferential surface of the second cylinder 55. A second vane groove 64 is formed in the second cylinder 55. The second vane 59 having one end contacting the outer circumferential surface of the second piston 58 is placed in the second vane groove 64. The second spring 60 is disposed in the second vane groove 64 so as to push the second vane 59 toward the second piston 58. The second working chamber 75 between the second cylinder 55 and the second piston 58 is divided by the second vane 59, and thus a second suction chamber 75a and a second compression-discharge chamber 75b are formed. The refrigerant to be compressed is guided to the second working chamber 75 (the second suction chamber 75a) through the second suction path 15 and a second suction port 77. A second discharge port 79 is formed in the upper bearing 6 so as to guide the compressed refrigerant from the second working chamber 75 (the second compression-discharge chamber 75b) to the internal space 28 of the closed casing 1. The second discharge port 79 is provided with a discharge valve which is not shown in the drawings.

[0051] The lower bearing 7 is covered with a muffler 23 having an internal space capable of receiving the refrigerant compressed by the first compression mechanism 3. The first discharge port 29 of the first compression mechanism 3 is formed in the lower bearing 7. A flow path 26 that penetrates through the lower bearing 7, the first cylinder 5, an intermediate plate 53, the second cylinder 55, and the upper bearing 6, is formed so that the refrigerant compressed by the first compression mech-

anism 3 travels from the internal space of the muffler 23 to the internal space 28 of the closed casing 1.

[0052] The direction in which the first eccentric portion 4a projects is shifted by 180 degrees from the direction in which the second eccentric portion 4b projects. That is, the phase of the first piston 8 is shifted from the phase of the second piston 58 by 180 degrees in terms of the rotational angle of the shaft 4.

[0053] The refrigerant is supplied to the first compression mechanism 3 through the first suction path 14, and is supplied to the second compression mechanism 33 through the second suction path 15. The refrigerant is compressed by the first compression mechanism 3 or the second compression mechanism 33, and is then discharged to the internal space 28 of the closed casing 1. The first suction path 14 and the second suction path 15 are each connected to the accumulator 12. One of the suction paths 14 and 15 may be branched from the other inside or outside the accumulator 12.

[0054] As shown in FIG. 9 and FIG. 10, since the return path 16 is not connected to the second compression mechanism 33, the suction volume of the second compression mechanism 33 is always constant. The return path 16 is connected only to the first compression mechanism 3 so that only the suction volume of the first compression mechanism 3 can be varied. The production cost of the rotary compressor 200 can be reduced by allowing only the suction volume of the first compression mechanism 3 to be varied. It should be understood that the return path 16 may be connected to each of the compression mechanisms 3 and 33 so that each of the suction volumes of the compression mechanisms 3 and 33 can be varied.

[0055] In the present embodiment, the first compression mechanism 3 is located on the far side with respect to the motor 2, and the second compression mechanism 33 is located on the near side with respect to the motor 2. That is, the motor 2, the second compression mechanism 33, and the first compression mechanism 3 are arranged in this order along the axial direction of the shaft 4. The second compression mechanism 33 has a constant suction volume, and thus requires a large load torque also in the low volume mode. Therefore, when the second compression mechanism 33 is located on the near side with respect to the motor 2, a load applied to the shaft 4 in the low volume mode is alleviated, which can result in reduction of loss in the bearings 6 and 7, and the like. In addition, when the first compression mechanism 3 having a small suction volume in the low volume mode is located on the lower side, it is possible to reduce pressure loss caused by a flow of the compressed refrigerant to the internal space 28 of the closed casing 1 through the muffler 23. However, the positional relationship between the first compression mechanism 3 and the second compression mechanism 33 is not limited to the above relationship.

[0056] As described in the first embodiment, in the case where the return port 16p is formed at a position

corresponding to 180 degrees, "V" or "V/2" can be selected as the suction volume of the first compression mechanism 3. In addition, when the suction volume of the second compression mechanism 33 is "V", "2V" or "1.5V" can be selected as the sum of the suction volumes of the compression mechanisms 3 and 33.

[0057] Meanwhile, in the low volume mode in which the refrigerant is permitted to return from the first working chamber 25 to the first suction path 14 through the return path 16, the suction volume of the first compression mechanism 3 can be made substantially zero. Specifically, as shown in FIG. 11, the return port 16p may be formed at a position in the vicinity of the first discharge port 29. With this configuration, in the low volume mode, almost all of the refrigerant drawn into the first suction chamber 25a is returned to the accumulator 12 through the return path 16 without being compressed. That is, the function of the first compression mechanism 3 can be canceled. The sum of the suction volumes of the compression mechanisms 3 and 33 in the low volume mode is equal to the suction volume V of the second compression mechanism 33.

[0058] "Making the suction volume of the first compression mechanism 3 substantially zero" does not necessarily mean that the suction volume of the first compression mechanism 3 is absolutely zero. For example, when the suction volume in the high volume mode is V, the position of the return port 16p can be determined so that the suction volume in the low volume mode is less than $\{1 - (1/2)^{1/2}\}V/2$, and preferably less than V/10. In this configuration, the first compression mechanism 3 does not perform the work of compressing the refrigerant in the low volume mode, and can be said to lose its function.

(Third embodiment)

[0059] As shown in FIG. 12, a rotary compressor 300 of the present embodiment corresponds to a rotary compressor obtained by omitting the check valve 35 from the rotary compressor 100 of the first embodiment. In the first and second embodiments, the volume varying mechanism 30 is composed of the on-off valve 32 and the check valve 35. The check valve 35 contributes to reduction of dead volume, but is not directly involved in varying the suction volume. Therefore, even when the volume varying mechanism 30 consists of only the on-off valve 32 as in the present embodiment, the suction volume of the compression mechanism 3 can be varied.

(Fourth embodiment)

[0060] As shown in FIG. 13, a rotary compressor 400 of the present embodiment includes a valve 80 (solenoid valve 80) as the volume varying mechanism 30, and the valve 80 is capable of directly opening and closing the return port 16p. The other components are as described in the first embodiment.

[0061] The solenoid valve 80 is composed of a plunger

81, a coil 83, and a housing 85. The housing 85 has an internal flow path 85h as the upstream portion of the return path 16, and also has the return port 16p opening toward the working chamber 25. The plunger 81 is placed in the housing 85 so as to be movable forward and backward along the internal flow path 85h. When electricity is applied to the coil 83, the plunger 81 moves in a direction away from the shaft 4, and thus the return port 16p is opened. This allows the refrigerant to return from the working chamber 25 to the suction path 14 through the return path 16. When the application of electricity to the coil 83 is stopped, the plunger 81 is pushed in a direction toward the shaft 4, and the return port 16p is closed by the end portion of the plunger 81.

[0062] In the low volume mode, electricity is applied to the coil 83 to open the return port 16p. In the high volume mode, the application of electricity to the coil 83 is stopped to close the return port 16p. Since the return port 16p is opened and closed directly with the plunger 81, the dead volume when the return port 16p is closed is approximately zero. That is, the solenoid valve 80 of the present embodiment not only allows switching between the high volume mode and the low volume mode, but also can prevent the refrigerant that has an intermediate pressure from flowing back to the suction chamber 25a and expanding again in the high volume mode. In addition, according to the present embodiment, since the return port 16p is always open in the low volume mode, the refrigerant to be compressed can be supplied to the suction chamber 25a from both the suction port 27 and the

This is preferable in view of reducing pressure loss in the suction process. This effect can be obtained also by the third and fifth embodiments.

[0063] In addition, the solenoid 83 may be controlled so that the return port 16p is opened and closed in synchronization with the rotation of the shaft 4. That is, by adjusting the times when the return port 16p is opened and closed, the suction volume of the compression mechanism 3 can be varied in a multistep manner or in a continuous manner. For example, in the period during which the rotational angle of the shaft 4 varies from 0 to 90 degrees, electricity is applied to the coil 83 so that the refrigerant can flow into the return path 16. In the period during which the rotational angle of the shaft 4 varies from 90 to 360 degrees, the application of electricity to the coil 83 is stopped. This allows the rotary compressor 400 to be operated in a middle volume mode, as well as in the high volume mode and the low volume mode which have been described above.

(Fifth embodiment)

[0064] As shown in FIG. 14, a rotary compressor 500 of the present embodiment has a volume varying mechanism 30 having a structure different from that of the volume varying mechanism 30 of the rotary compressor 100 of the first embodiment. The other components are as described in the first embodiment.

[0065] The rotary compressor 500 has a three-way valve 90, a volume control valve 91, and a high-pressure path 92 which function together as the volume varying mechanism 30. The return path 16 has the upstream portion 16h formed inside the compression mechanism 3 (in particular, inside the cylinder 5), and also has the return port 16p opening toward the working chamber 25. The volume control valve 91 is disposed in the upstream portion 16h so as to be able to open and close the return port 16p. The high-pressure path 92 has one end connected to the three-way valve 90 and the other end connected to the oil reservoir 22. The high-pressure path 92 is a path for supplying, to the volume control valve 91, a pressure equal to the pressure of the refrigerant having been compressed. The rotary compressor 500 of the present embodiment is a so-called high-pressure shell type compressor in which the internal space 28 of the closed casing 1 is filled with the compressed refrigerant. In the oil reservoir 22, an oil having a pressure approximately equal to the pressure of the compressed refrigerant is retained. The three-way valve 90 is configured to allow either the suction path 14 or the high-pressure path 92 to connect with the upstream portion 16h of the return path 16. By controlling the three-way valve 90, the rotary compressor 500 can be operated in either the high volume mode or the low volume mode.

[0066] As shown in FIG. 15A and FIG. 15B, the volume control valve 91 includes a plunger 96 and a spring 97. The plunger 96 has the shape of a cylinder with a bottom surface facing the return port 16p, and is disposed so as to be able to slide into the upstream portion 16h which is cylindrical. The spring 97 is joined to the inside of the plunger 96, and imparts, to the plunger 96, a force directed away from the return port 16p. In the upstream portion 16h of the return path 16, a groove 16g is formed along the outer circumferential surface of the plunger 96. The groove 16g extends along the sliding direction of the plunger 96, and has a length greater than that of the plunger 96 in the sliding direction.

[0067] As shown in FIG. 15A, in the low volume mode, the three-way valve 90 is controlled so that the suction path 14 communicates with the upstream portion 16h of the return path 16. Thus, the plunger 96 is moved away from the return port 16p, and as a result, the refrigerant can flow from the working chamber 25 to the return path 16 thorough the return port 16p and the groove 16g. That is, when the suction path 14 is connected to the upstream portion 16h of the return path 16 by the three-way valve 90, the volume control valve 91 is opened to permit a flow of the refrigerant from the working chamber 25 to the suction path 14.

[0068] On the other hand, in the high volume mode, the three-way valve 90 is controlled so that the high-pressure path 92 communicates with the upstream portion 16h of the return path 16 as shown in FIG. 15B. Thus, the pressure of the oil in the oil reservoir 22 is exerted on the back surface of the plunger 96, and the plunger 96 is pushed against the return port 16p with a force

greater than the force of the spring 97, which disables the refrigerant from flowing from the working chamber 25 to the return path 16. That is, when the high-pressure path 92 is connected to the upstream portion 16h of the return path 16 by the three-way valve 90, the volume control valve 91 is closed to prohibit a flow of the refrigerant from the working chamber 25 to the suction path 14.

[0069] As described in the first embodiment, in the case where the check valve 35 is employed, the check valve 35 is opened and closed in synchronization with the rotation of the shaft 4. By contrast, the volume control valve 91 employed in the present embodiment is always open or is always closed. This is advantageous to reduction of vibration, noise, and pressure loss. In addition, also in the present embodiment, since the volume control valve 91 is configured to directly open and close the return port 16p, the problem of dead volume can be solved.

