



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

(43) Date of publication:
01.01.2014 Bulletin 2014/01

(51) Int Cl.:
F25B 1/00 (2006.01) F25B 1/04 (2006.01)

(21) Application number: **11859245.0**

(86) International application number:
PCT/JP2011/053839

(22) Date of filing: **22.02.2011**

(87) International publication number:
WO 2012/114454 (30.08.2012 Gazette 2012/35)

(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 MK MT NL NO PL PT RO RS SE SI SK SM TR

• **KOTANI, Masanao**
Tokyo 100-8220 (JP)
• **IJIMA, Ryota**
Tokyo 100-8220 (JP)

(71) Applicant: **Hitachi, Ltd.**
Tokyo 100-8280 (JP)

(74) Representative: **MERH-IP**
Matias Erny Reichl Hoffmann
Paul-Heyse-Strasse 29
80336 München (DE)

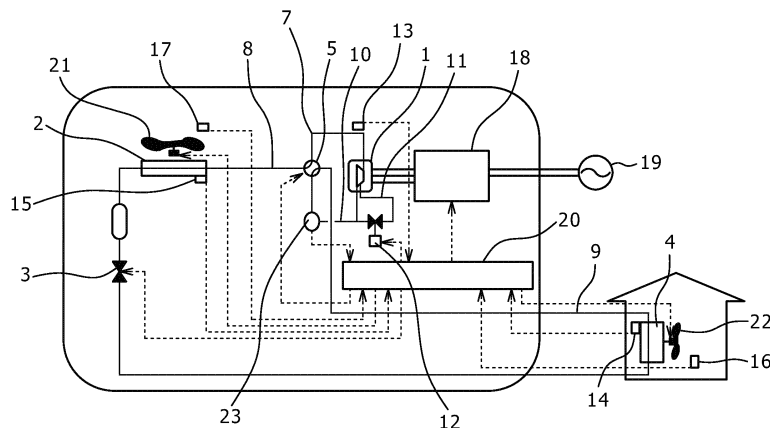
(72) Inventors:
• **KOYAMA, Masaki**
Tokyo 100-8220 (JP)

(54) **REFRIGERATION CYCLE APPARATUS**

(57) A refrigerating cycle system is provided with a compressor 1, an outdoor heat exchanger 2, an expansion valve 3 capable of controlling an opening thereof, and an indoor heat exchanger 4. The refrigerating cycle system is further provided with a bypass flow path 11 for causing a refrigerant in the middle of compression in the compressor to bypass toward a suction side of the compressor; a solenoid valve 12 for opening or closing the bypass flow path; and a controller 20 for controlling open state time of the solenoid valve, and closed state time thereof to adjust a flow rate of a refrigerant discharged

from the compressor into a refrigerating cycle, thereby executing a capacity-control. The controller executes a control on the basis of a duty ratio, that is, a ratio of the open time of the solenoid valve against a duty period representing the sum of the open time of the solenoid valve, and the closed time thereof, and further, if a pressure on the suction side of the compressor when the solenoid valve is in the state is higher than a suction pressure before the solenoid valve is in the state by a allowable deviation, the controller controls the solenoid valve so as to be in the closed state, the closed time being decided on the basis of the duty period.

FIG. 1



Description

Technical Field

[0001] The present invention relates to a refrigerating cycle system provided with a capacity-control compressor capable of executing a capacity-control. Further, the present invention is suitable for application to a refrigerating cycle system such as an air-conditioning water-heating system intended for a new-generation housing high in ecological (environmental responsiveness) effect, and is suitable for application to a refrigerating cycle system provided with a scroll compressor capable of carrying out an operation in a wide range, and efficiently performing a capacity-control even in an ultrasmall capacity-operation mode, in particular.

Background Art

[0002] From the standpoint of reducing energy consumed in common housing, that is, energy consumed by an air-conditioner, and energy consumed by a water heater, there has lately been seen an increasing tendency toward the use of material high in heat-insulating properties as a heat-insulating material in a building. Furthermore, there exists a concept of implementing a fossil-fuel zero-use housing whereby integrated power consumption for one year is zeroed by the installation of solar power generator, and soar heaters.

[0003] Under this concept, a refrigerating cycle system, such as an air-conditioner, a water heater, and so forth, is in use. For example, in the case of a scroll compressor, it is required that a capacity-control in a wide range can be realized by use of one unit. More specifically, in a cooling operation by use of an air-conditioner, a room air-temperature is generally high at the time of starting the operation, so that there is the need for a quick operation. In such a case, a high-speed operation (high-speed rotation) large in volume is executed at a startup, however, when the interior is cooled to some extent, and the operation shifts to a steady operation state, a low-speed operation (low-speed rotation) small in volume is executed. In a low-speed operation in the steady operation state, the operation will be carried out at a very low rotation speed assuming the case where an air-conditioning machine is installed in a building provided with a highly heat-insulating material, using latest energy-saving techniques, in particular.

[0004] However, if an excessively low-speed rotation is executed in the scroll compressor, this will structurally cause occurrence of rupture of an oil film at a plain bearing to thereby render the plain bearing susceptible to damage, rendering it difficult to perform stable operative actions since smooth motor-driving for rotating a crankshaft is prevented because of the low-speed rotation, and so forth. Accordingly, at the time of a mall-capacity operation, a general practice is adopted whereby the rotation speed is maintained to some extent in order to per-

form a capacity-control, and if, for example, the indoor is cooled to some extent, the scroll compressor is stopped while if a room air temperature rises, the scroll compressor is started again, this operation pattern being repeated.

[0005] However, the operation pattern for repeating stop and startup at the time of the mall-capacity operation is not only poor in efficiency but also incapable of performing comfortable air conditioning, there has been proposed a technique for devising a new capacity-control. In the case of executing a capacity-control with the use of the scroll compressor, there are generally adopted techniques including a technique for controlling a rotational speed by a motor rotation, a technique whereby a structure is partially modified to keep a rotation speed constant to thereby perform a control for rendering a discharge rate variable, or a technique for using these techniques in combination. Well known techniques for rendering a discharge volume variable include, for example, a scroll-type machine (refer to Patent Document 1) provided with a capacity-adjustment mechanism structured such that sealing in the axial direction of the crankshaft is released to stop compression, and an air-conditioner (refer to Patent Document 2) with a scroll compressor mounted therein, the scroll compressor provided with a capacity-adjustment mechanism whereby a refrigerant gas in the middle of compression is discharged toward a suction side to thereby delay the start of the compression.

[0006] In Patent Document 1, a high-pressure chamber, a discharge chamber, a low-pressure suction pipe that are formed between outer-shell connected fittings provided on one end side of the compressor, and a piston connected to a non-orbiting scroll are connected with each other via piping with a solenoid valve interposed therebetween, respectively, so that when the solenoid valve is turned ON (opened) by pulse width modulation (PWM) control, the high-pressure chamber is communicated with a low pressure suction pipe inside the piping, whereupon the non-orbiting scroll moves toward the crankshaft side, whereupon sealing is released in the axial direction of the crankshaft, thereby preventing compression. Then, when the solenoid valve is turned OFF (closed), the high-pressure chamber is connected with the discharge chamber inside the piping, whereupon the non-orbiting scroll moves toward the crankshaft side, on the opposite side of the outer-shell connected fittings, whereupon sealing is executed in the axial direction of the crankshaft, thereby performing a normal compression action.

[0007] The scroll-type machine according to Patent Document 1 is operated by turning the solenoid valve OFF (closed) at the time of a normal capacity-control while the solenoid valve is turned ON (open) at the time of the small capacity-control, and the refrigerant gas is returned to the suction pipe on the low pressure side to thereby adjust the discharge volume of the refrigerant gas, enabling capacity-control in a wide range of 0 to

100% to be implemented. As a result, this will enable a compression action at a small-capacity-control (the ultrasmall capacity-operation mode) corresponding to the case of an ultra-low speed operation at not higher than a low-limit set value of a motor rotational speed that cannot be actually realized owing to a problem of the rupture of the oil film at the sliding bearing, described as above, and torque change (in a drive signal to a motor, frequency is on the order of 5Hz), however, frequency in actual designing is set to a value higher than that, to a range of about 15 to 20Hz). The refrigerant gas subjected compression is guided to a refrigerating cycle via a discharge pipe, thereby enabling refrigerant gas to be slowly circulated.

[0008] In Patent Document 2, there are provided the scroll compressor having a bypass port, a flow path where the bypass port is open to a suction pressure atmosphere, a control valve for opening/closing the flow path, and a control means for opening/closing the control valve according to a plurality of control patterns based on short-period time allocation that is set according to an operation load of the air-conditioner.

In the case of the air-conditioner with the scroll compressor mounted therein, according to Patent Document 2, the refrigerant gas in the middle of compression is discharged into a suction chamber, and a containment volume upon suction completion is reduced, thereby enabling capacity-control by 60% to be realized. Further, an operation with a capacity-control in a range of 60 to 100% by stages is realized by opening/closing the control valve for discharging the refrigerant gas in the middle of compression into the suction chamber according to the plural control patterns based on the short-period time allocation.

Citation List

Patent Literature

[0009]

Patent Document 1: Japanese Unexamined Patent Application Publication No. Hei8(1996)-334094

Patent Document 2: Japanese Unexamined Patent Application Publication No. Hei11(1999)-324951

Summary of Invention

Technical Problem

[0010] In a capacity-control operation using a compressor for variably controlling a discharge volume, a discharge pressure, and a suction pressure undergo change due to the opening/closing of a control valve for capacity-control, such as the solenoid valve, and so forth. If ON (opening) - OFF (closing) cycle (a duty period) is large at the time of a pulse width modulation control (hereinafter referred to as PWM control), in particular, change

in a suction pressure is large, and if this capacity-control scheme is applied to an air-conditioner, a blow-off temperature will undergo variation, and comfortability can be no longer maintained. Furthermore, in the case of the capacity-control scheme described as above, losses occur at the time of the opening/closing of the control valve, so that if the duty period is rendered shorter, the losses will increase although the variation decreases, thereby causing deterioration in efficiency.

With the one according to Patent Document 1, capacity-adjustment is performed by subjecting the solenoid valve to the PWM control to be turned ON/OFF, and the capacity-control in a wide range can be performed. However, in Patent Document 1, it is simply described that the solenoid valve is subjected to the PWM control to vary an ON - OFF time (a duty period) for the purpose of adjustment to a target capacity, thereby effecting the capacity-control, however, no consideration is given to enhancement in comfortability while suppressing deterioration in efficiency.

[0011] In the case of the one according to Patent Document 2, a load state of a refrigerating cycle is detected on the basis of respective signals from a temperature sensor, and a pressure sensor, provided at the condenser, and the evaporator, respectively, thereby switching between a capacity-control operation, and a full-load operation. In the capacity-control operation, a bypass operation is executed according to predetermined time allocation. However, in the case of the one according to Patent Document 2, no consideration is given to enhancement in comfortability while suppressing deterioration in efficiency either as is the case with Patent Document 1.

[0012] It is therefore an object of the invention to obtain a refrigerating cycle system capable of not only realizing highly efficient operation-control even in an ultrasmall capacity-operation mode, but also realizing enhancement in comfortability.

40 Solution to Problem

[0013] In accordance with one aspect of the invention, there is provided a refrigerating cycle system provided with a compressor, an outdoor heat exchanger, an expansion valve capable of controlling an opening thereof, and an indoor heat exchanger, said refrigerating cycle system comprising a bypass flow path for causing a refrigerant in the middle of compression to bypass toward a suction side of the compressor, a solenoid valve for opening or closing the bypass flow path, and a controller for controlling open (ON) state time (τ_1) of the solenoid valve, and closed (OFF) state time (τ_2) thereof to adjust a flow rate of a refrigerant discharged from the compressor into a refrigerating cycle, thereby executing a capacity-control. The controller executes a control on the basis of a duty ratio (d) that is a ratio of open time of the solenoid valve against a duty period (T) representing the sum of the open time of the solenoid valve, and closed time

thereof, and further, if a pressure (P_s) on the suction side of the compressor when the solenoid valve is in the ON state is higher than a suction pressure (P_{s0}) before the solenoid valve is in the ON state by a allowable deviation (ΔP), the controller controls the solenoid valve so as to be in the closed state, the closed time being decided on the basis of the duty period.

[0014] The invention provides in its another aspect a refrigerating cycle system provided with a compressor, an outdoor heat exchanger, an expansion valve capable of controlling an opening thereof, and an indoor heat exchanger, said refrigerating cycle system comprising a bypass flow path for causing a refrigerant in the middle of compression to bypass toward a suction side of the compressor, a solenoid valve for opening or closing the bypass flow path; and a controller for controlling open (ON) state time (τ_1) of the solenoid valve, and closed (OFF) state time (τ_2) thereof to adjust a flow rate of a refrigerant discharged from the compressor into a refrigerating cycle, thereby executing a capacity-control. The controller executes a control on the basis of a duty ratio d , that is, a ratio of open time of the solenoid valve against a duty period (T) representing the sum of the open time of the solenoid valve, and closed time thereof, and further, if an evaporator temperature (T_{ev}) of the indoor heat-exchanger or the outdoor heat-exchanger (an evaporator-side heat-exchanger), serving as the evaporator, when the solenoid valve is in the ON state, becomes higher than an evaporator temperature (T_{ev0}) before the solenoid valve is in the ON state by a allowable deviation (ΔT_{ev}), the controller controls the solenoid valve so as to be in the closed state, the closed time being decided on the basis of the duty period.

Advantageous Effects of Invention

[0015] The present invention can provide a refrigerating cycle system capable of executing a highly efficient operation control even in an ultrasmall-capacity operation mode and improving comfortability.

Brief Description of Drawings

[0016]

Fig. 1 is a schematic block diagram showing a first embodiment of a refrigerating cycle system according to the invention;

Fig. 2 is a diagram for describing PWM control in the refrigerating cycle system, and change in an evaporating pressure;

Fig. 3 is a flow chart for describing a compressor rotational-speed control routine according to the first embodiment;

Fig. 4 is a flow chart for describing an expansion-valve opening-control routine in the refrigerating cycle system according to the first embodiment;

Fig. 5 is a schematic block diagram of a refrigerating

cycle system according to a second embodiment of the invention:

Fig. 6 is a flow chart for describing a compressor rotational-speed control routine in the case of the refrigerating cycle system according to the second embodiment of the invention;

Fig. 7 is a schematic block diagram of a refrigerating cycle system according to a third embodiment of the invention;

Fig. 8 is a diagram for describing PWM control, and change in an evaporator temperature in the case of a refrigerating cycle system according to a third embodiment of the invention;

Fig. 9 is a flow chart for describing a compressor rotational-speed control routine in the case of the refrigerating cycle system according to the third embodiment of the invention;

Fig. 10 is a longitudinal sectional view showing an example of a capacity-control compressor used in the present invention:

Fig. 11 is an expanded sectional view of the principal part of the scroll compressor shown in Fig. 10, for describing a refrigerant gas flow at the time of a normal operation; and

Fig. 12 is an expanded sectional view of the principal part of the scroll compressor shown in Fig. 10, for describing a refrigerant gas flow at the time of a bypass operation. Description of Embodiments

[0017] Specific embodiments of a refrigerating cycle system according to the invention are described hereinafter with reference to the accompanied drawings.

First Embodiment

[0018] A first embodiment of a refrigerating cycle system according to the invention is described with reference to Figs. 1 to 4. Fig. 1 is a schematic block diagram of the refrigerating cycle system, showing the first embodiment of the invention, representing the case where the present invention is applied to a room air-conditioner (an air-conditioner).

[0019] The refrigerating cycle system shown in Fig. 1, together with an operation at the time of a cooling operation, is described hereinafter. A refrigerant compressed by a compressor 1 repeats a circulation whereby the refrigerant flows from a high-pressure side connection piping 7 into a four-way valve 5, passes through the four-way valve 5 before flowing out to an outdoor connection piping 8. Thereafter, the refrigerant is subjected to heat exchange with an outdoor air in an outdoor heat exchanger 2 to release heat, thereby undergoing condensation liquefaction before being subjected to decompression by the agency of an expansion valve 3. The refrigerant that has turned low in temperature · pressure after subjected to the decompression enters an indoor heat exchanger 4 to cool an indoor air while the refrigerant itself undergoes evaporation · gasification before flowing from an

indoor connection piping 9 into the four-way valve 5 again, thereafter flowing out from a low-pressure side connection port of the four-way valve 5, and passing through a low-pressure side connection piping 10 before returning to the suction side of the compressor 1 to be compressed again.

[0020] Further, in the case of switching from the cooling operation to a heating operation, the connection destination of a piping for refrigerant from the four-way valve 5 is switched. At the time of the heating operation, a refrigerant high in temperature · high pressure, discharged from the compressor 101 (1), passes through the four-way valve 5 from the high-pressure side connection piping 7 to flow out to an indoor connection piping 9 before flowing to the indoor heat exchanger 4, thereby carrying out the heating operation by releasing heat to an indoor air while the refrigerant itself undergoes condensation. Thereafter, a circulation is repeated whereby the refrigerant subjected to the condensation undergoes decompression by the agency of an expansion valve 3 to be subjected to heat exchange with an outdoor air in an outdoor heat exchanger 2, thereby undergoing condensation · liquefaction before flowing from the outdoor connection piping 8 into the four-way valve 5, and subsequently flowing to the low-pressure side connection piping 10 before returning to the suction side of the compressor 1 to be compressed again.

[0021] Reference numeral 11 denotes a bypass piping (bypass flow path) for guiding a refrigerant gas at a discharge pressure to the suction side of the compressor 1, and the bypass piping 11 has one end connected to the low-pressure side connection piping 10 on the suction side of the compressor. The bypass piping 11 is provided with a solenoid valve 12 to be controlled so as to be in an open (ON) state, and a closed (OFF) state by the agency of a pulse width modulation (PWM) control signal such that communication between the bypass piping 11 and the low-pressure side connection piping 10 is turned ON/OFF.

[0022] For example, at the time of an ultralow-load operation mode (an ultrasmall-capacity operation mode), the solenoid valve 12 is caused to make a repetitive action so as to be in an open state, and a closed state to repeatedly cause ON/OFF of a discharge-side refrigerant flow into a suction side, thereby realizing a capacity-adjustment mechanism for carrying out a small-capacity control of the refrigerant discharged from the compressor into a refrigerating cycle.

[0023] Next, there is described a control system of the refrigerating cycle system shown in Fig. 1. Reference numeral 13 shown in Fig. 1 denotes a discharge-temperature sensor attached to a discharge-side piping (the high-pressure side connection piping 7) of the compressor 1, the discharge-temperature sensor being for detecting a discharge temperature of the refrigerant from the compressor (an inlet temperature of the refrigerant flowing into a condenser). Further, 14 indicates an indoor heat-exchanger temperature sensor installed at a sub-

stantially central position of the indoor heat exchanger 4, and this temperature sensor 14 is used for detection of an evaporation temperature of the refrigerant at the time of the cooling operation when the indoor heat exchanger 4 functions as an evaporator. Furthermore, 15 indicates an outdoor heat-exchanger temperature sensor installed at a substantially central position of the outdoor heat exchanger 2, and this temperature sensor 15 is used for detection of an evaporation temperature of the refrigerant at the time of the heating operation when the outdoor heat exchanger 2 functions as an evaporator. Still further, 16 indicates an indoor temperature sensor for detecting an air temperature inside a room where the indoor heat exchanger 4 is installed, and 17 indicates an outdoor temperature sensor for detecting an outdoor air temperature around the location of the outdoor heat exchanger 2.

[0024] Meanwhile, an inverter (a motor drive circuit) 18 is connected to the compressor 1, and the inverter 18 is connected to a commercial AC power supply source 19. The inverter 18 rectifies a voltage of the commercial AC power supply source 19 to be converted into a voltage at a frequency according to a command, thereby outputting the voltage to a motor installed inside the compressor 1. Further, the inverter 18 is connected to a controller 20, driving the motor on the basis of a command from controller 20. Further, the four-way valve 5, the expansion valve 3, an outdoor fan 21, an indoor fan 22, the indoor heat-exchanger temperature sensor 14, the outdoor heat-exchanger temperature sensor 15, the indoor temperature sensor 16, the outdoor temperature sensor 17, the discharge-temperature sensor 13, a suction pressure sensor 23, the inverter 18, a remote-control manipulator (not shown, hereinafter referred to as "remote controller"), and so forth are each connected to the controller 20 so that the refrigerating cycle system (the room air-conditioner) in its entirety is controlled by the controller 20.

[0025] Next, there is described hereinafter an operation of the refrigerating cycle system at the time of the cooling operation. At the time of starting the cooling operation, the controller 20 set the four-way valve 5 to a state for the cooling operation to drive the compressor 1, the outdoor fan 21, and the indoor fan 22 each at a specified rotational speed preset as an initial-value. The refrigerant discharged from the compressor 1 repeats a circulation whereby the refrigerant passes through the four-way valve 5, the outdoor heat exchanger 2, the expansion valve 3, and the indoor heat exchanger 4 in sequence before passing through the four-way valve 5 again to return to the compressor 1, whereupon the cooling operation is carried out. The expansion valve 3 is made up of, for example, an electronic expansion valve, and a pulse motor incorporated in the electronic expansion valve is rotated such that the expansion valve 3 is opened at a predetermined initial opening. At the time of the cooling operation, the indoor heat exchanger (heat exchanger being used) 4 functions as an evaporator.

[0026] With the room air-conditioner serving as the refrigerating cycle system, a room air-temperature is detected by the indoor temperature sensor 16 provided in the vicinity of the inlet of a ventilation passage of the indoor heat exchanger 4, and the controller 20 controls the inverter 18 according to a difference from a set temperature set by the remote controller to render the rotational speed of the compressor variable. Thereby, an operation of the compressor 1 is carried out corresponding to an air-conditioning load.

[0027] Further, a sensed temperature (discharge refrigerant temperature) of the discharge-temperature sensor 13 is detected once in every predetermined control time periods, and the opening of the expansion valve 3 is controlled once in every control time periods described as above according to a difference between the sensed temperature and a target discharge temperature dependent on a sensed temperature (an evaporation temperature) of the outdoor heat-exchanger temperature sensor 14, a detection temperature (a room air temperature) of the indoor temperature sensor 16, the rotational speed of the compressor 1 and a rotational speed command value of the outdoor fan 21. By virtue of this discharge superheat control, control can be implemented such that a suction superheat on the suction side of the compressor 1 is rendered nearly zero, thereby maintaining excellent coefficient of performance of the refrigerating cycle system.

[0028] On the other hand, if the discharge refrigerant temperature sensed by the discharge-temperature sensor 13 rises to a set value, or higher, an operation frequency of the compressor 1 is reduced until the sensed temperature falls to a predetermined set value and the opening of the expansion valve 3 is controlled so that the discharged refrigerant temperature will be at the set value. This discharge temperature control prevents the compressor 1 from being abnormally heated, thereby preventing breakdown of the compressor 1 due to a problem such as seizure, and so forth.

[0029] Next, there is described hereinafter an operation of the refrigerating cycle system at the time of the heating operation. At the time of the heating operation, the controller 20 switches the four-way valve 5 over to a heating operation side, thereby operating the compressor 1, the outdoor fan 21, and the indoor fan 22, respectively, at a specified rotational speed preset as an initial-value. The refrigerant discharged from the compressor 1 repeats a circulation whereby the refrigerant passes through the four-way valve 5, the indoor heat exchanger 4, the expansion valve 3, and the outdoor heat exchanger 2 in sequence before passing through the four-way valve 5 again to return to the compressor 1, whereupon the heating operation is carried out. At the time of the heating operation, the indoor heat exchanger (the heat exchanger being used) 4 functions as a condenser.

[0030] The controller 20 detects a difference between the set temperature set by the remote controller and the room air temperature sensed by the indoor temperature

sensor 16 as an air-conditioning load, controlling an operation frequency (an output frequency of the inverter 18) of the compressor according to the air-conditioning load. Thereby, an operation of the compressor 1 is carried out corresponding to a heating load.