[0070] In the present embodiment, the high-pressure path 92 has one end connected to (opening toward) the oil reservoir 22. In order to achieve the objective of supplying a high pressure to the volume control valve 91, one end of the high-pressure path 92 may be connected to any portion in the internal space 28 of the closed casing 1. In addition, in the case where the rotary compressor 500 is used in a refrigeration cycle apparatus, the high-pressure path 92 may be connected to a high-pressure portion of a refrigerant circuit (for example, a portion between the rotary compressor 500 and a heat radiator). However, according to the present embodiment, when the volume control valve 91 is closed by exerting a high pressure on the plunger 96, a sealing effect can be obtained by the oil. This is preferable in view of preventing reduction of the efficiency due to leak of the refrigerant. According to the present embodiment, it is possible to prevent a situation where the refrigerant amount is insufficient because of accumulation of the liquid refrigerant in the upstream portion 16h of the return path 16. Even if the upstream portion 16h of the return path 16 is filled with the oil, the change of the volume of the oil with respect to the change of temperature is small. Therefore, no defect occurs even when the rotary compressor 500 is stopped, with the oil being confined in the upstream portion 16h of the return path 16. It should be understood that the rotary compressor 500 may be stopped, with the suction path 14 communicating with the upstream portion 16h of the return path 16.

(Sixth embodiment)

[0071] As shown in FIG. 16, a scroll compressor 600 of the present embodiment includes a scroll compression mechanism 603. The compression mechanism 603 includes an orbiting scroll 607, a stationary scroll 608, an Oldham ring 611, a bearing member 610, and a muffler 616. The orbiting scroll 607 and the stationary scroll 608 respectively have scroll laps 627 and 628. A working chamber 612 having a crescent shape is formed between the lap 627 and the lap 628. The orbiting scroll 607 is

fitted to the eccentric shaft 4a of the shaft 4, and is also prohibited from rotating on its axis by the Oldham ring 611. A discharge port 638 is formed at a central portion of the stationary scroll 608. A flow path 617 is formed in the stationary scroll 608 and the bearing member 610 in such a manner as to penetrate through the stationary scroll 608 and the bearing member 610.

[0072] When the shaft 4 rotates, the orbiting scroll 607 makes an orbital motion, with the lap 627 being engaged with the lap 628. The working chamber 612 reduces its volume while moving from outside to inside. As a result, the refrigerant drawn from the suction path 14 is compressed. The compressed refrigerant is discharged to the internal space 28 of the closed casing 1, passing through the discharge port 638, an internal space 619 of the muffler 616, and the flow path 617 in this order. The refrigerant discharged to the internal space 28 is then guided to the outside of the compressor 600 through the discharge path 11.

[0073] The scroll compressor 600 has the volume varying mechanism 30 described in the first embodiment. In the present embodiment, the upstream portion 16h of the return path 16 is formed inside the compression mechanism 603, in particular, inside the stationary scroll 608. In addition, the return port 16p is formed in the stationary scroll 608 so as to allow the working chamber 612 to communicate with the return path 16. The check valve 35 is attached to the stationary scroll 608 so as to be able to open and close the return port 16p. As in the rotary compressor 100, the ratio of the suction volume in the low volume mode to the suction volume in the high volume mode varies depending on the position of the return port 16p.

[0074] The configuration and operation of the volume varying mechanism 30 are as described in the first embodiment. Control for switching between the high volume mode and the low volume mode is also as described in the first embodiment. Therefore, with the scroll compressor 600, the same effect as provided by the rotary compressor 100 can be obtained. In the present embodiment, no accumulator is provided, and the return path 16 is directly connected to the suction path 14 in the vicinity of the compression mechanism 603. However, an accumulator may be provided as in the several embodiments described above.

(Seventh embodiment)

[0075] As shown in FIG. 17, a scroll compressor 700 of the present embodiment has the volume varying mechanism 30 including the three-way valve 90, the volume control valve 91, and the high-pressure path 92. That is, the scroll compressor 700 has the volume varying mechanism 30 described in the fifth embodiment. As described in the sixth embodiment, the upstream portion 16h and the return port 16p of the return path 16 are formed in the stationary scroll 608. The volume control valve 91 is attached to the stationary scroll 608 so as to be able to

open and close the return port 16p. In the scroll compressor 700, the configuration and operation of the volume varying mechanism 30 are as described in the fifth embodiment. Control for switching between the high volume mode and the low volume mode is also as described in the fifth embodiment. Therefore, with the scroll compressor 700, the same effect as provided by the rotary compressor 500 can be obtained.

[0076] Alternatively, the configurations described with reference to FIG. 12 and FIG. 13 can be applied to a scroll compressor.

(Applied embodiment)

[0077] As shown in FIG. 18, a refrigeration cycle apparatus 800 can be built using the rotary compressor 100. The refrigeration cycle apparatus 800 includes the rotary compressor 100, a heat radiator 802, an expansion mechanism 804, and an evaporator 806. These devices are connected in the above order by refrigerant pipes so as to form a refrigerant circuit. The heat radiator 802 is composed of, for example, an air-refrigerant heat exchanger, and cools the refrigerant compressed by the rotary compressor 100. The expansion mechanism 804 is composed of, for example, an expansion valve, and expands the refrigerant cooled by the heat radiator 802. The evaporator 806 is composed of, for example, an air-refrigerant heat exchanger, and heats the refrigerant expanded by the expansion mechanism 804. The compressor 200, 300, 400, 500, 600, or 700 according to one of the second to fifth embodiments may be used instead of the rotary compressor 100 of the first embodiment.

(Other embodiments)

[0078] The several embodiments described in the present specification can be combined with each other without departing from the gist of the invention. For example, also when the check valve 35 described in the first embodiment is used in combination with the three-way valve 90 described in the fifth embodiment, the effect described in the first embodiment can be obtained.

[0079] In addition, at startup of the rotary compressor 100, the volume varying mechanism 30 can be controlled so as to permit the refrigerant to return from the working chamber 25 to the suction path 14 through the return path 16. That is, at startup, the rotary compressor 100 is operated temporarily in the low volume mode.

INDUSTRIAL APPLICABILITY

[0080] The present invention is useful for a compressor of a refrigeration cycle apparatus which is usable for a hot water dispenser, a hot water heater, an air conditioner, or the like. The present invention is particularly useful for an air conditioner for which a broad range of power is required.

Claims

1. A positive displacement compressor comprising:

- 5 a compression mechanism having a working chamber;
- a motor that moves the compression mechanism;
- 10 a suction path that guides a working fluid to be compressed to the working chamber;
- a return path that returns the working fluid from the working chamber to the suction path;
- a volume varying mechanism that is provided in the return path, permits the working fluid to return from the working chamber to the suction path through the return path when a suction volume of the compression mechanism should be set to a relatively small value, and prohibits the working fluid from returning from the working chamber to the suction path through the return path when the suction volume should be set to a relatively large value;
- 15 an inverter that drives the motor; and
- a controller that controls the volume varying mechanism and the inverter so as to compensate for a decrease in the suction volume with an increase in a rotational speed of the motor.

2. The positive displacement compressor according to claim 1, wherein when the rotational speed of the motor has been decreased to a first rotational speed or lower during operation in a high volume mode in which the working fluid is prohibited from returning from the working chamber to the suction path through the return path, or when a flow rate of the working fluid is excessive even if the rotational speed of the motor is decreased to the first rotational speed in the high volume mode, the controller executes a process for the volume varying mechanism so as to decrease the suction volume, and a process for the inverter so as to increase the rotational speed of the motor.

3. The positive displacement compressor according to claim 1, wherein the controller controls the volume varying mechanism and the inverter so as to compensate for an increase in the suction volume with a decrease in the rotational speed of the motor.

4. The positive displacement compressor according to claim 3, wherein when the rotational speed of the motor has been increased to a second rotational speed or higher during operation in a low volume mode in which the working fluid is permitted to return from the working chamber to the suction path through the return path, or when a flow rate of the working fluid is insufficient even if the rotational speed of the motor is increased to the second rota-

tional speed in the low volume mode, the controller executes a process for the volume varying mechanism so as to increase the suction volume, and a process for the inverter so as to decrease the rotational speed of the motor.

5. The positive displacement compressor according to claim 3, wherein

when the rotational speed of the motor has been decreased to a first rotational speed or lower during operation in a high volume mode in which the working fluid is prohibited from returning from the working chamber to the suction path through the return path, or when a flow rate of the working fluid is excessive even if the rotational speed of the motor is decreased to the first rotational speed in the high volume mode, the controller executes a process for the volume varying mechanism so as to decrease the suction volume, and a process for the inverter so as to increase the rotational speed of the motor, and when the rotational speed of the motor has been increased to a second rotational speed or higher during operation in a low volume mode in which the working fluid is permitted to return from the working chamber to the suction path through the return path, or when a flow rate of the working fluid is insufficient even if the rotational speed of the motor is increased to the second rotational speed in the low volume mode, the controller executes a process for the volume varying mechanism so as to increase the suction volume, and a process for the inverter so as to decrease the rotational speed of the motor.

6. The positive displacement compressor according to claim 2 or 5, wherein the first rotational speed is set to a rotational speed lower than or equal to 30 Hz.

7. The positive displacement compressor according to any one of claims 1 to 6, further comprising an accumulator that has an internal space capable of retaining the working fluid and to which the suction path and the return path are connected, wherein the return path is connected to the suction path via the internal space of the accumulator.

8. The positive displacement compressor according to any one of claims 1 to 7, wherein, at startup of the positive displacement compressor, the controller controls the volume varying mechanism so as to permit the working fluid to return from the working chamber to the suction path through the return path.

9. The positive displacement compressor according to any one of claims 1 to 8, wherein the volume varying mechanism includes an on-off valve provided in the return path.

10. The positive displacement compressor according to

claim 9, wherein

the return path includes an upstream portion formed inside the compression mechanism, the volume varying mechanism further includes a check valve provided in the upstream portion, and the check valve blocks a flow of the working fluid from the return path to the working chamber.

11. The positive displacement compressor according to any one of claims 1 to 8, wherein the return path includes an upstream portion formed inside the compression mechanism, the volume varying mechanism includes: a three-way valve; a volume control valve provided in the upstream portion; and a high-pressure path that supplies, to the volume control valve, a pressure equal to a pressure of the working fluid having been compressed, and the three-way valve is configured to connect either the suction path or the high-pressure path to the upstream portion of the return path.

12. The positive displacement compressor according to claim 11, wherein the volume control valve includes a plunger and a spring, the volume control valve is closed to prohibit a flow of the working fluid from the working chamber to the suction path when the high-pressure path is connected to the upstream portion of the return path by the three-way valve, and the volume control valve is opened to permit the flow of the working fluid from the working chamber to the suction path when the suction path is connected to the upstream portion of the return path by the three-way valve.

13. The positive displacement compressor according to any one of claims 1 to 12, wherein

the compression mechanism is a rotary compression mechanism including: a cylinder; a piston disposed inside the cylinder so as to form the working chamber between an outer circumferential surface of the piston and an inner circumferential surface of the cylinder; and a vane that divides the working chamber into a suction chamber and a compression-discharge chamber, and

when the cylinder, the piston, the vane, the working chamber, and the compression mechanism are defined as a first cylinder, a first piston, a first vane, a first working chamber, and a first compression mechanism, respectively, the positive displacement compressor further comprises a second compression mechanism including a second cylinder, a second piston, and a second vane, the second piston being moved by the same motor that moves the first compression mechanism,

a suction volume of the second compression mechanism is constant, and
the return path is connected only to the first compression mechanism so that only the suction volume of the first compression mechanism can be varied.

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- 14.** The positive displacement compressor according to claim 13, wherein the suction volume of the first compression mechanism is substantially zero in a low volume mode in which the working fluid is permitted to return from the first working chamber to the suction path through the return path.

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FIG.1

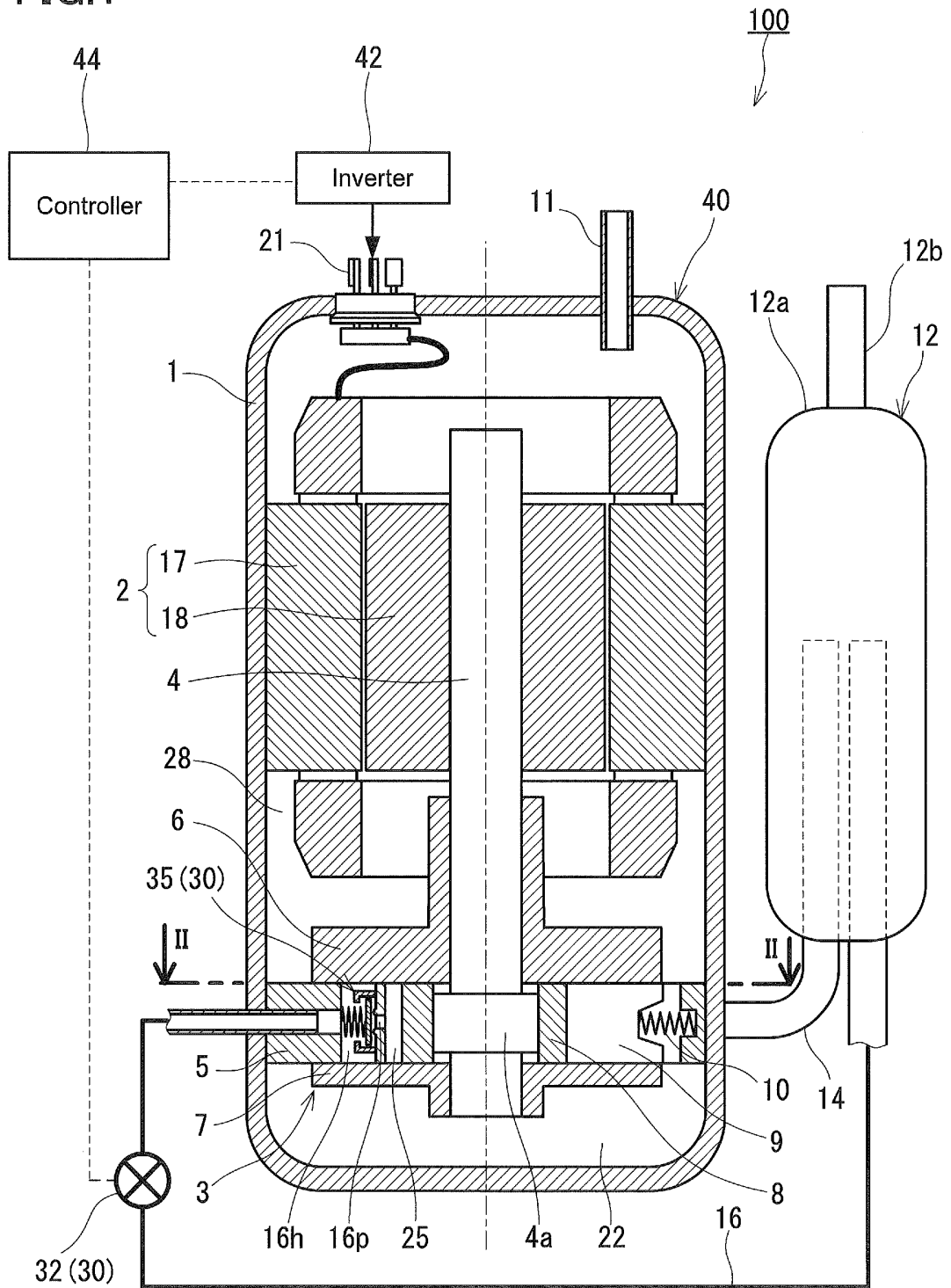


FIG. 2

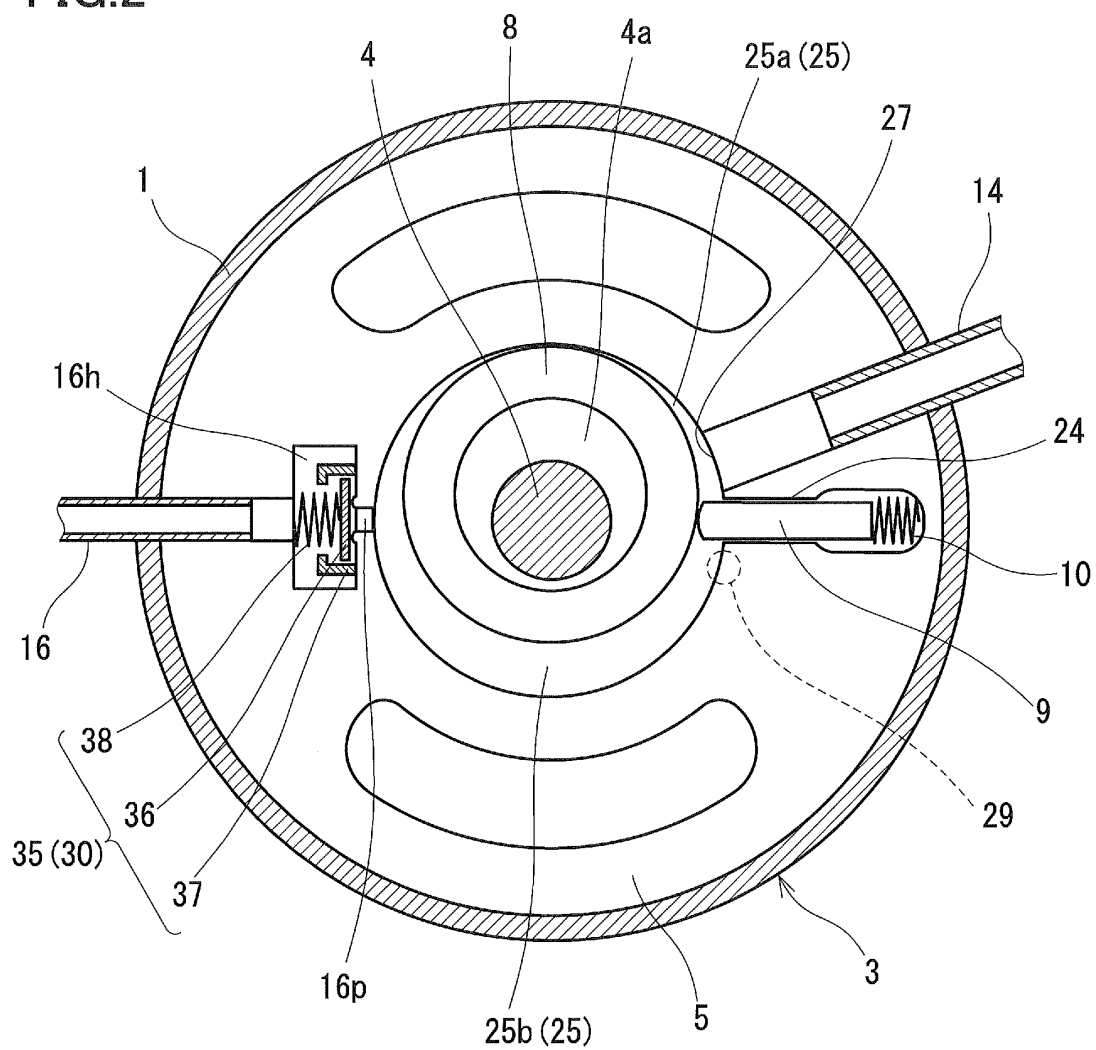


FIG.3

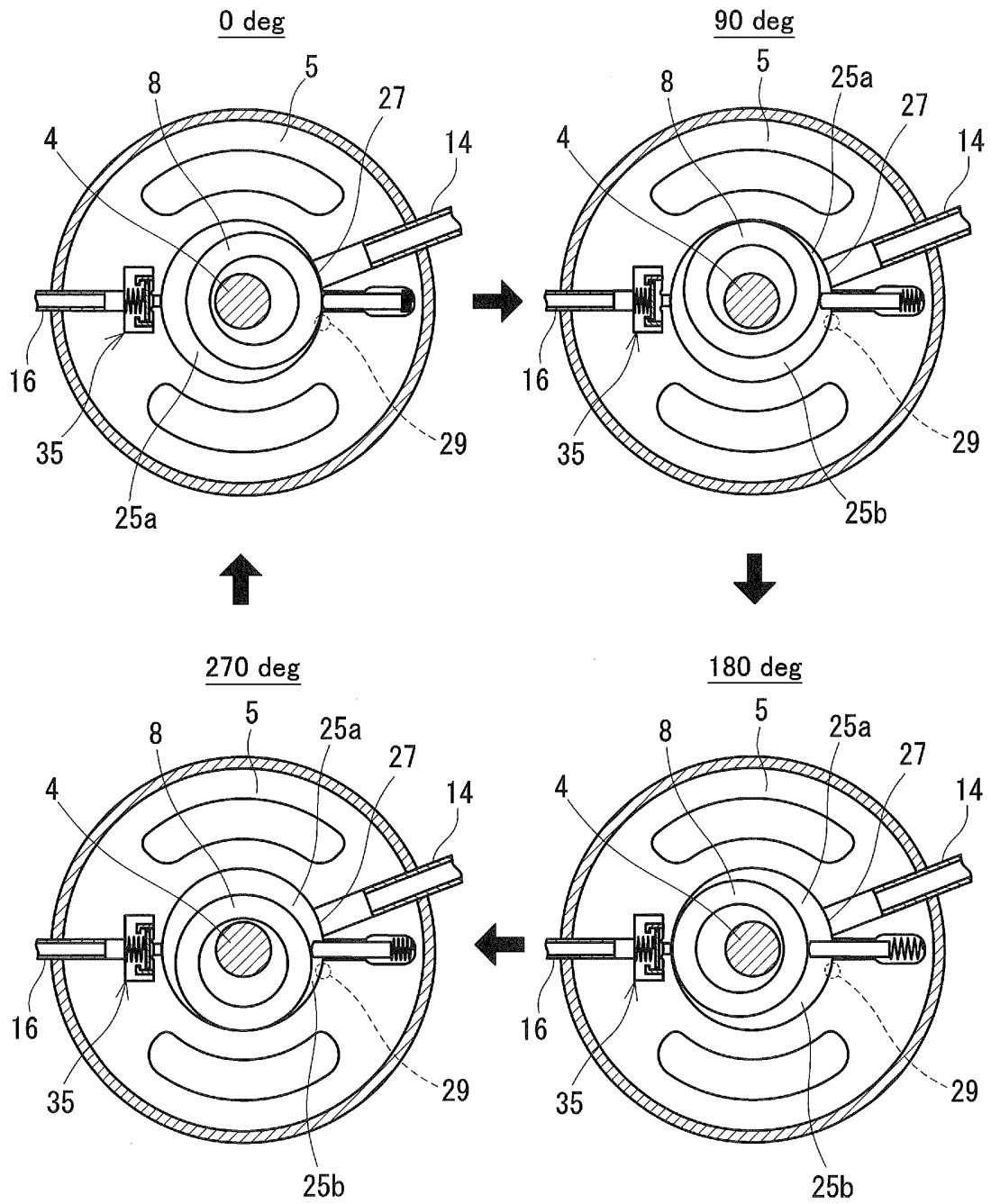


FIG.4A

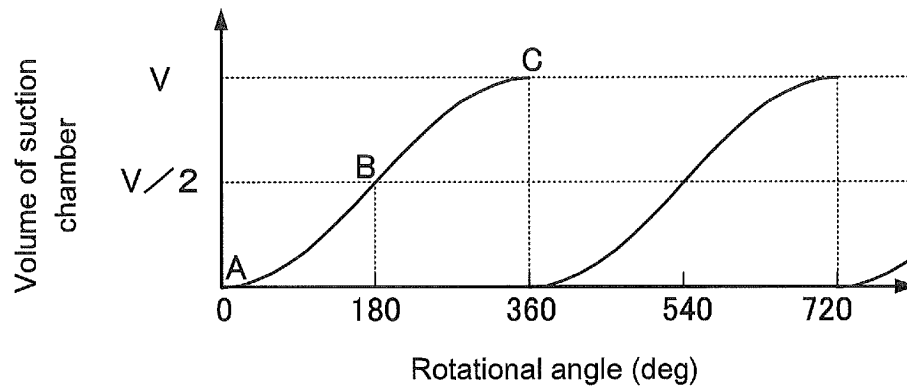


FIG.4B

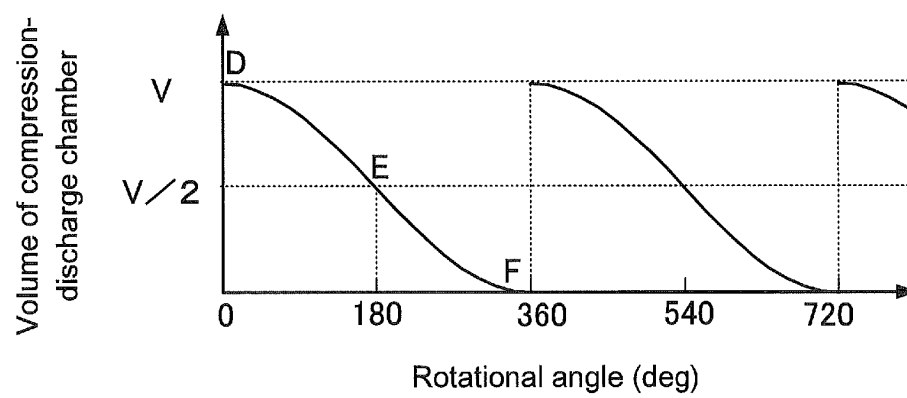


FIG.5A

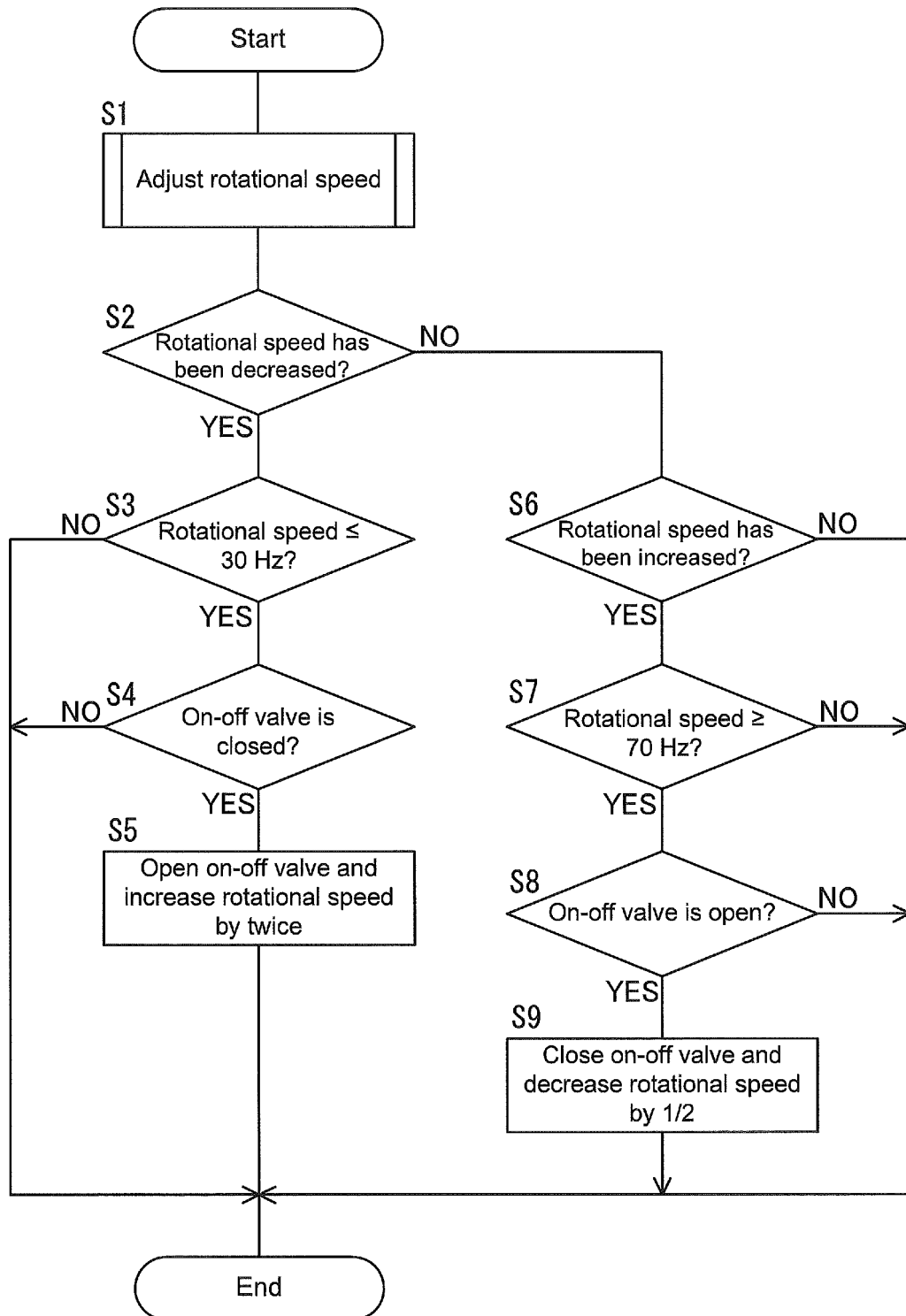


FIG.5B

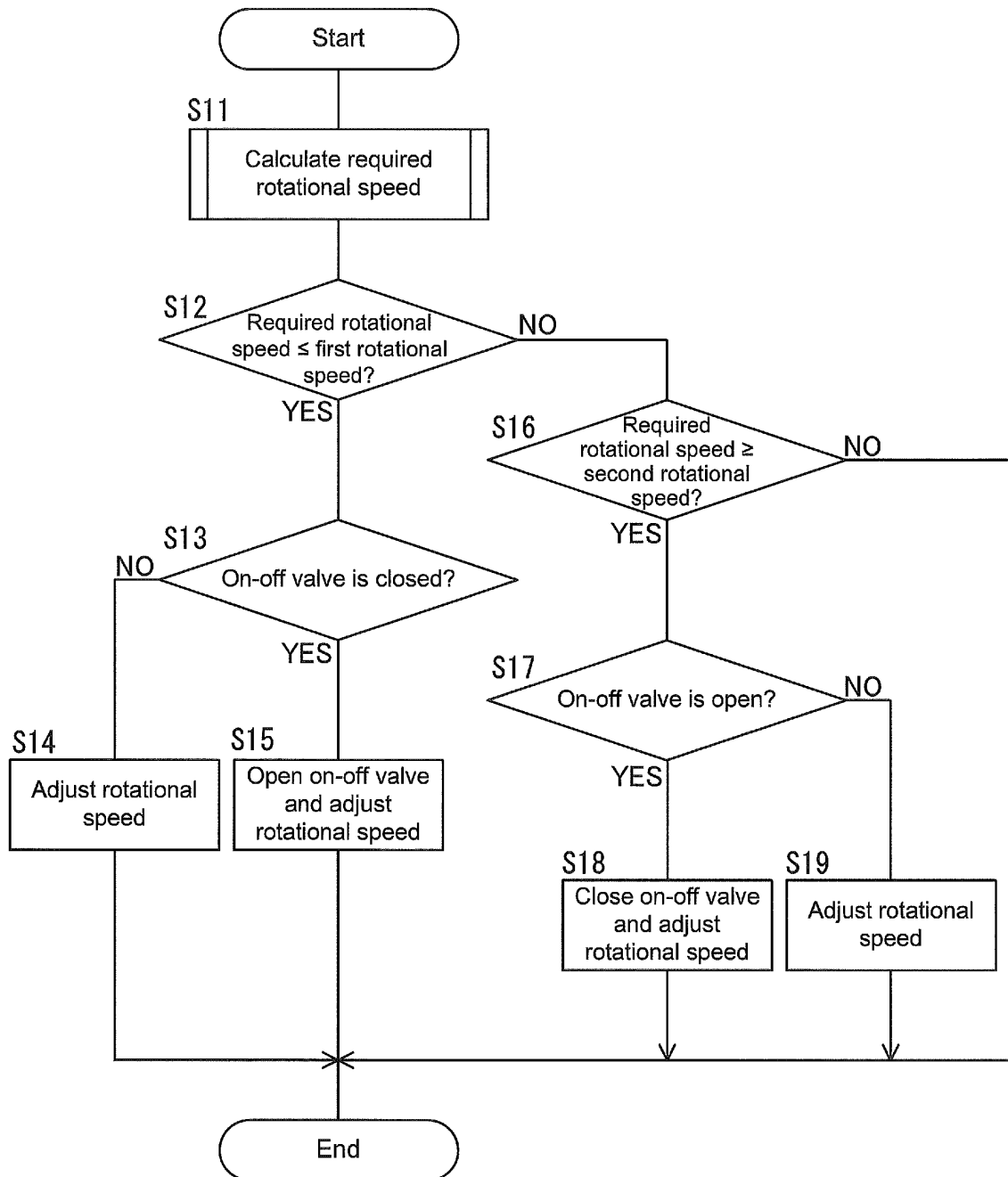


FIG.6

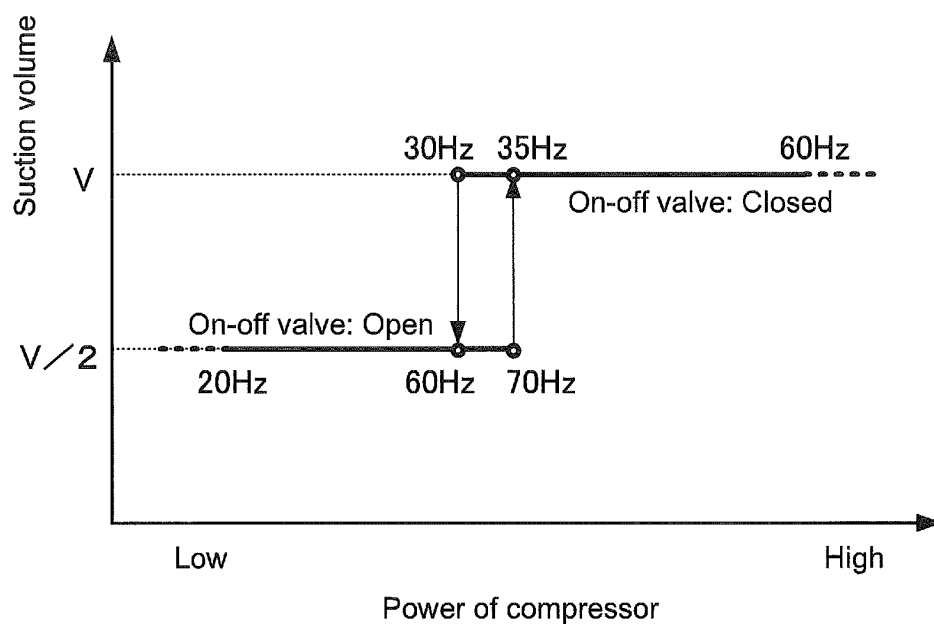


FIG.7

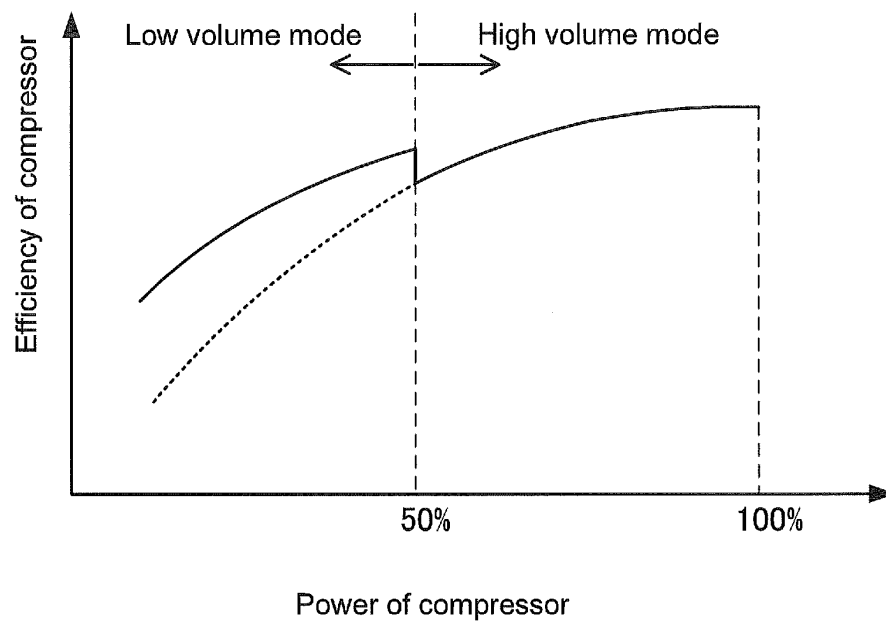


FIG.8A

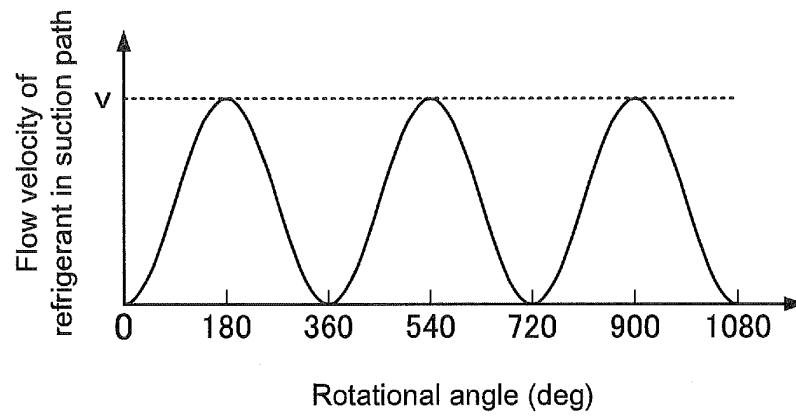