[0031] Further, a discharge refrigerant temperature is detected by the discharge-temperature sensor 13 once in every predetermined control time periods, and the opening of the expansion valve 3 is controlled once in every control time periods described as above according to a difference between the discharge refrigerant temperature as sensed and a target discharge temperature dependent on a sensed temperature (an evaporation temperature) of the outdoor heat-exchanger temperature sensor 15, a detection temperature (an outdoor air temperature) of the outdoor temperature sensor 17, the rotational speed of the compressor 1 and the rotational speed command value of the outdoor fan 21. By virtue of this discharge superheat control, control can be implemented such that the suction superheat on the suction side of the compressor 1 is rendered nearly zero, thereby maintaining excellent coefficient of performance of the refrigerating cycle system.

[0032] On the other hand, if the discharge refrigerant temperature sensed by the discharge-temperature sensor 13 rises to the set value, or higher, the operation frequency of the compressor 1 is reduced until the sensed temperature falls to the predetermined set value, and the opening of the expansion valve 3 is controlled so that the discharged refrigerant temperature will be at the set value. The discharge temperature control prevents the compressor 1 from being abnormally heated, thereby preventing breakdown of the compressor 1 due to the problem such as seizure, and so forth.

[0033] Next, there is described a control using a capacity-adjustment mechanism for performing a small-capacity control of the refrigerant discharged from the compressor into the refrigerating cycle at the time of the ultralow-load operation, that is, the ultralow-load operation mode (the ultrasmall-capacity operation mode). With the capacity-adjustment mechanism for executing an ultrasmall-capacity control in the ultralow-load operation mode, the solenoid valve 12 provided in the bypass piping 11 is controlled so as to in the open (ON) state, and the closed (OFF) state by the pulse width modulation (PWM) control, thereby performing a capacity-adjust operation.

[0034] At the time when the solenoid valve 12 is in the open state, a check valve 121 (refer to Fig. 10) provided at an outlet of the compressor 1 is closed, and a discharge refrigerant gas passes through the bypass piping 11 to flow to a low-pressure side connection piping (a suction pipe). For this reason, the refrigerant does not flow toward the four-way valve 5, and a flow rate of the refrigerant in the refrigerating cycle decreases, thereby causing a decrease in capacity. On the other hand, if the solenoid valve 12 is set in the closed state, it is possible to cause the discharge refrigerant gas from the compressor to flow toward the four-way valve 5.

[0035] Accordingly, at the time of an operation in the ultralow-load operation mode, when the capacity-adjustment mechanism is activated, the solenoid valve 12 is caused to make a repetitive action so as to be in the open state, or the closed state to thereby repeat opening/closing of the bypass piping 11, rendering it possible to execute capacity-adjustment.

[0036] Now, the state of change in an evaporating pressure at the time when the solenoid valve 12 is under the PWM control is described hereinafter with reference to Fig. 2. If the PWM control signal is turned ON (the solenoid valve 12 is in the open state), the evaporating pressure rises. Further, if the PWM control signal is turned OFF (the solenoid valve 12 is in the closed state), the evaporating pressure falls. Thus, as the solenoid valve 12 is turned ON/OFF, variation (change) in the evaporating pressure is repeated.

[0037] At the time of a capacity-adjustment operation making use of the bypass piping 11, a variation width of an evaporating pressure is defined $\Delta P1$. Further, at the time of the capacity-adjustment operation, the evaporating pressures as a whole rises, and a mean pressure of various evaporating pressures rises by $\Delta P2$ against the evaporating pressure prior to the capacity-adjustment operation. If $\Delta P1$ at the time of the capacity-adjustment operation is large, the evaporating pressure subsequently will undergo variation to thereby cause variation in a heat exchange amount in the evaporator, so that the capacity of the refrigerating cycle system undergoes variation, thereby causing occurrence of variation in a blow-out temperature. For this reason, $\Delta P1$ is preferably reduced in order to maintain comfortable air-conditioning. Further, if $\Delta P2$ is large, endothermic energy will decrease, so that the heat exchange amount, as well, will decrease. In order to render both $\Delta P1$, and $\Delta P2$ smaller, it need only be sufficient to reduce a duty period $T (= \tau1 + \tau2)$ of the PWM control signal. However, there occur energy losses because of a back flow of the discharge refrigerant gas, losses in the pressure of the bypass piping 11, and so forth, occurring at the time of opening/closing the solenoid valve 12, in a PWM capacity-control operation. For this reason, in order to efficiently perform the capacity-adjustment operation, it is preferable not to reduce the duty period T .

[0038] Accordingly, with the present embodiment, in order to realize air-conditioning high in efficiency, and small in variation (change) of the evaporating pressure, an adequate duty period is decided in accordance with flow charts shown in Figs. 3 and 4, respectively, thereby effecting control of the capacity-adjustment operation.

[0039] First, referring to a flow chart shown in Fig. 3, a compressor rotational-speed control routine is described. As for the rotational speed of the compressor, a room air temperature Tea_{in} detected by the indoor temperature sensor 16 provided in the vicinity of the inlet of the ventilation passage of the indoor neat exchanger 4, as previously described, is read (step 31), a difference ΔTea_{in} from the set temperature (a room air temperature

target value) Tea_{in}^* set by the remote controller is found (step 32), and the rotational speed of the compressor 1 is rendered variable according to the difference ΔTea_{in} by the agency of the inverter 18 (step 33, 34). In this case, the control is executed such that the smaller the difference between the set temperature and the room air temperature as detected is, the smaller a compressor rotational speed f_z becomes.

[0040] In step 35, if the compressor rotational speed f_z becomes smaller than a rotational speed f_{zopt} at the time of starting the capacity-control operation, the compressor rotational speed is fixed to f_{zopt} , deciding an initial duty ratio $d \{d = \tau1 / (\tau1 + \tau2)\}$ determined from a difference between the room air temperature and the room air temperature target value (steps 36, 37), whereupon the PWM capacity-control operation for turning the solenoid valve 12 ON/OFF is performed. At this point in time, upon the PWM control signal being turned ON, a controller timer as well is turned ON, thereby starting to count elapsed-time $\tau1$. Further, the suction pressure sensor 23 starts measuring a suction pressure, repeating a measurement on the pressure, while the PWM control signal remains in the ON state until a measured suction pressure Ps exceeds a suction pressure $Ps0$ before the PWM control signal is turned ON by a preset allowable deviation ΔP (steps 38 to 41). If a difference between the measured suction pressure Ps and the initial suction pressure $Ps0$ exceeds ΔP , the PWM control signal is turned OFF, the solenoid valve 12 is turned into the closed state, and the timer is turned OFF, whereupon counting of the elapsed-time is completed, deciding $\tau1$ as opening time (step 42). On the basis of the opening time $\tau1$, and a present duty ratio d , closing time $\tau2$ is decided, and the PWM capacity-control operation based on this duty period is performed (step 43). Thereby, change in the suction pressure, caused by the opening/closing of the solenoid valve 12, can be decided so as to fall within ΔP , so that an operation based on an optimum duty period is enabled by setting the ΔP in such a range as not to impair comfortability.

[0041] Fig. 4 is a flow chart for describing an expansion-valve opening-control routine in the refrigerating cycle system according to the present embodiment. Upon the start of the PWM capacity-control operation according to the compressor rotational-speed control routine as described with reference to Fig. 3, the evaporation temperature rises and the evaporating pressure become larger by $\Delta P2$ (refer to Fig. 2) against the evaporating pressure prior to the capacity-control operation. In order to render the $\Delta P2$ as small as possible, the control of the opening of the expansion valve 3 is performed.

[0042] First, in step 45, a state quantity of a refrigerating cycle is read. More specifically, the room air temperature, the outdoor temperature, temperature of the indoor heat exchanger, temperature of the outdoor heat exchanger, and so forth, detected by the various sensors, respectively, are each read, and further, the rotational speed of the compressor 1, the rotational speed of the

outdoor fan 21 as well as the indoor fan 22, the opening of the expansion valve 3, and so forth, as well, are read. Then, when the PWM control signal is OFF (in step 46), the opening of the expansion valve 3 is controlled such that the sensed temperature (the discharge refrigerant temperature) T_d of the discharge-temperature sensor 13 will approach the target discharge temperature T_d^* determined from a sensed temperature (condensation temperature) T_{ao} of the outdoor heat-exchanger temperature sensor 15, a detection temperature (an outdoor air temperature) T_{ai} of the outdoor temperature sensor 17, the rotational speed f_z of the compressor 1, and the rotational speed command value f_p of the outdoor fan 21 (steps 47 to 51).

[0043] If the PWM control signal is ON in the step 46, a corrected compressor rotational speed f_z' , obtained by dividing the rotational speed $f_{z,opt}$ at the time of starting the capacity-control operation by a duty ratio "d" at that point in time is decided (step 52), thereupon switching is made to a routine whereby the expansion valve 3 is controlled such that the sensed temperature (the discharge refrigerant temperature) T_d of the discharge-temperature sensor 13 will approach the target discharge temperature T_d^* determined from the corrected compressor rotational speed f_z' , the sensed temperature (condensation temperature) T_{ao} of the outdoor heat-exchanger temperature sensor 15, the detection temperature (the outdoor air temperature) T_{ai} of the outdoor temperature sensor 17, and the rotational speed command value f_p of the outdoor fan 21 (steps 52 to 57). At the time of the PWM capacity-control operation, the larger the duty ratio d is, the less a refrigerant circulation amount becomes, so that the opening of the expansion valve 3 can be changed to an adequate opening even against a reduced refrigerant circulation amount at the time of the PWM capacity-control operation, and ΔP_2 can be prevented from rising.

[0044] With the refrigerating cycle system according to the present embodiment, variation (change) in the suction pressure (the evaporating pressure), due to the opening/the closing of the solenoid valve 12, can be decided so as to be in a range based on the allowable deviation ΔP , so that it is possible to suppress the variation (change) in the suction pressure to fall within a given range, thereby enabling comfortability in air-conditioning, and so forth to be enhanced. Further, since it is possible to prevent occurrence of an increase in losses, caused by an excessively shortened duty period, a highly efficient capacity-control operation can be realized. In addition, an advantageous effect of realizing the capacity-control in the wide range of 0 to 100% by use of a simple structure can be obtained.

[0045] Thus, with the present embodiment of the invention, it is possible to obtain a refrigerating cycle system capable of realizing an operation control excellent in efficiency even in an ultrasmall volume-operation mode, and enhancing comfortability as well.

Second Embodiment

[0046] Fig. 5 is a schematic block diagram of a refrigerating cycle system, showing a second embodiment of the invention, representing the case where the present invention is applied to the room air-conditioner as is the case with the first embodiment. In Fig. 5, parts denoted by like reference numerals shown in Fig. 1 are parts identical, or corresponding thereto. The second embodiment differs from the first embodiment in that the suction pressure sensor is removed, and a blow-out temperature sensor 24, in place of the suction pressure sensor, is installed in the vicinity of the outlet of a ventilation passage of the indoor heat exchanger 4, thereby enabling a blow-out temperature to be detected by the blow-out temperature sensor 24.

[0047] In the case of executing control by use of the capacity-adjustment mechanism for small-capacity control, upon the PWM control signal being turned ON (that is, the solenoid valve 12 is in the open state), an evaporating pressure rises, while upon the PWM control signal being turned OFF (that is, the solenoid valve 12 is in the closed state), the evaporating pressure falls. At this point in time, an evaporation temperature as well undergoes variation (change), and a heat exchange amount in an evaporator undergoes variation (change), so that a refrigerating cycle undergoes variation (change), thereby causing occurrence of variation (change) in the blow-out temperature, and a refrigeration capacity. For this reason, variation (change) in the evaporation temperature can be estimated on the basis of the temperature of an evaporator-side heat exchanger (the indoor heat exchanger 4 at the time of the cooling operation, and an outdoor heat exchanger 2 at the time of the heating operation) and the blow-out temperature of the indoor heat exchanger 4, measured by the blow-out temperature sensor 24.