FIG.8B

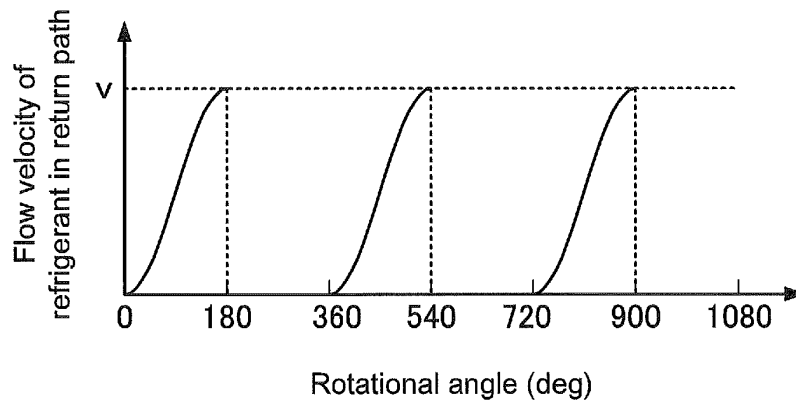


FIG.8C

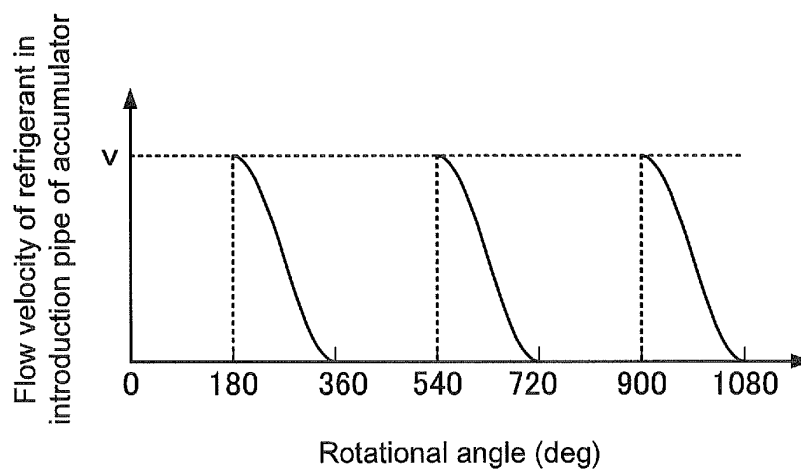


FIG.9

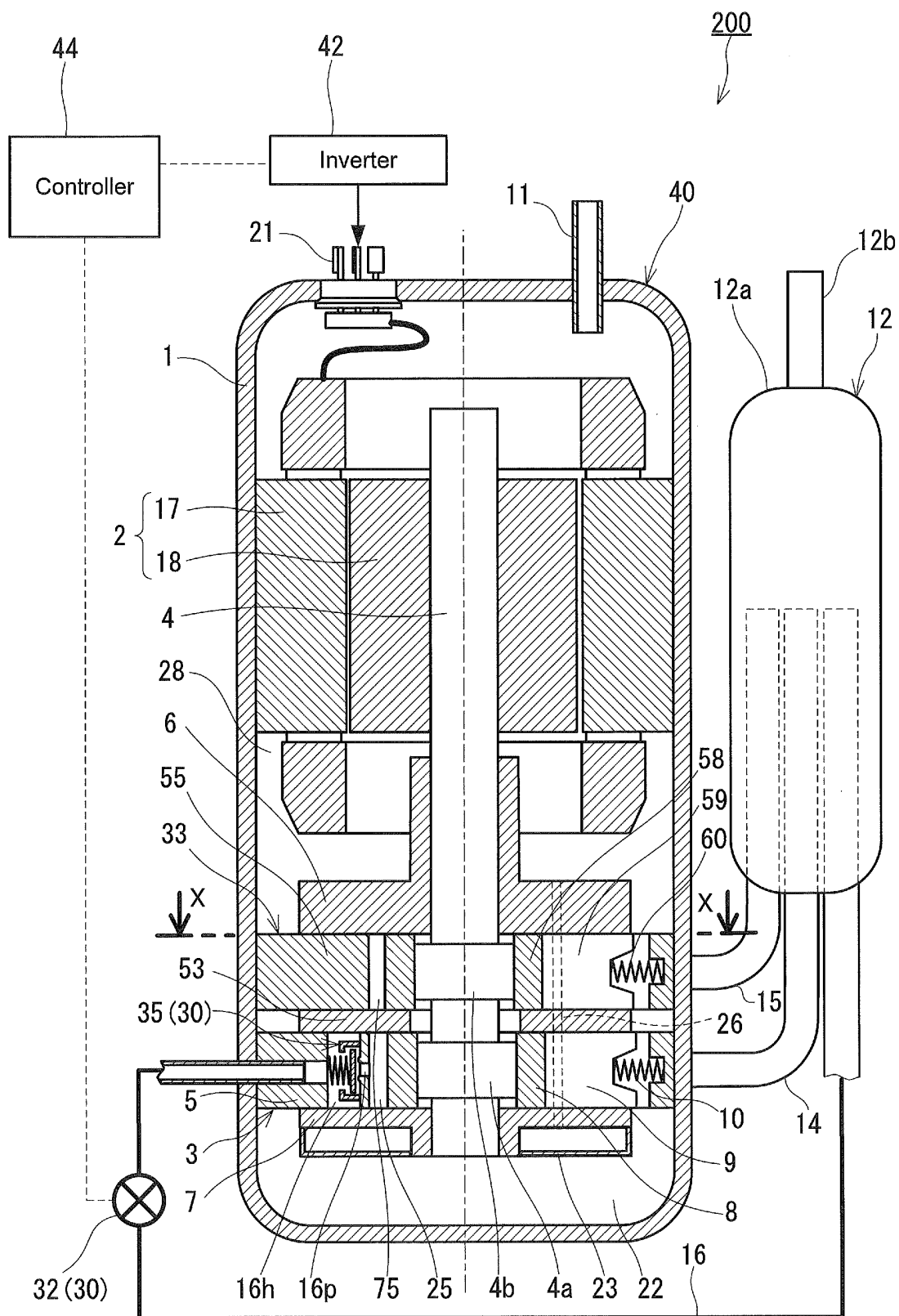


FIG. 10

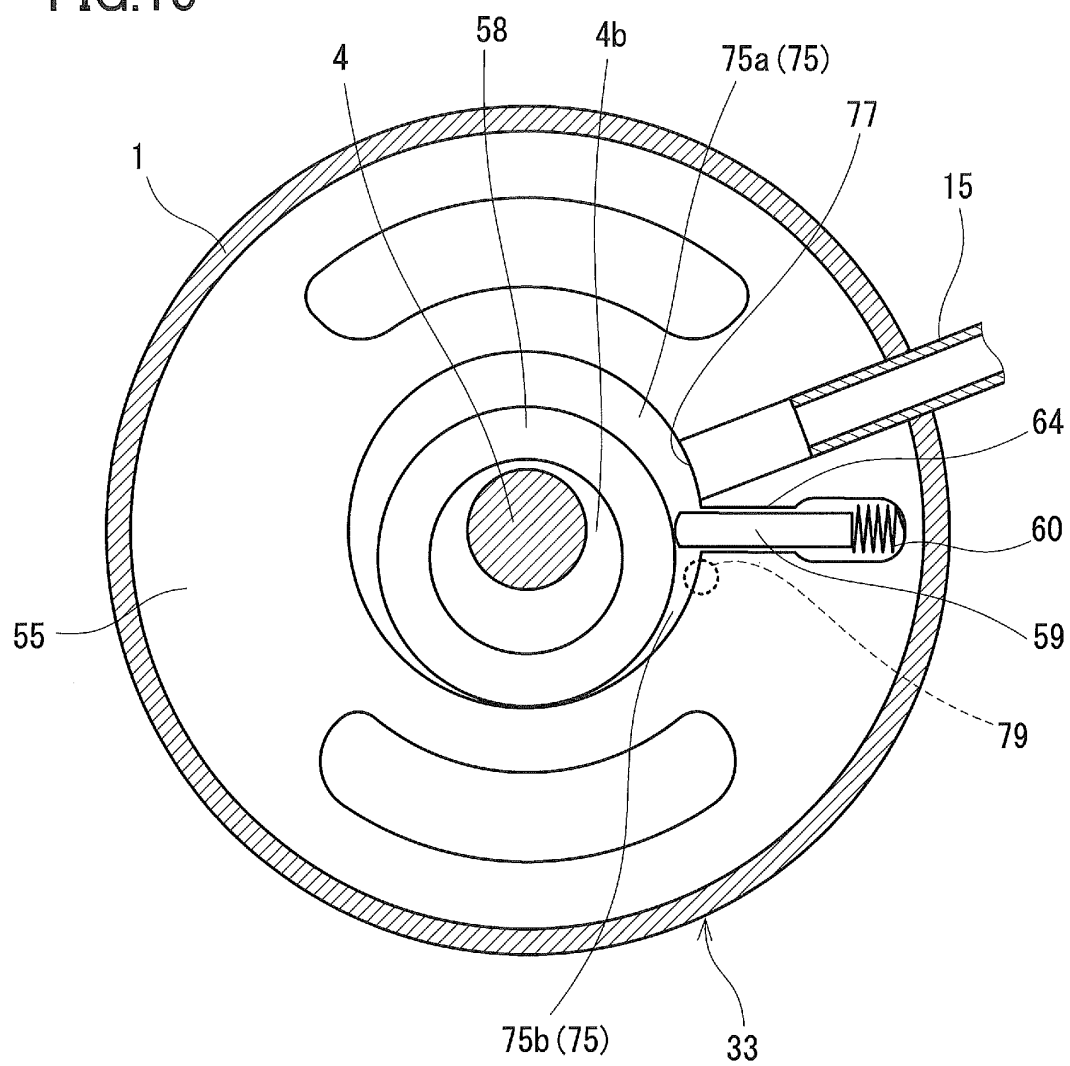


FIG.11

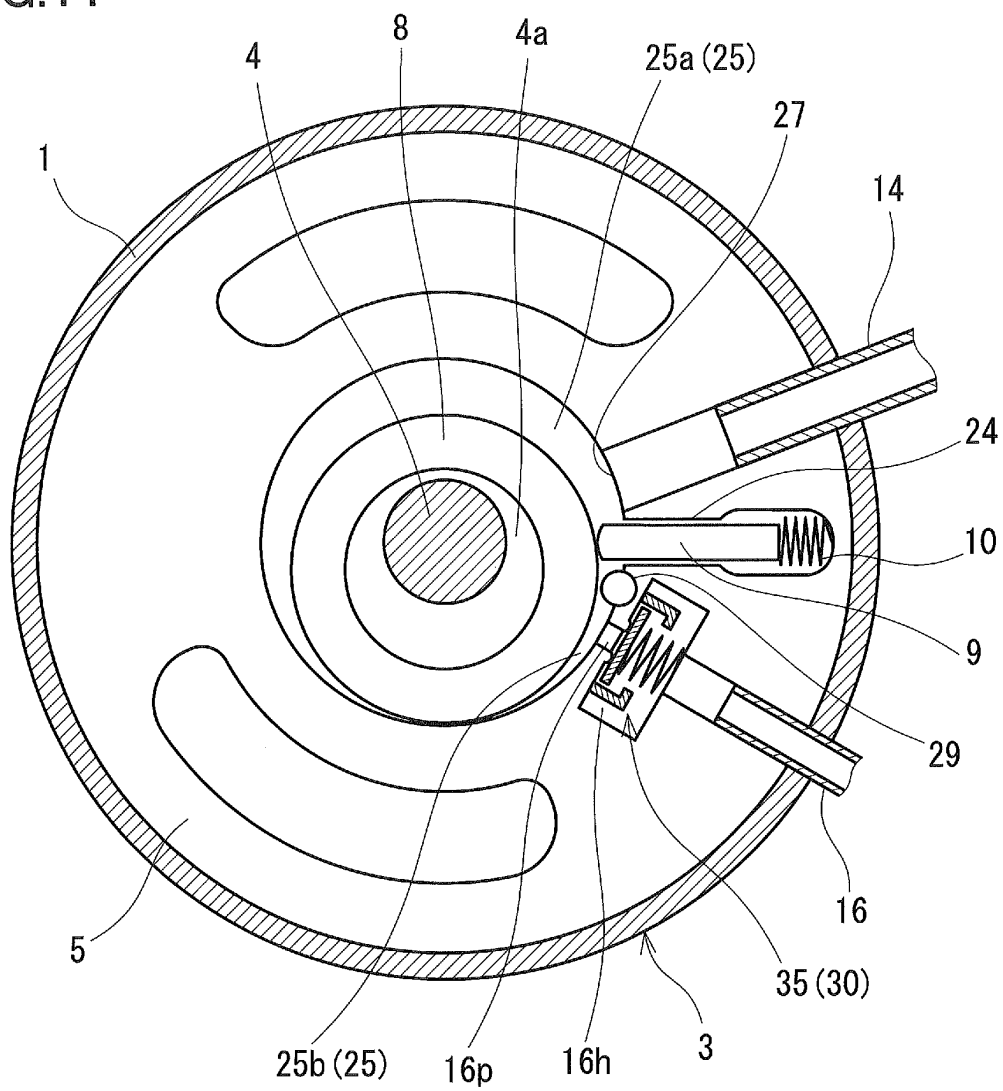


FIG.12

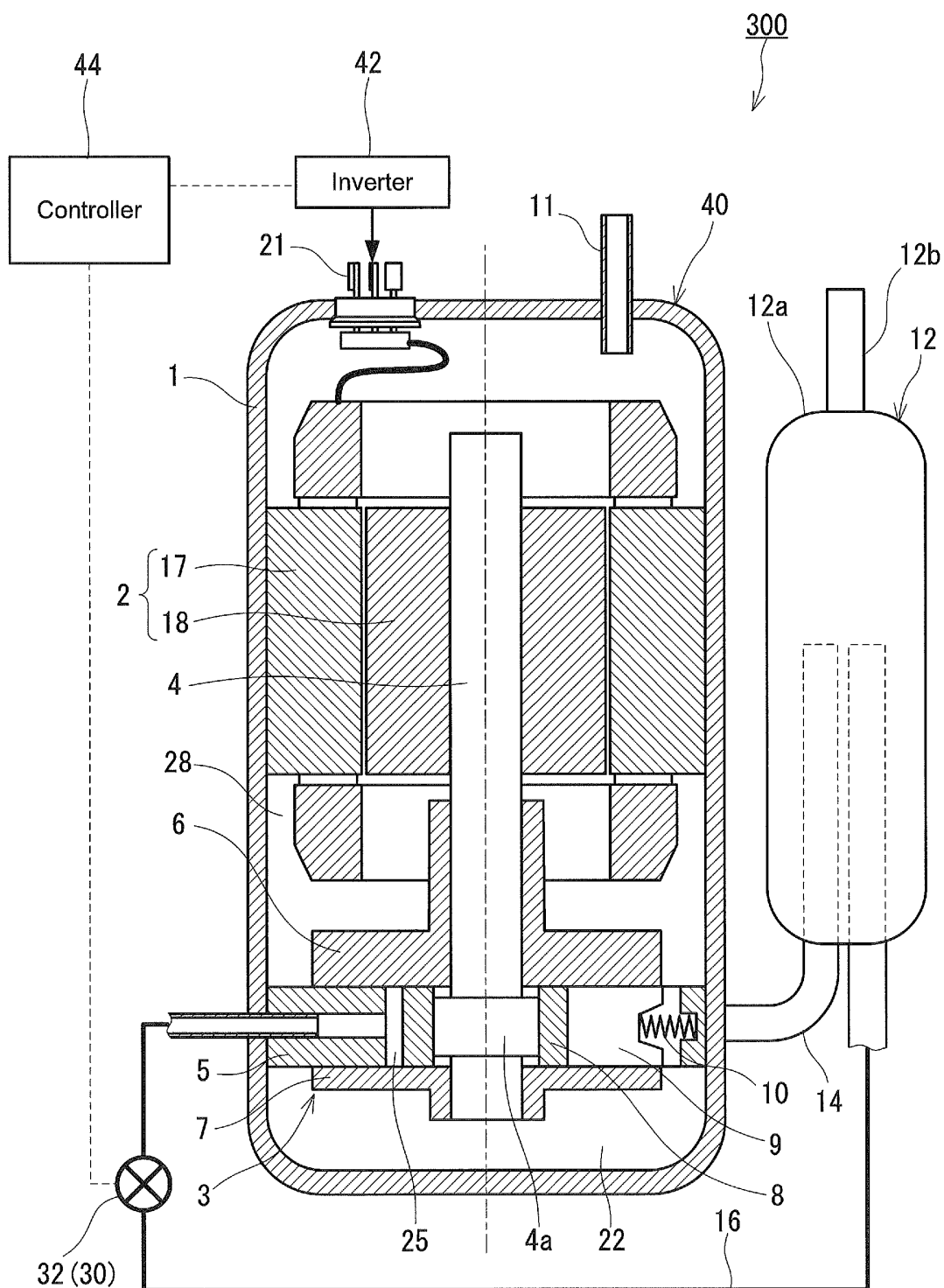


FIG.13

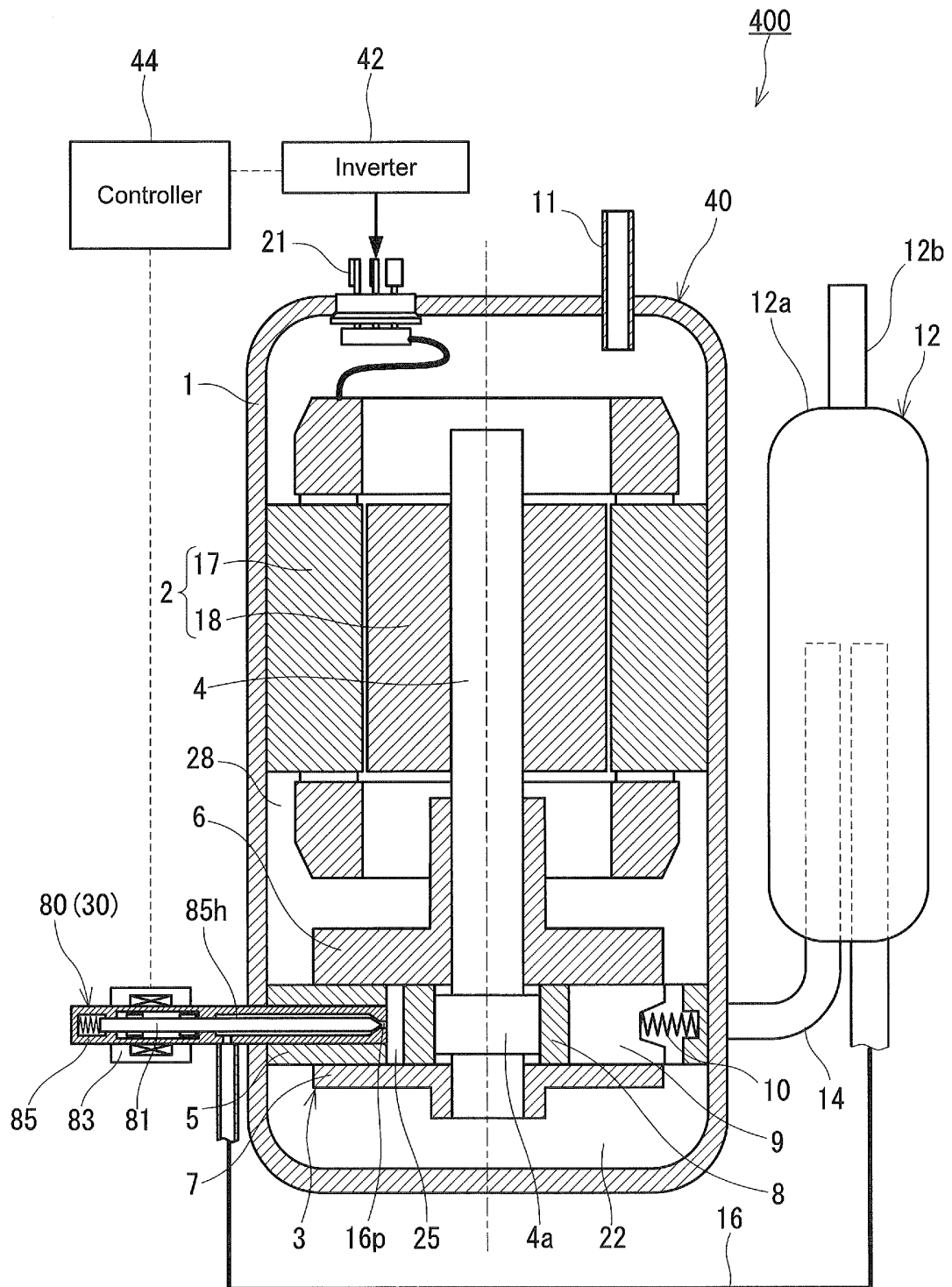


FIG.14

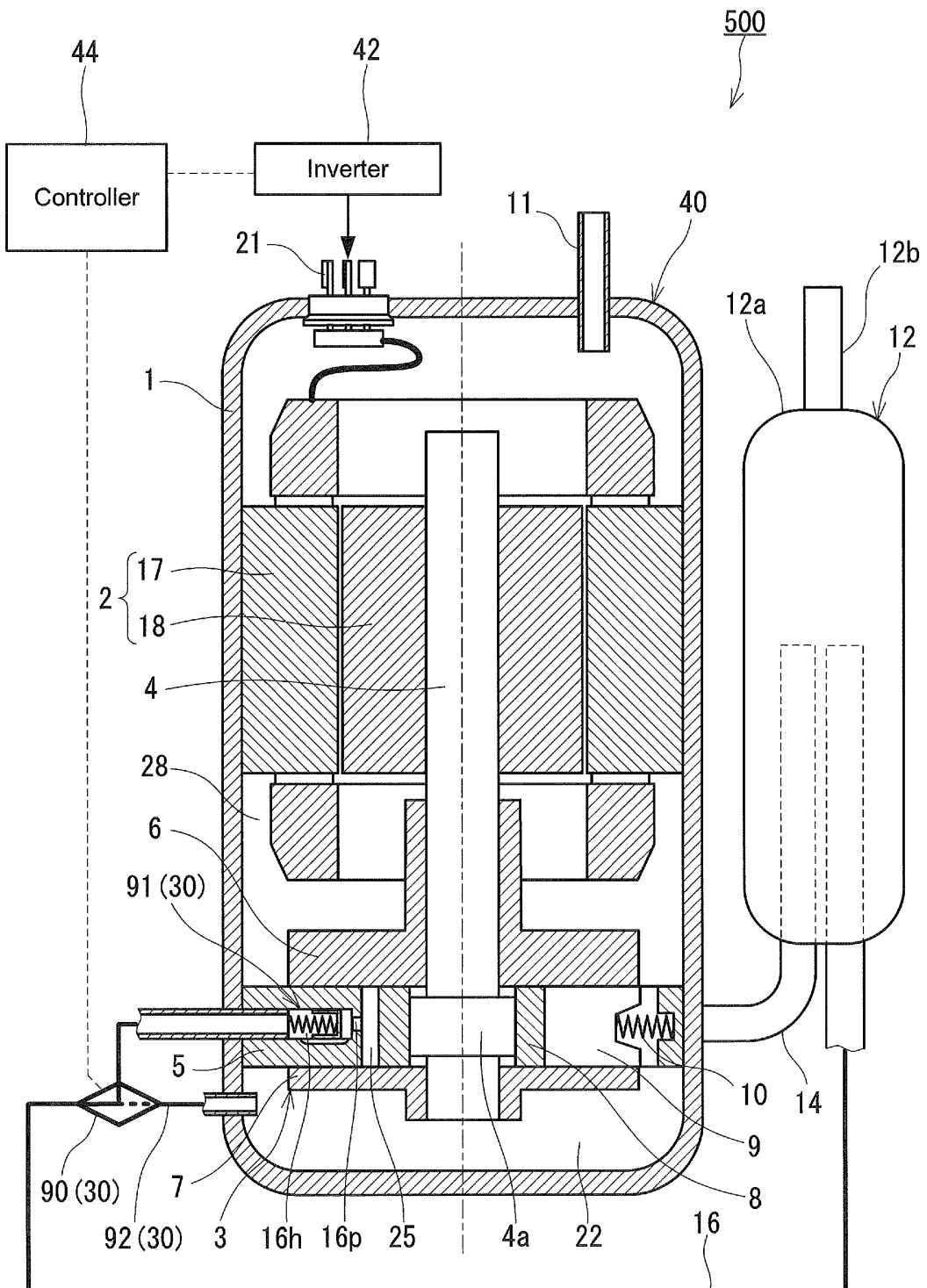


FIG.15A

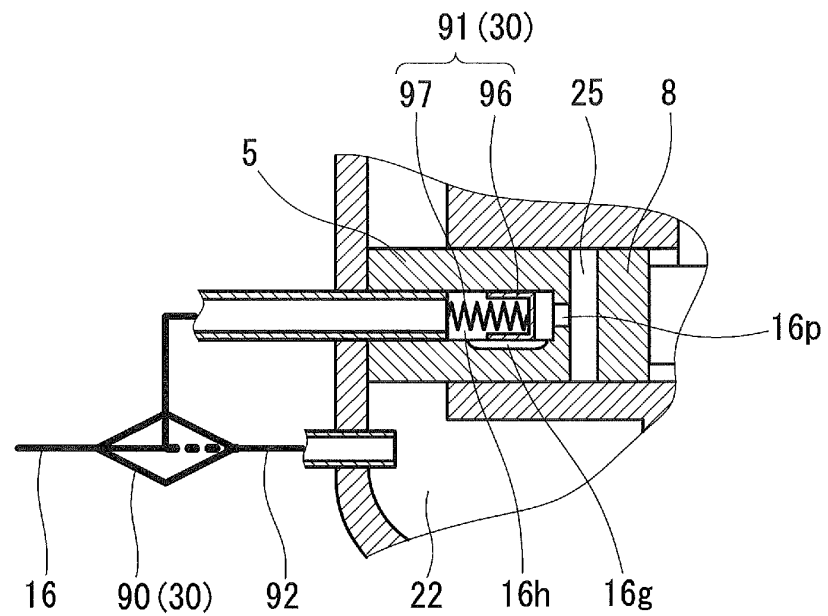


FIG.15B

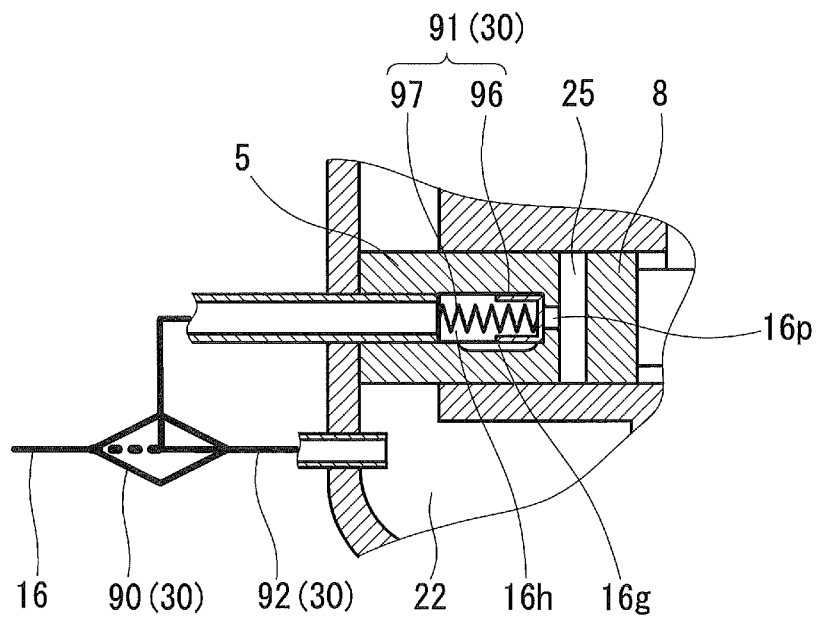


FIG.16

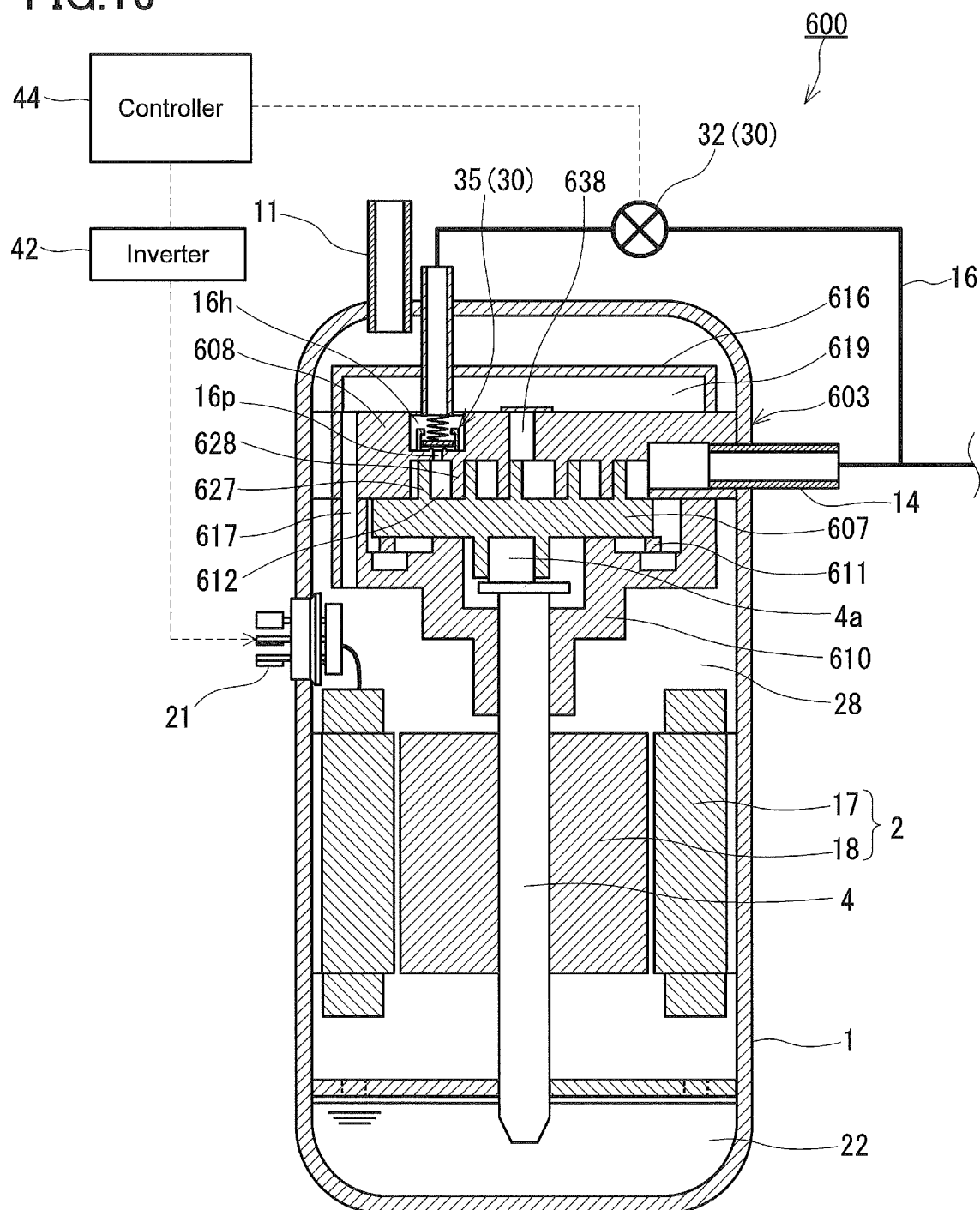


FIG.17

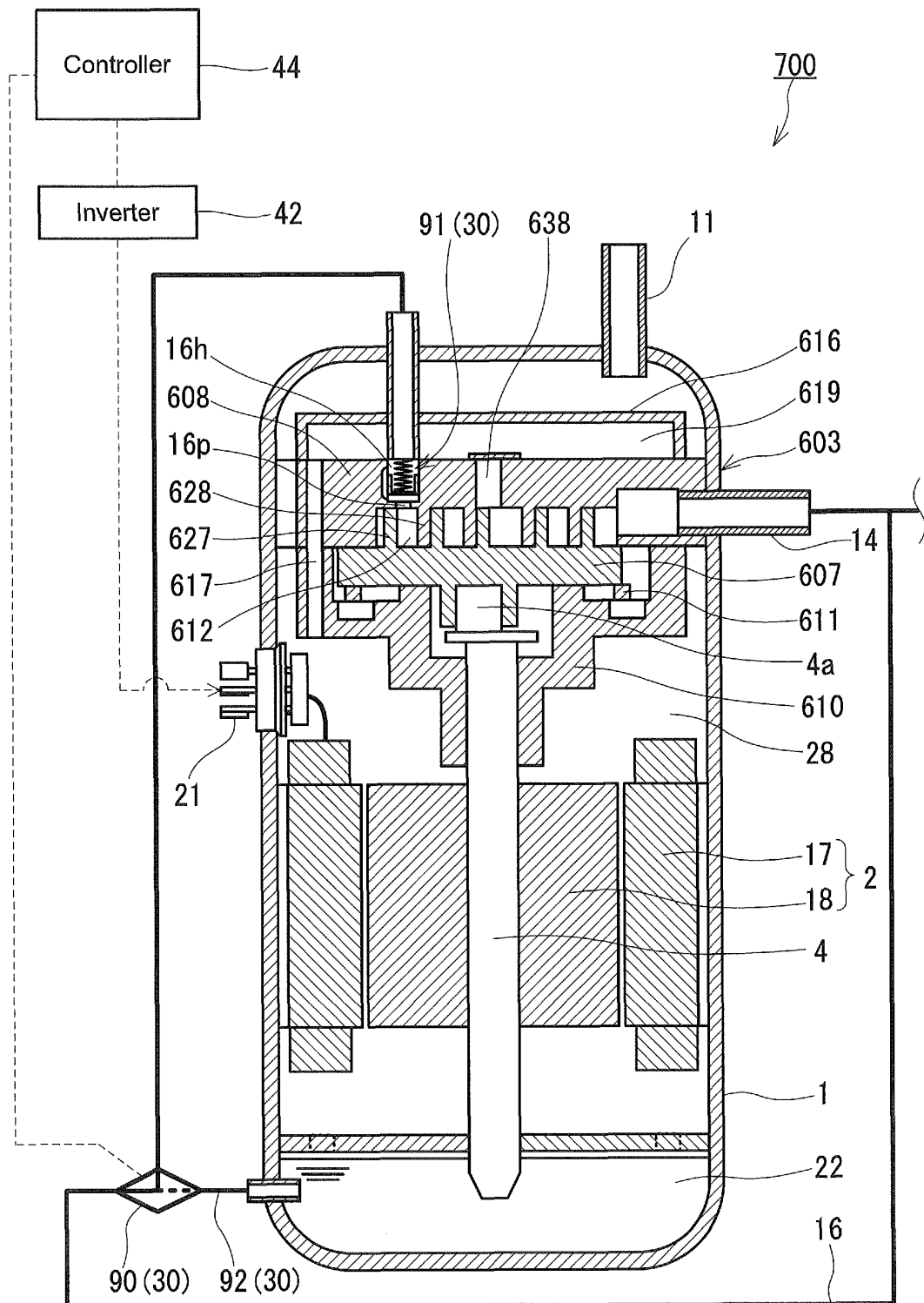


FIG.18

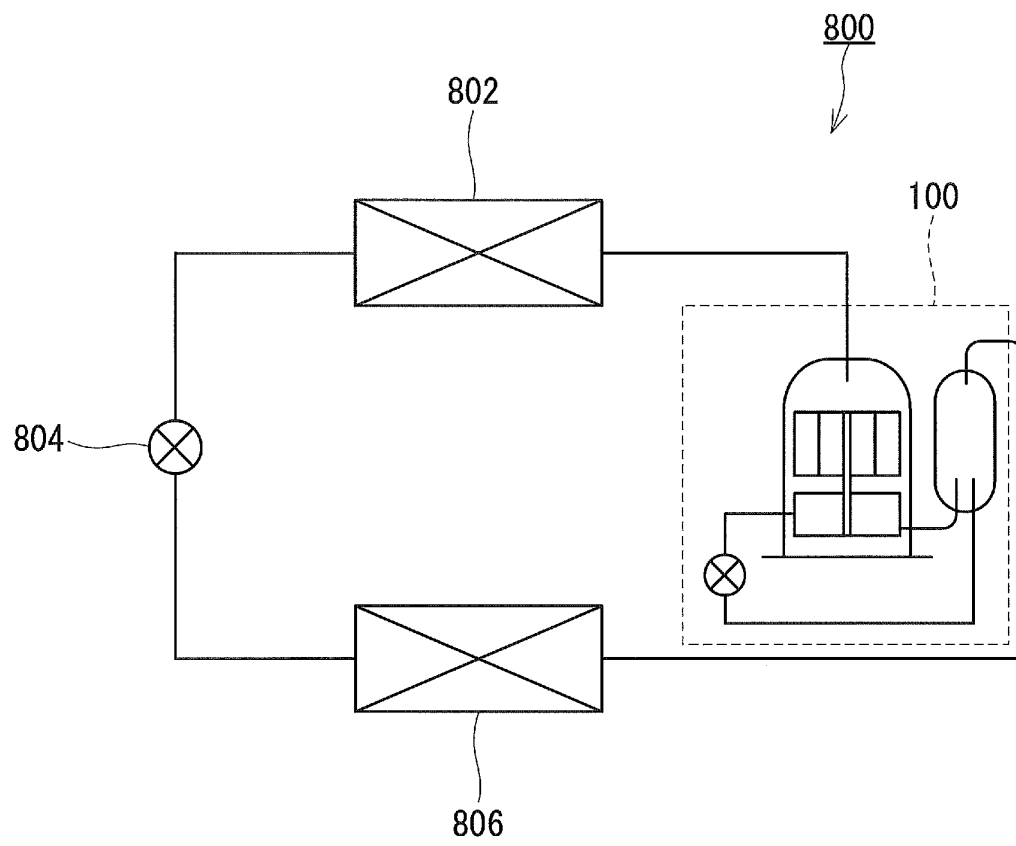
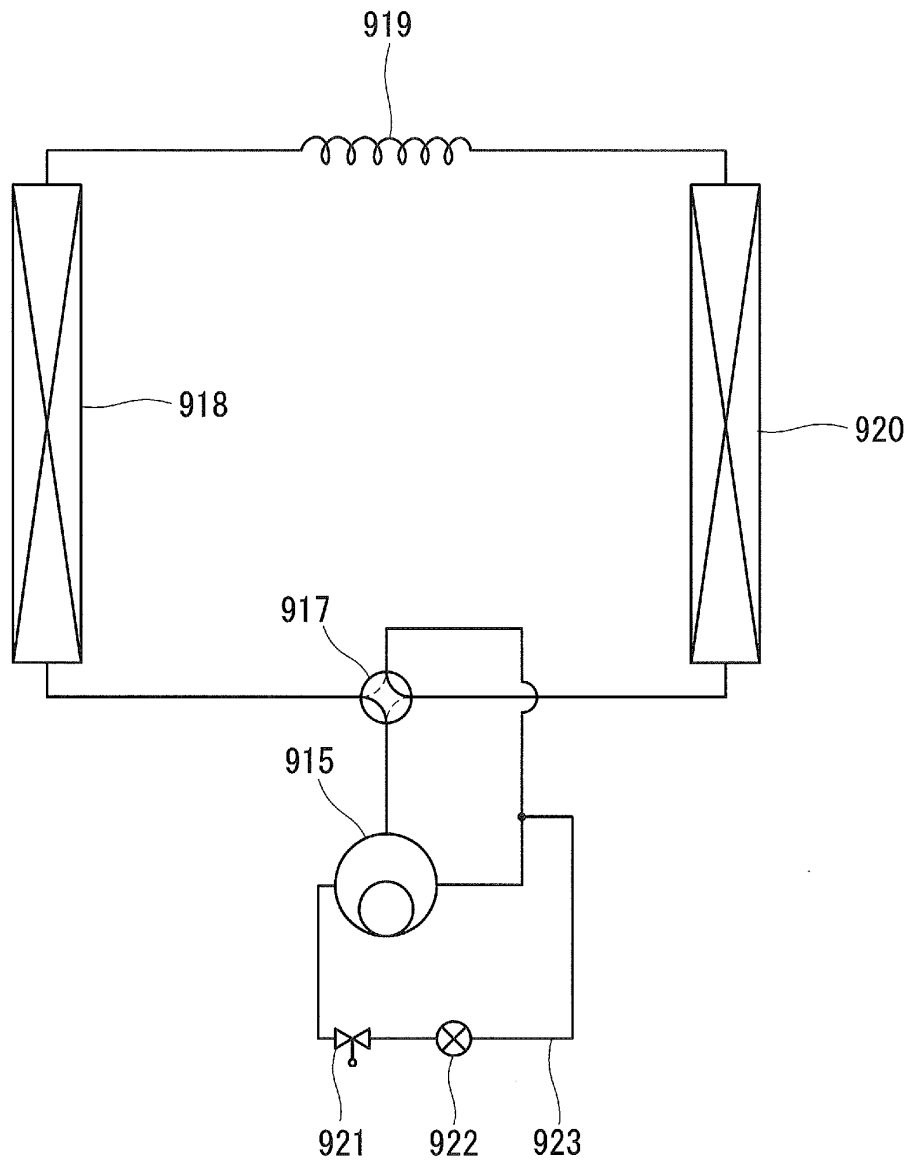


FIG.19



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2011/005522

A. CLASSIFICATION OF SUBJECT MATTER

F04C28/26(2006.01)i, F04C18/356(2006.01)i, F04C23/00(2006.01)i, F04C28/08(2006.01)i, F04C29/12(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)

F04C28/26, F04C18/356, F04C23/00, F04C28/08, F04C29/12

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