[0048] Referring to Fig. 6, there is described hereinafter a compressor rotational-speed control routine in the case of the refrigerating cycle system according to the second embodiment of the invention. As for the rotational speed of a compressor 1, the room air temperature Tea_{in} detected by the indoor temperature sensor 16 provided in the vicinity of the inlet of the ventilation passage of the indoor heat exchanger 4, as previously described, is read (step 31), the difference ΔTea_{in} from the set temperature (the room air temperature target value) Tea_{in}^* set by the remote controller is found (step 32), and the rotational speed of the compressor 1 is rendered variable according to the difference ΔTea_{in} by the agency of the inverter 18 (steps 33, 34). In this case, the control is executed such that the smaller the difference between the set temperature and the room air temperature as detected becomes, the smaller the compressor rotational speed f_z becomes.

[0049] In step 35, if the compressor rotational speed f_z becomes smaller than a rotational speed $f_{z,opt}$ at the time of starting the capacity-control operation, the com-

pressor rotational speed is fixed to f_{zopt} , and an initial duty ratio decided from a difference between the room air temperature and the room air temperature target value is determined (steps 36, 37), thereby executing the PWM capacity-control operation for turning the solenoid valve 12 ON/OFF. At this point in time, upon the PWM control signal being turned ON, a controller timer as well is turned ON, thereby starting to count elapsed-time τ_1 . Further, measurement on an evaporator-side heat exchanger temperature T_{ev0} , by use of the heat exchanger temperature sensor on the evaporator-side (reference numeral 14, or 15), is started (step 61). Further, an allowable deviation ΔT_{ev} is worked out according to a table for holding the evaporator-side heat exchanger temperature T_{ev0} at the time of starting the measurement, and a blow-out temperature $T_{ea_{out}}$ of the indoor heat exchanger 4, detected by the blow-out temperature sensor 24, as preset controlled parameters (step 62). The PWM control signal remains in the ON state until a measured evaporator-side heat exchanger temperature T_{ev} exceeds the allowable deviation ΔT_{ev} , as compared with the evaporator-side heat exchanger temperature T_{ev0} before the PWM control signal is turned ON, and the measurement on the evaporator-side heat exchanger temperature T_{ev} is repeated (steps 63 to 65). If a difference between the measured evaporator-side heat exchanger temperature T_{ev} and the initial evaporator-side heat exchanger temperature T_{ev0} exceeds the allowable deviation ΔT_{ev} , the PWM control signal is turned OFF, and the solenoid valve 12 will be in the closed state, whereupon the timer will be turned OFF, completing counting of the elapsed time, and τ_1 is determined as open time. On the basis of τ_1 , and the present duty ratio d , closed time τ_2 is determined, and the PWM capacity-control operation based on this duty period is performed (steps 66, 67).

[0050] Meanwhile, the opening-control of an expansion valve 3 is executed according to the same routine as the expansion-valve opening control routine executed in the first embodiment, as shown in Fig. 4.

With the present embodiment, variation (change) in the suction pressure, due to the opening/closing of the solenoid valve 12, can be decided so as to fall within a range based on the allowable deviation ΔT_{ev} , so that if the allowable deviation ΔT_{ev} is set within an appropriate range, it becomes possible to suppress the variation (change) in the suction pressure to be within a given range even without the pressure sensor for measuring the evaporating pressure (the suction pressure sensor), so that it is possible to realize a refrigerating cycle system that can be manufactured at a lower cost, and can execute a capacity-control operation not only excellent in air-conditioning properties, but also high in efficiency.

Third Embodiment

[0051] Fig. 7 is a schematic block diagram of a refrigerating cycle system, showing a third embodiment of the invention, representing the case where the present in-

vention is applied to the room air-conditioner, as is the case with the first embodiment and the second embodiment, respectively. In Fig. 7, parts denoted by like reference numerals shown in Fig. 1 and Fig. 5, respectively are parts identical or corresponding thereto. The third embodiment differs from the first embodiment, and the second embodiment, respectively, in that the suction pressure sensor 23, as shown in the first embodiment, and the blow-out temperature sensor 24 installed in the vicinity of the outlet of the ventilation passage of the indoor heat exchanger 4, as shown in the second embodiment, are removed.

[0052] In the case of executing the control by use of the capacity-adjustment mechanism for effecting the small-capacity control, upon the PWM control signal being turned ON (that is, the solenoid valve 12 is in the open state), an evaporating pressure rises, while upon the PWM control signal being turned OFF (that is, the solenoid valve 12 is in the closed state), the evaporating pressure falls. At this point in time, an evaporation temperature as well undergoes change, and a heat exchange amount in an evaporator undergoes change, so that a refrigerating cycle undergoes change, thereby causing occurrence of change in the blow-out temperature, and a refrigeration capacity. For this reason, change in the evaporation temperature is estimated by measuring the temperature of an evaporator-side heat exchanger (an indoor heat exchanger 4 at the time of the cooling operation, and an outdoor heat exchanger 2 at the time of the heating operation), whereupon the control can be executed.

[0053] At the time of a normal operation, the evaporator-side heat exchanger is operated so as to effect the discharge superheat control such that a suction superheat at the outlet thereof is rendered nearly zero, that is, dryness 1, as previously described. As dryness at the inlet of the evaporator-side heat exchanger is normally in a range of about 0.1 to 0.3, the evaporator-side heat exchanger has a distribution such that dryness gradually increases from the inlet toward the outlet inside the heat exchanger. At the time of the capacity-control operation, a circulation amount of the refrigerant decreases, and an amount of the refrigerant flowing out of the evaporator-side heat exchanger will decrease against an amount of the refrigerant flowing into the evaporator-side heat exchanger, so that the evaporating pressure rises, and the evaporation temperature rises, however, a liquid refrigerant undergoes phase change from a liquid phase to a gas phase, and dryness gradually increases. Accordingly, inside the evaporator-side heat exchanger, the refrigerant gradually becomes drier starting from the outlet side of the heat exchanger, a heat exchange amount becomes extremely smaller starting from a point closer to the outlet side. If the heat-exchanger temperature sensor (the reference numeral 14, or 15) is installed in the vicinity of the center of the relevant heat-exchanger, the evaporator-side heat exchanger temperature (the evaporator temperature) T_{ev} measured by the sensor will be-

have as shown in Fig. 8. More specifically, if the PWM control signal is turned ON, the evaporator temperature T_{ev} will gently rises, and the heat exchanger gradually becomes drier starting from the outlet side thereof, so that as the neighborhood of an installation position of the heat-exchanger temperature sensor (the reference numeral 14, or 15) becomes drier, a measured temperature abruptly rises. For this reason, a dryness distribution inside the evaporator can be grasped on the basis of a temperature measurement position inside the evaporator-side heat exchanger. Accordingly, if timing for turning the PWM control signal OFF is determined as when the allowable deviation ΔT_{ev} of the evaporator temperature is at a value after the occurrence of an abrupt increase in temperature, this will enable the duty period T to be decided according to a degree of dryness at the temperature measurement position.

[0054] With the present embodiment, an installation position of the heat-exchanger temperature sensor (the reference numeral 14, or 15) is in the vicinity of the center of the heat exchanger, however, the installation position is preferably selected as appropriate so as to be in a range where change in the air-conditioning capacity is adequately permissible.

[0055] A compressor rotational-speed control routine in the case of the refrigerating cycle system according to the third embodiment of the invention is described with reference to Fig. 9. As for the rotational speed of a compressor 1, a room air temperature $T_{ea_{in}}$ detected by the indoor temperature sensor 16 provided in the vicinity of the inlet of the ventilation passage of the indoor heat exchanger 4, as previously described, is read (step 31), a difference $\Delta T_{ea_{in}}$ from the set temperature (the room air temperature target value) $T_{ea_{in}^*}$ set by the remote controller is found (step 32), and the rotational speed of the compressor 1 is rendered variable according to the difference $\Delta T_{ea_{in}}$ by the agency of the inverter 18 (steps 33, 34). In this case, the control is executed such that the smaller the difference between the set temperature and the room air temperature as detected becomes, the smaller the compressor rotational speed f_z becomes.

[0056] In step 35, if the compressor rotational speed f_z becomes smaller than a rotational speed f_{zopt} at the time of starting the capacity-control operation, the compressor rotational speed is fixed to f_{zopt} , deciding an initial duty ratio determined from a difference between the room air temperature and the room air temperature target value (steps 36, 37), whereupon the PWM capacity-control operation for turning the solenoid valve 12 ON/OFF is performed. At this point in time, as the PWM control signal is turned ON, a controller timer is concurrently turned ON, thereby starting to count elapsed-time τ_1 . Further, measurement of an evaporator-side heat exchanger temperature (evaporator temperature) T_{ev0} , by use of the evaporator-side heat exchanger (reference numeral 14, or 15), is started (step 61). Further, an allowable deviation ΔT_{ev} is worked out according to a table for holding the evaporator-side heat exchanger temper-

ature T_{ev0} at the time of starting the measurement, and an air temperature T_{ai} , or T_{ao} , measured by an indoor temperature sensor 16, or an outdoor temperature sensor 17, provided in the vicinity of the inlet of a ventilation passage of the indoor heat exchanger 4, or the outdoor heat exchanger 2, serving as the evaporator, respectively, as preset controlled parameters.

This allowable deviation ΔT_{ev} is set to such a value as reached after the occurrence of an abrupt rise in temperature due to the heat exchanger being dried (dryness of the refrigerant becoming greater) at a temperature measurement position where the heat-exchanger temperature sensor 14, or 15 is installed (step 68). The PWM control signal remains in the ON state until a measured evaporator-side heat exchanger temperature T_{ev} exceeds the allowable deviation ΔT_{ev} , as compared with the evaporator-side heat exchanger temperature T_{ev0} before the PWM control signal is turned ON, and a measurement on the evaporator-side heat exchanger temperature T_{ev} is repeated (steps 63 to 65). If a difference between the measured evaporator-side heat exchanger temperature T_{ev} and the initial evaporator-side heat exchanger temperature T_{ev0} exceeds the allowable deviation ΔT_{ev} , the PWM control signal is turned OFF, and the solenoid valve 12 will be in the closed state, whereupon the timer will be turned OFF, completing counting of the elapsed-time is, and τ_1 is decided as opening time. On the basis of τ_1 , and the present duty ratio d , closing time τ_2 is decided, and the PWM capacity-control operation based on this duty period is performed (steps 66, 67).

[0057] Meanwhile, the opening-control of an expansion valve 3 is executed according to the same routine as the expansion-valve opening control routine executed in the first embodiment, as shown in Fig. 4.

With the present embodiment, change in the suction pressure (the evaporating pressure), due to the opening/closing of the solenoid valve 12, can be decided so as to fall within a range based on the allowable deviation ΔT_{ev} , so that if the allowable deviation ΔT_{ev} is set within an appropriate range, it becomes possible to suppress the change in the suction pressure to be within a given range even without the suction pressure sensor 23 for measuring the evaporating pressure (refer to Fig. 1), and the blow-out temperature sensor 24 (refer to Fig. 5) for measuring the blow-out temperature of air into a room, so that it is possible to realize a refrigerating cycle system that can be manufactured at a still lower cost, and can execute a capacity-control operation not only excellent in air-conditioning properties, but also high in efficiency.

[0058] Next, there is described hereinafter an example of the capacity-control compressor used in the refrigerating cycle system according to the respective embodiments of the invention. Fig. 10 is a longitudinal sectional view showing a scroll compressor as an example of the capacity-control compressor used in the present invention, Fig. 11 an expanded sectional view of the principal part of the scroll compressor shown in Fig. 10, for describing a refrigerant gas flow at the time of a normal

operation (at the time of an operation mode when the solenoid valve 12 of the capacity-adjustment mechanism is a closed state), and Fig. 12 an expanded sectional view of the principal part of the scroll compressor shown in Fig. 10, for describing a refrigerant gas flow at the time of a bypass operation (at the time of an operation mode when the solenoid valve 12 of the capacity-adjustment mechanism is in an open state).

[0059] A scroll compressor 1 is provided with a sealed chamber 115, having a suction pipe 113 for taking in a refrigerant gas, and a discharge pipe 114 for discharging a depressed refrigerant gas, the closed casing 115 incorporating a compression mechanism part comprised of a fixed scroll 102 having a spiral lap, and an orbiting scroll 101 having such a spiral lap as to be engaged with the spiral lap of the fixed scroll 102. Further, a motor 100 comprised of a rotor 100a, and a stator 100b is provided below the compression mechanism part, and a crankshaft 106 serving as a rotational main shaft is integrally linked with the rotor 100a. The crankshaft 106 is rotatively supported by a main bearing 105a provided in a frame 105, and a secondary bearing 112 provided in a lower frame 111 in a lower part inside the closed casing 115. A sliding bearing 130 is provided on a back face of the orbiting scroll 101, and an eccentric part 106a provided on the upper end side of the crankshaft 106 is inserted in the sliding bearing 130. Reference numeral 107 denotes an Oldham coupling ring (a rotation-preventive member), and upon the crankshaft 106 being rotated, the orbiting scroll 101 is set in a swing motion by the agency of this Oldham coupling ring 107 without undergoing rotation, thereby compressing the refrigerant gas taken in from the suction pipe 113.

[0060] The spiral laps provided on the respective end plates of the orbiting scroll 101, and the fixed scroll 102 are made up to form asymmetrical laps in which respective winding angles of the spiral laps differ from each other, so that respective compression chamber formed on the external line side, and the inner line side of the orbiting scroll lap by engaging the orbiting scroll 101 with the fixed scroll 102 are in the shape of an asymmetrical scroll where the two compression chambers differ in maximum stroke volume from each other.

[0061] That is, the respective spiral laps formed by respective involute curves of the orbiting scroll 101, and the fixed scroll 102 are engaged with each other, and the respective compression chamber are formed on the outer line side of the lap on the end side of the orbiting scroll 101, and the inner line side of the lap on the winding end side thereof. However, the compression chamber formed on the outer line side differs in size from the compression room formed on the inner line side, and these compression chambers each are formed in such a way as to be out of phase with the axial rotation of the crankshaft by about 180 degrees.

[0062] More specifically, a discharge port 108 is open near the center of the fixed scroll 102, and the end of the spiral lap thereof, on the inner line side, is extended about

180 degrees up to the vicinity of the end of the orbiting scroll 101. Accordingly, when the respective laps of the orbiting scroll 101, and the fixed scroll 102, are combined with each other to form the compression rooms, a first compression room formed due to confinement with the outer line side of the spiral lap of the orbiting scroll 101, and the inner line side of the spiral lap of the fixed scroll 102 differs in size from a second compression room formed due to confinement with the inner line side of the spiral lap of the orbiting scroll 101, and the outer line side of the spiral lap of the fixed scroll 102, so that the first compression room, and the second compression room are formed in such a way as to be out of phase with the axial rotation of the crankshaft by about 180 degrees.

[0063] Further, with the scroll compressor, a release port 125 communicating with the compression room is formed on the outer peripheral side of the discharge port 108, and the release port 125 is provided with a release valve 124 serving as an excessive-compression relief valve. A discharge head cover 118 attached to a top plate (an end plate top-face) of the fixed scroll 102 covers both the discharge port 108, and the release valve 12, to thereby form a discharge head space 123, and further, the discharge head cover 118 is provided with a discharge valve 121 functioning as a check valve for opening/closing a through-hole 119 formed at a predetermined location.

[0064] The bypass piping 11 is for use in guiding the refrigerant gas in the discharge head space 123 to outside the closed casing 115, and one end of the bypass piping 11 is connected to the discharge head cover 118, the bypass piping 11 being extended through the closed casing 115 before the other end thereof is drawn out of the closed casing 115. The other end of the bypass piping 11 is connected with the suction pipe 113 for taking in the refrigerant gas, and further, the bypass piping 11 is provided with a solenoid valve 12 at some midpoint on its way from the closed casing 115. The solenoid valve 12 is made up so as to be drive-controlled in the open state, or the closed state by the agency of the pulse width modulation (PWM) control signal as described in the respective embodiments.

[0065] The discharge head cover 118, the bypass piping 11, and a solenoid valve 12 form a bypass flow path for guiding the refrigerant gas in the discharge head space 123 from the bypass piping 11 to the suction pipe 113 when the solenoid valve 12 is turned into the open state. Further, at the time of the ultrasmall volume-operation mode, the solenoid valve 12 is caused to make a repetitive action so as to be in either the opening state, or the closing state to repeat use/non-use of the bypass flow path, thereby putting the capacity-adjustment mechanism for the small-capacity-control at work.

[0066] The suction pipe 113 for taking in the refrigerant gas in the refrigerating cycle is connected with the fixed scroll 102. A portion of the interior of the closed casing 115, on the lower end side of the crankshaft 106, is an oil reservoir 116. Further, a flywheel for 117 for stabilizing

rotation is provided at a part of the crankshaft 106, between the rotor 100a and the rotor 100a and the secondary bearing 112.

[0067] Oil supplied from the oil reservoir 116 is guided, through the sliding bearing 130 provided around the eccentric part 106a of the crankshaft 106, to a back-pressure (an intermediate) chamber 109 formed by the fixed scroll 102, the orbiting scroll 101, and the frame 105. The back pressure chamber 109 is made up such that if the refrigerant gas in the oil is foamed, causing a rise in pressure, a control valve (not shown) will let a rising pressure escape to a suction side part thereof to thereby retain a predetermined pressure level. The suction side part is communicated with a fixed outer peripheral groove provided on the outer periphery of the fixed scroll 102, however, since the fixed outer peripheral groove is communicated with a suction inlet of the refrigerant gas, the interior of the fixed outer peripheral groove is constantly at the suction pressure. In the orbiting scroll 101, the discharge pressure acts on the central part thereof, and an intermediate pressure acts on a part thereof, on the outer peripheral side. For this reason, the orbiting scroll 101 is pressed against the fixed scroll 102 at PWM control signal an adequate pressure, so that sealing between scroll laps can be maintained.

[0068] In the case of this scroll compressor, a refrigerant gas compressed in the compression room is at a pressure higher than a pressure inside the discharge head space 123, or equal thereto, the refrigerant gas is discharged into the discharge head space 123 via the a release port 125 and the release valve 124. If the refrigerant gas is at a pressure less than the pressure inside the discharge head space 123, the release valve 12 is closed, the refrigerant gas is discharged from the discharge port 108 into the discharge head space 123, and the refrigerant gas is further discharged into a discharge room 103 by pushing the discharge valve 121 away from the through-hole 119. The refrigerant gas discharged into the discharge chamber 103 passes through a passage formed among the fixed scroll 102, the frame 105, and the closed casing 115 to flow into a discharge space 104 where the motor 100 is provided to be subsequently discharged into the refrigerating cycle via the discharge pipe 114. Accordingly, the scroll compressor has a high-pressure chamber structure where a space inside the closed casing 115 is at the discharge pressure.

[0069] Installed outside the scroll compressor 1 are an inverter 18 as a motor drive circuit for driving a motor 100, a solenoid drive circuit 12a for generating a pulse-width modulation control signal for drive-control of an open state, and a closed state of the solenoid valve 12, and a controller 20 as an operation-instruction control means for controlling the respective actions of the inverter 18, and the solenoid drive circuit 12a.

[0070] A compression operation of this scroll compressor is divided into a first operation mode with the solenoid valve 12 in the closed state and a second operation mode with the solenoid valve 12 in the open state. Fig. 11 shows

a flow of a refrigerant gas in the first operation mode when the solenoid valve 12 of a capacity-adjustment mechanism provided in the scroll compressor is in the closed state.

[0071] In the first operation mode, the solenoid drive circuit 12a turns the solenoid valve 12 into the closed state in a cycle τ_2 during a HIGH to LOW transition of a rectangular wave of the pulse-width modulation control signal, and the inverter 18 drives the motor 100 to thereby rotate the rotor 100a, and the crankshaft 106. In consequence, the orbiting scroll 101 starts a swing motion. This action causes the first compression room, and the second compression room, formed due to the engagement of respective spiral parts of the orbiting scroll 101 and the fixed scroll 102, with each other, to move toward the center while respective internal volumes are reduced.

[0072] Thereby, the refrigerant gas flowing from the suction pipe 113 is compressed by the first compression room, and the second compression chamber, respectively, whereupon a high-pressure refrigerant gas is discharged from the discharge port 108 formed in the fixed scroll 102 into the discharge head space 123. In the process of this compression, if a pressure in the compression chamber becomes higher than the pressure in the discharge head space 123, the high-pressure refrigerant gas is discharged into the discharge head space 123 via the release port 125, and the release valve 124, as previously described.

[0073] Further, the release valve 124 indicates a valve plate fitted to the tip of a coil spring 127 attached to a part of a presser part 126, on the tip side thereof, however, a release valve mechanism in whole, including the presser part 126, and the coil spring 127, is occasionally referred to as a release valve.

[0074] When a refrigerant gas pressure of the discharge head space 123 becomes slightly higher than the discharge pressure to be higher than the pressure of the discharge room (chamber) 103, the discharge valve 121 covering the through-hole 119 in the discharge head cover 118 is pushed to open, thereby causing the refrigerant gas to be discharged into the discharge room (chamber) 103.

[0075] In the first operation mode, the solenoid valve 12 is set in the closed state, thereby causing the refrigerant gas to flow toward the refrigerating cycle without use of the bypass piping 11. Therefore, the first operation mode may be called a load-operation.

[0076] Fig. 12 shows a flow of the refrigerant gas in the second operation mode when the solenoid valve 12 of the capacity-adjustment mechanism provided in the scroll compressor is in the open state.

[0077] In the second operation mode, the solenoid drive circuit 12a turns the solenoid valve 12 into the open state in a cycle τ_1 during a LOW to HIGH transition of the rectangular wave of the pulse-width modulation control signal, and the inverter 18 drives the motor 100 to thereby rotate the rotor 100a, and the crankshaft 106. In consequence, the orbiting scroll 101 starts a swing mo-

tion. This action causes the first compression room, and the second compression room, formed due to the engagement of the respective spiral parts of the orbiting scroll 101 and the fixed scroll 102, with each other, to move toward the center while the respective internal volumes are reduced.

[0078] In the second operation mode, since the solenoid valve 12 is in the open state, the refrigerant gas in the discharge head space 123 flows into the suction pipe 113 via the bypass piping 11 for connecting the discharge head space 123 to the suction pipe 113. Accordingly, a pressure inside the discharge head space 123 falls down to a suction pressure substantially at a level slightly higher than the suction pressure.

[0079] For this reason, the pressure of the discharge head space 123 is lower than the pressure of the discharge chamber 103, and the discharge valve 121 covering the through-hole 119 in the discharge head cover 118 is closed, so that the refrigerant gas is not discharged into the discharge chamber 103. In the state of the second operation mode, when the refrigerant gas taken into the suction pipe 113 is compressed by the first compression chamber, and the second compression chamber, respectively, the refrigerant gas will be at a pressure higher than the pressure in the discharge head space 123, so that the refrigerant gas is discharged into the discharge head space 123 via the release port 125 and the release valve 124. Further, the refrigerant gas moved further toward the center side of the release port 125, in the compression chamber, is discharged the discharge port 108 into the discharge head space 123. The refrigerant gas discharged into the discharge head space 123 passes through the bypass piping 11, and the solenoid valve 12 in the open state before flowing into the suction pipe 113.

[0080] In the second operation mode, the solenoid valve 12 is set in the open state, and the refrigerant gas from the bypass piping 11 is returned to the suction pipe 113, without discharging the refrigerant gas toward the refrigerating cycle. Therefore, the second operation mode may be called an unload-operation.

[0081] The release port 125 and the release valve 124 each are preferably installed at such a position as to enable communication with compression chambers in all rotation angle regions. The reason is because internal compression within the scroll lap can be avoided in this way, and a compression action in the unload-operation can be reduced.

[0082] With the scroll compressor according to the first embodiment of the invention, the motor 100 is driven by the inverter 18, and switchover is made between the load operation (the first operation mode) whereby the solenoid valve 12 is turned into the closed state in the cycle τ_2 during the HIGH to LOW transition of the rectangular wave of the pulse-width modulation control signal, and the unload operation (the second operation mode) whereby the solenoid valve 12 is turned into the open state in the cycle τ_1 during the LOW to HIGH transition of the rectangular wave, thereby enabling a capacity-control to

be performed.