Jitsuyo Shinan Koho	1922-1996	Jitsuyo Shinan Toroku Koho	1996-2011
Kokai Jitsuyo Shinan Koho	1971-2011	Toroku Jitsuyo Shinan Koho	1994-2011

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.
X Y	JP 2-191882 A (Hitachi, Ltd.), 27 July 1990 (27.07.1990), entire text; all drawings (Family: none)	1-6, 8-12 7, 13-14
Y	JP 2-118362 A (Hitachi, Ltd.), 02 May 1990 (02.05.1990), page 6, lower left column, lines 3 to 5; fig. 8 to 9 & US 4989414 A	7, 13-14
Y	JP 5-172076 A (Mitsubishi Electric Corp.), 09 July 1993 (09.07.1993), entire text; all drawings (Family: none)	13-14

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

☐ See patent family annex.

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"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

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Date of the actual completion of the international search
06 December, 2011 (06.12.11)

Date of mailing of the international search report
13 December, 2011 (13.12.11)

Name and mailing address of the ISA/
Japanese Patent Office

Authorized officer

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INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2011/005522

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	Microfilm of the specification and drawings annexed to the request of Japanese Utility Model Application No. 115568/1986 (Laid-open No. 21792/1988) (Sanyo Electric Co., Ltd.), 13 February 1988 (13.02.1988), entire text; all drawings (Family: none)	7, 11-12

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Patent documents cited in the description

- JP S611986184365 A [0004]