[0083] Even at the time of a high-speed operation mode when the scroll compressor is operated at a relatively high-speed, the capacity-control is enabled by the opening/closing of the solenoid valve, however, in a rotation range of from a high-speed rotation to a predetermined set value slightly higher than the low limit set value of the motor rotational speed by motor-driving, control of the rotational speed of the motor 100 is executed by the inverter 18, while in the case where it is necessary to further reduce the capacity in a low-speed rotation range lower than the predetermined set value, the capacity-adjustment mechanism (for controlling the opening/closing of a bypass passage by use of the solenoid valve 12) for executing a small-capacity control is preferably set in motion to operate in the ultrasmall-capacity operation mode by changing a ratio of the load-operation to the unload-operation.

[0084] With the scroll compressor provided with the capacity-adjustment mechanism, the small-capacity control can be efficiently carried out by the capacity-adjustment mechanism simple in structure even at the time of the ultrasmall-capacity operation mode. More specifically, it becomes possible to implement the compression action at the ultrasmall-capacity control (the ultrasmall-capacity operation mode) corresponding to the case of the ultra-low speed operation executed at not higher than the low-limit set value of the motor rotational speed (in the drive signal to the motor, the frequency is on the order of 5Hz) without deteriorating efficiency in motor driving, so that an excellent scroll compressor capable of realizing the capacity-control in the wide range of 0 to 100% can be obtained. Furthermore, since the capacity-adjustment mechanism provided in the scroll compressor according to the present embodiment is simple in structure, reduction in cost, miniaturization, reduction in weight, and mass-production can be easily realized with respect to the scroll compressor.

[0085] As described in the foregoing, with the refrigerating cycle system according to the present embodiment, the duty period as a cycle of a switchover time between the load-operation and the unload-operation is controlled such that the deviation of the evaporating pressure falls within a given value, and a rise as well as change of the suction pressure can therefore be controlled within a threshold value, thereby enabling comfortability such as suitable air conditioning, and so forth to be enhanced. Further, with the present embodiment, the losses due to an excessively shortened duty period can be prevented, so that it is possible to realize a high-performance refrigerating cycle system capable of implementing an operation high in efficiency, and a highly efficient capacity-control in a wide range of 0 to 100%. Further, with the present embodiment, since the highly efficient capacity-control in the wide range can be implemented, reduction in cost is enabled. List of Reference Signs

[0086]

1: compressor 1,
 2: outdoor heat exchanger,
 3: expansion valve,
 4: indoor heat exchanger,
 5: four-way valve,
 7: high-pressure side connection pipe,
 8: outdoor connection pipe,
 9: indoor connection pipe,
 10: low-pressure side connection pipe,
 11: bypass pipe (bypass flow path),
 12: solenoid valve, 12a: solenoid drive circuit,
 13: discharge-temperature sensor,
 14: indoor heat-exchanger temperature sensor,
 15: outdoor heat-exchanger temperature sensor,
 16: indoor temperature sensor,
 17: outdoor temperature sensor,
 18: inverter,
 19: commercial AC power supply source,
 20: controller, 21: outdoor fan, 22: indoor fan,
 23: suction pressure sensor,
 24: blow-out temperature sensor,
 100: motor (100a: rotor, 100b: stator),
 101: orbiting scroll 101, 102: fixed scroll,
 103: discharge chamber, 104: discharge space,
 105: frame, 105a: main bearing,
 106: crankshaft, 16a: eccentric part,
 107: Oldham coupling ring, 18: discharge port,
 109: back-pressure chamber (intermediate chamber),
 111: lower frame, 112: secondary bearing,

113: suction pipe, 114: discharge pipe,
 115: closed casing, 116: oil reservoir,
 5 117: flywheel, 118: discharge head cover,
 119: through-hole, 121: discharge valve,
 123: discharge head space, 124: release valve,
 10 125: release port, 126: presser part, 127: coil spring, and
 130: sliding bearing
 15

Claims

1. A refrigerating cycle system provided with a compressor, an outdoor heat exchanger, an expansion valve capable of controlling an opening thereof, and an indoor heat exchanger, said refrigerating cycle system comprising:

a bypass flow path for causing a refrigerant in the middle of compression to bypass toward a suction side of the compressor;
 a solenoid valve for opening or closing the bypass flow path; and
 a controller for controlling open (ON) state time of the solenoid valve, and closed (OFF) state time thereof to adjust a flow rate of a refrigerant discharged from the compressor into a refrigerating cycle, thereby executing a capacity-control, wherein the controller executes a control on the basis of a duty ratio that is a ratio of open time of the solenoid valve against a duty period representing the sum of the open time of the solenoid valve, and closed time thereof, and further, if a pressure on the suction side of the compressor when the solenoid valve is in the ON state is higher than a suction pressure before the solenoid valve is in the ON state by a allowable deviation, the controller controls the solenoid valve so as to be in the closed state, the closed time being decided on the basis of the duty period.

2. The refrigerating cycle system according to claim 1, wherein the duty period is decided on the basis of a difference between a room air temperature and a room air temperature target value as set.

3. The refrigerating cycle system according to claim 1, wherein the controller controls the opening of the expansion valve such that a discharge refrigerant temperature of a refrigerant discharged from the

compressor approaches a target discharge temperature.

4. The refrigerating cycle system according to claim 3, wherein when the solenoid valve is controlled so as to be in the closed-state, the target discharge temperature is decided on the basis of a temperature (an evaporation temperature) of the outdoor heat-exchanger, an outdoor air temperature, a rotational speed of the compressor, and a rotational speed command value of an outdoor fan, when the solenoid valve is controlled so as to be in the open-state, a corrected compressor rotational speed is determined by multiplying a compressor rotational speed at the time of starting opening/closing control of the solenoid valve by a duty ratio at that point in time, and the target discharge temperature is decided on the basis of the corrected compressor rotational speed, the temperature (the evaporation temperature) of the outdoor heat-exchanger temperature sensor, the outdoor air temperature, and the rotational speed command value of the outdoor fan.
5. The refrigerating cycle system according to claim 1, wherein a pressure on the suction side of the compressor is detected by a suction pressure sensor provided on the suction side of the compressor.
6. The refrigerating cycle system according to claim 1, wherein for a pressure on the suction side of the compressor to become higher than a suction pressure P_{s0} before the solenoid valve is turned into the open state by a preset allowable deviation ΔP is determined by estimating variation (change) in suction pressure on the basis of a temperature T_{ev0} of the indoor heat-exchanger or the outdoor heat-exchanger, serving as the evaporator, and a blow-out temperature of the indoor heat exchanger.
7. The refrigerating cycle system according to claim 1, wherein a temperature sensor for detecting a temperature in the vicinity of the center of the indoor heat-exchanger or the outdoor heat-exchanger, serving as the evaporator, is provided, and variation (change) in an evaporating pressure is estimated on the basis of a temperature detected by the temperature sensor, thereby determining that a pressure on the suction side of the compressor becomes higher than a suction pressure P_{s0} before the solenoid valve is turned into the open state by a preset allowable deviation ΔP .
8. A refrigerating cycle system provided with a compressor, an outdoor heat exchanger, an expansion valve capable of controlling an opening thereof, and an indoor heat exchanger, said refrigerating cycle system comprising:
 - a bypass flow path for causing a refrigerant in the middle of compression to bypass toward a suction side of the compressor;
 - a solenoid valve for opening or closing the bypass flow path; and
 - a controller for controlling open (ON) state time of the solenoid valve, and closed (OFF) state time thereof to adjust a flow rate of a refrigerant discharged from the compressor into a refrigerating cycle, thereby executing a capacity-control, wherein the controller executes a control on the basis of a duty ratio that is a ratio of open time of the solenoid valve against a duty period representing the sum of the open time of the solenoid valve, and closed time thereof, and further, if an evaporator temperature of the indoor heat-exchanger or the outdoor heat-exchanger (an evaporator-side heat-exchanger), serving as the evaporator, when the solenoid valve is in the ON state, becomes higher than an evaporator temperature before the solenoid valve is in the ON state by a allowable deviation, the controller controls the solenoid valve so as to be in the closed state, the closed time being decided on the basis of the duty period.
9. The refrigerating cycle system according to claim 8, wherein a blow-out temperature sensor is provided in the vicinity of the outlet of a ventilation passage of the indoor heat exchanger, and the allowable deviation ΔT_{ev} is worked out according to a table for holding the evaporator temperature before the solenoid valve is in the ON state, and a blow-out temperature T_{ea_out} of the indoor heat exchanger, detected by the blow-out temperature sensor, as preset controlled parameters.
10. The refrigerating cycle system according to claim 8, further comprising an evaporator-temperature sensor for detecting a temperature in the vicinity of the center of evaporator-side heat exchanger, and a temperature sensor provided in the vicinity of the inlet of a ventilation passage of the indoor heat exchanger, wherein the allowable deviation ΔT_{ev} is worked out according to a table for holding an evaporator temperature at the time of starting measurement, detected by the evaporator-temperature sensor, and an air temperature measured by the temperature sensor provided in the vicinity of the inlet of a ventilation passage of the evaporator-side heat exchanger, as preset controlled parameters.
11. The refrigerating cycle system according to claim 8, wherein the duty period is decided on the basis of a difference between a room air temperature and a room air temperature target value as set.

12. The refrigerating cycle system according to claim 8, wherein the controller controls the opening of the expansion valve such that a discharge refrigerant temperature of a refrigerant discharged from the compressor approaches a target discharge temperature. 5
13. The refrigerating cycle system according to claim 12, wherein when the solenoid valve is controlled so as to be in the closed-state, the target discharge temperature is decided on the basis of a temperature (an evaporation temperature) of the outdoor heat-exchanger, an outdoor air temperature, a rotational speed of the compressor, and a rotational speed command value of an outdoor fan, when the solenoid valve is controlled so as to be in the open-state, a corrected compressor rotational speed is determined by multiplying a compressor rotational speed at the time of starting opening/closing control of the solenoid valve by a duty ratio at that point in time, and the target discharge temperature is decided on the basis of the corrected compressor rotational speed, the temperature (the evaporation temperature) of the outdoor heat-exchanger temperature sensor, the outdoor air temperature, and the rotational speed command value of the outdoor fan. 10 15 20 25
14. The refrigerating cycle system according to claim 1, wherein the compressor is a scroll compressor incorporated in a closed casing, the scroll compressor comprising a fixed scroll having a spiral body, and an orbiting scroll having such a spiral body as to be engaged with the spiral body of the fixed scroll to thereby form compression chambers, and a discharge port is formed around the center of the fixed scroll, a release port connecting with the compression chamber, together with a release valve for opening/closing the release port, being formed on the outer peripheral side of the discharge port. 30 35 40
15. The refrigerating cycle system according to claim 14, wherein the bypass flow path is a bypass piping 11 connecting the release port provided in the scroll compressor to a suction pipe provided on the suction side of the scroll compressor, and the solenoid valve 12 is provided in the bypass piping. 45
16. The refrigerating cycle system according to claim 15, wherein the scroll compressor is provided with a discharge head cover attached to a top plate of the fixed scroll, covering both the discharge port, and the release valve to thereby form a discharge head space, the discharge head cover is provided with both a through-hole communicating with a discharge room inside the closed casing, and a discharge valve for opening/closing the through-hole, and the bypass piping is provided in such a way as to connect the discharge head space to the suction pipe while the solenoid valve is driven to be controlled so as to be in the open state, and the closed state by the agency of a pulse width modulation (PWM) control signal. 50 55

FIG. 1

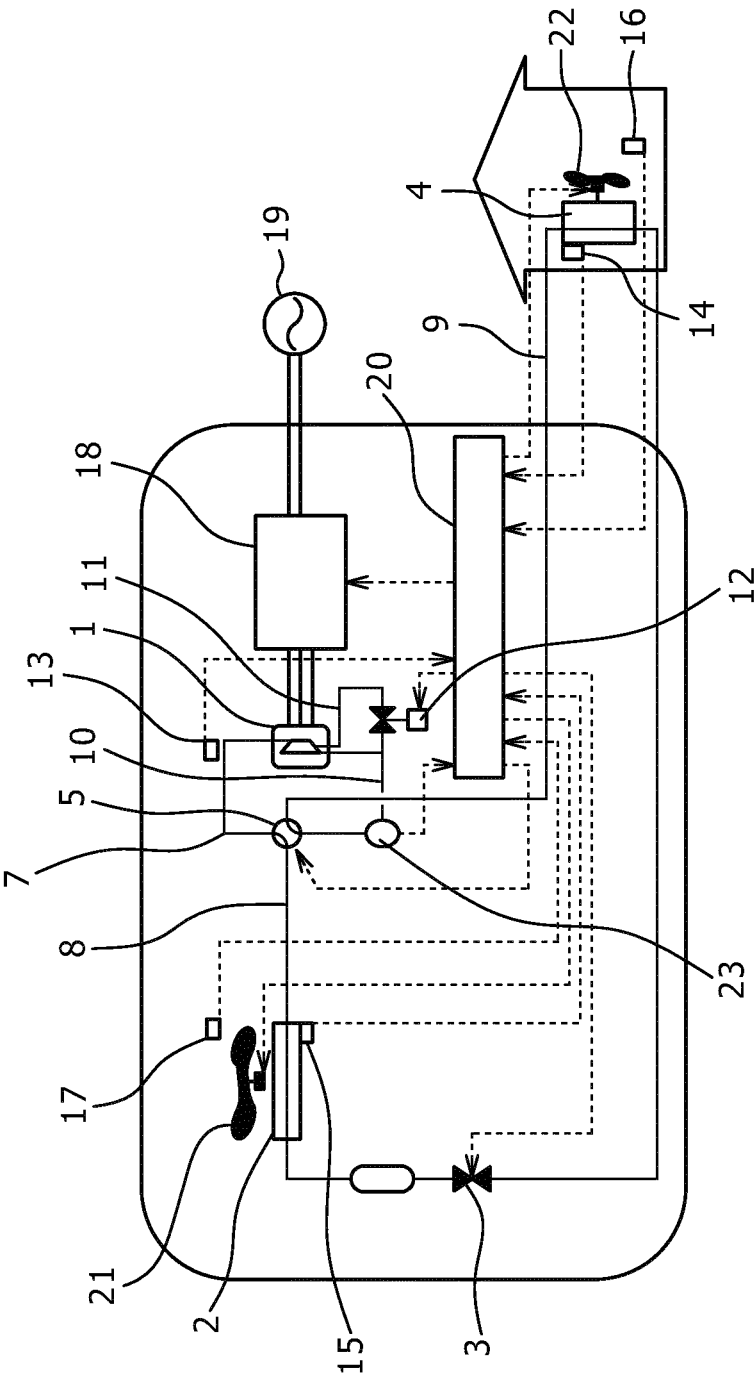


FIG. 2

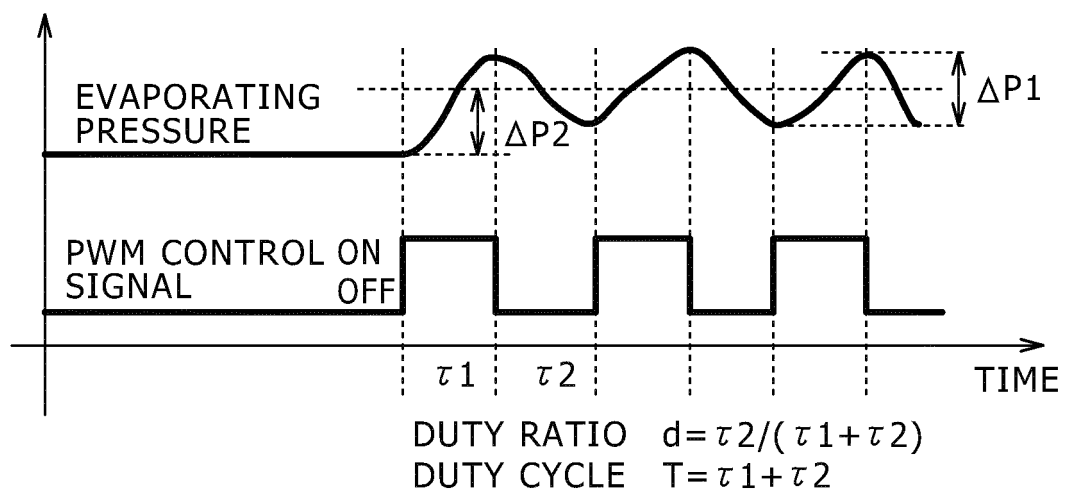


FIG. 3

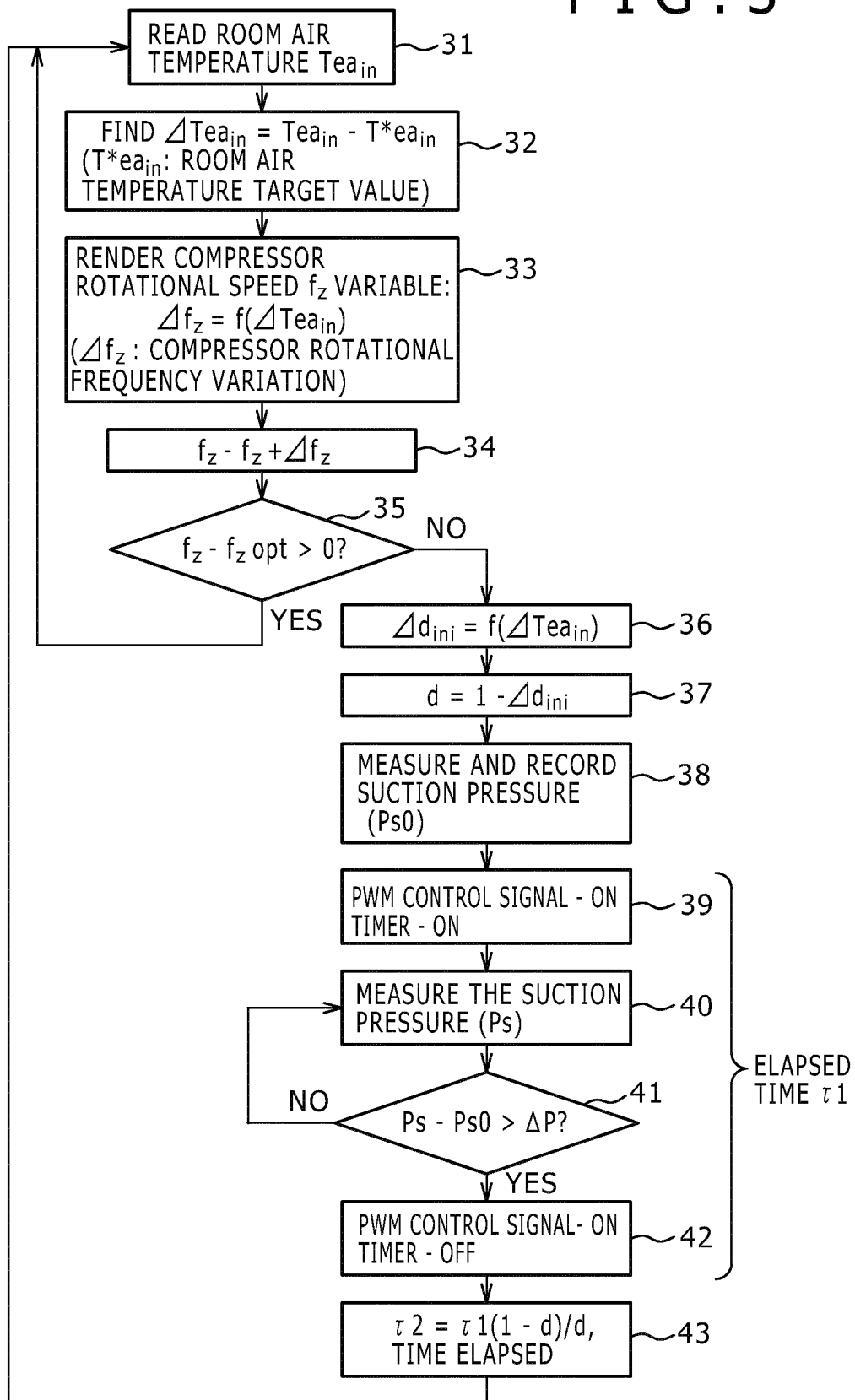


FIG. 4

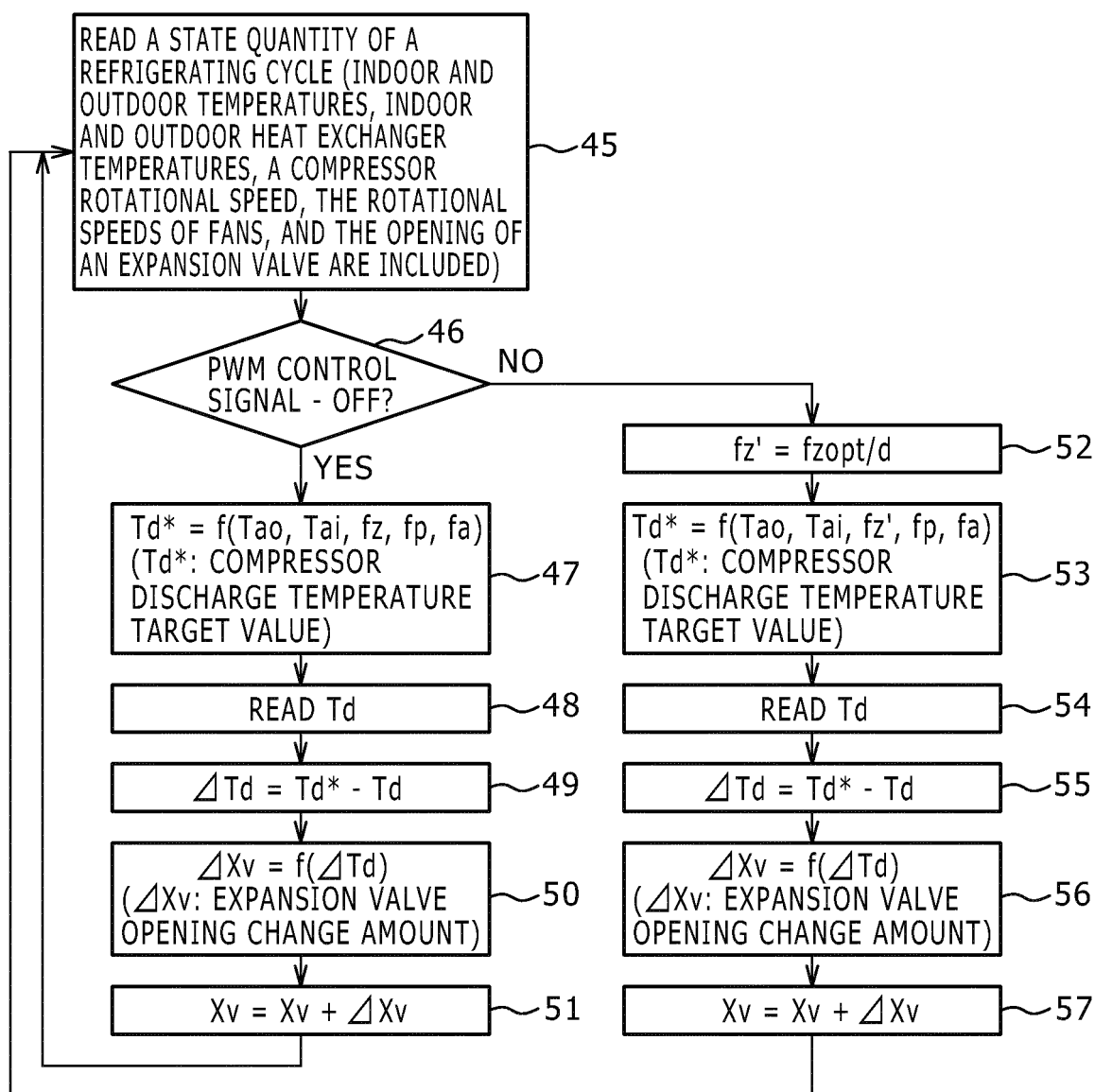


FIG. 5

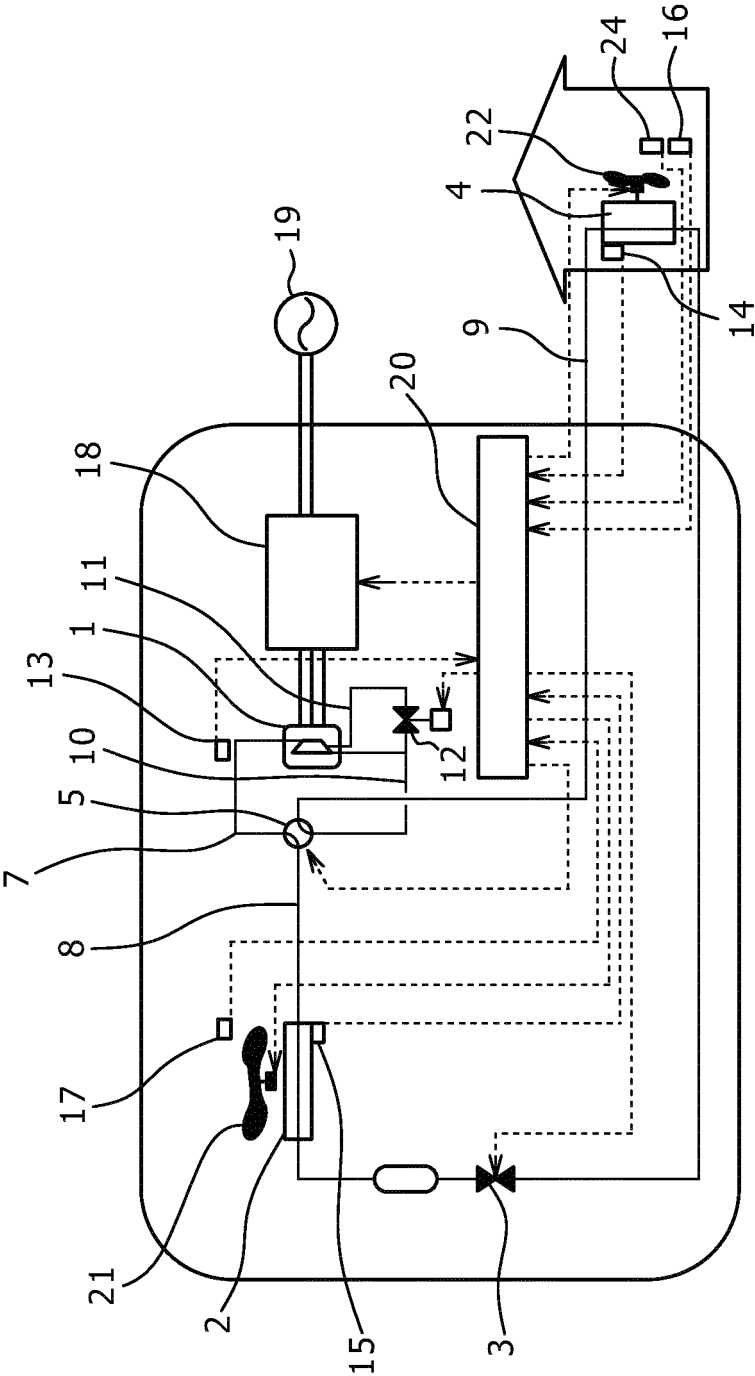


FIG. 6

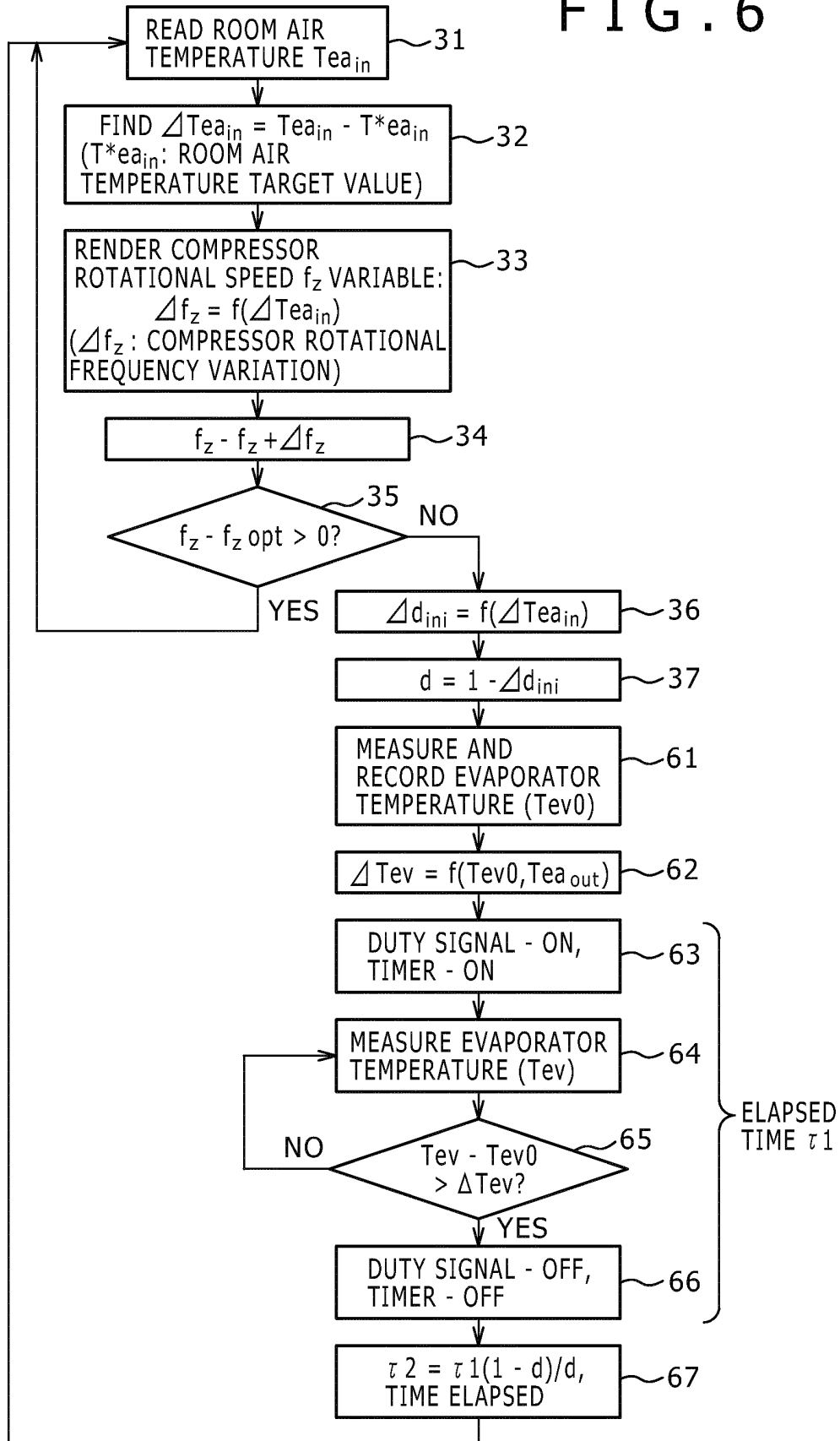


FIG. 7

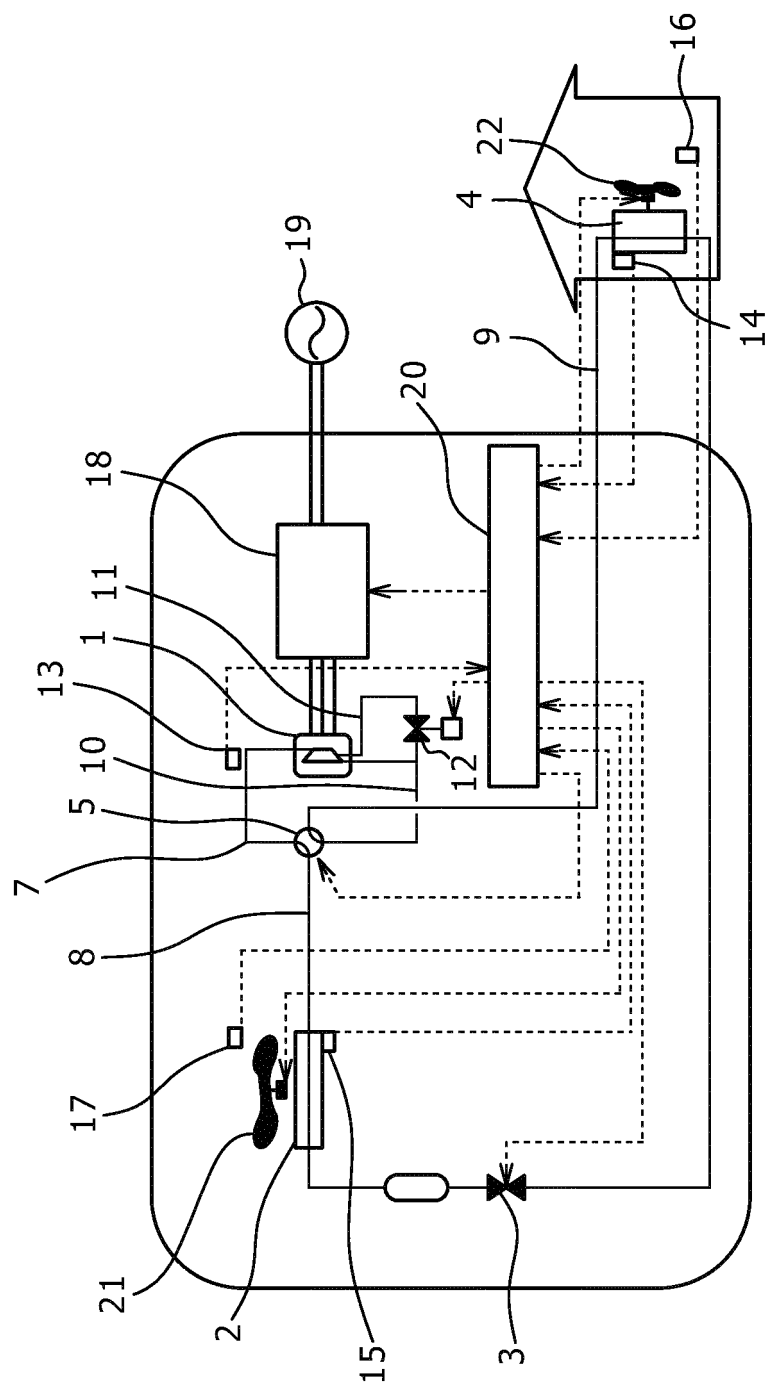


FIG. 8

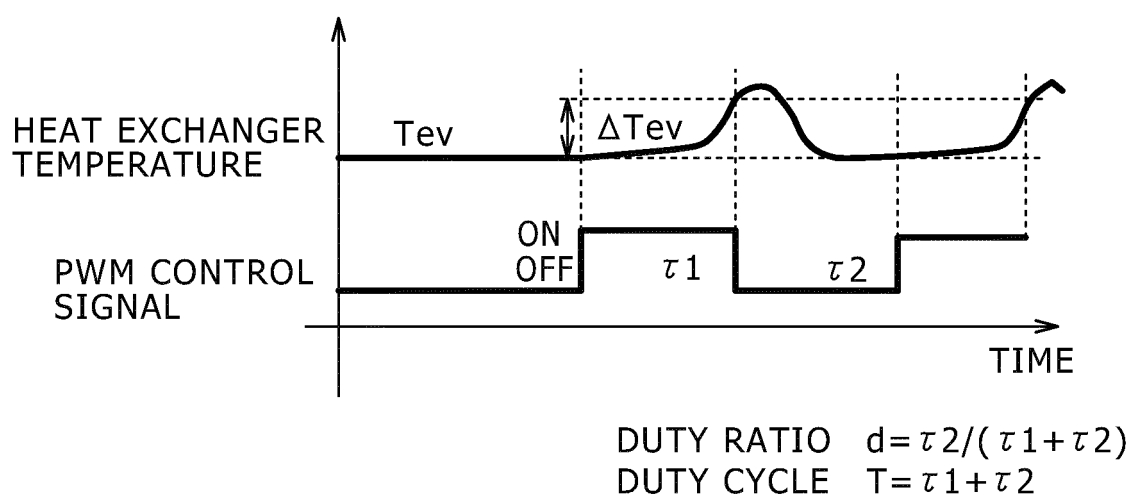


FIG. 9

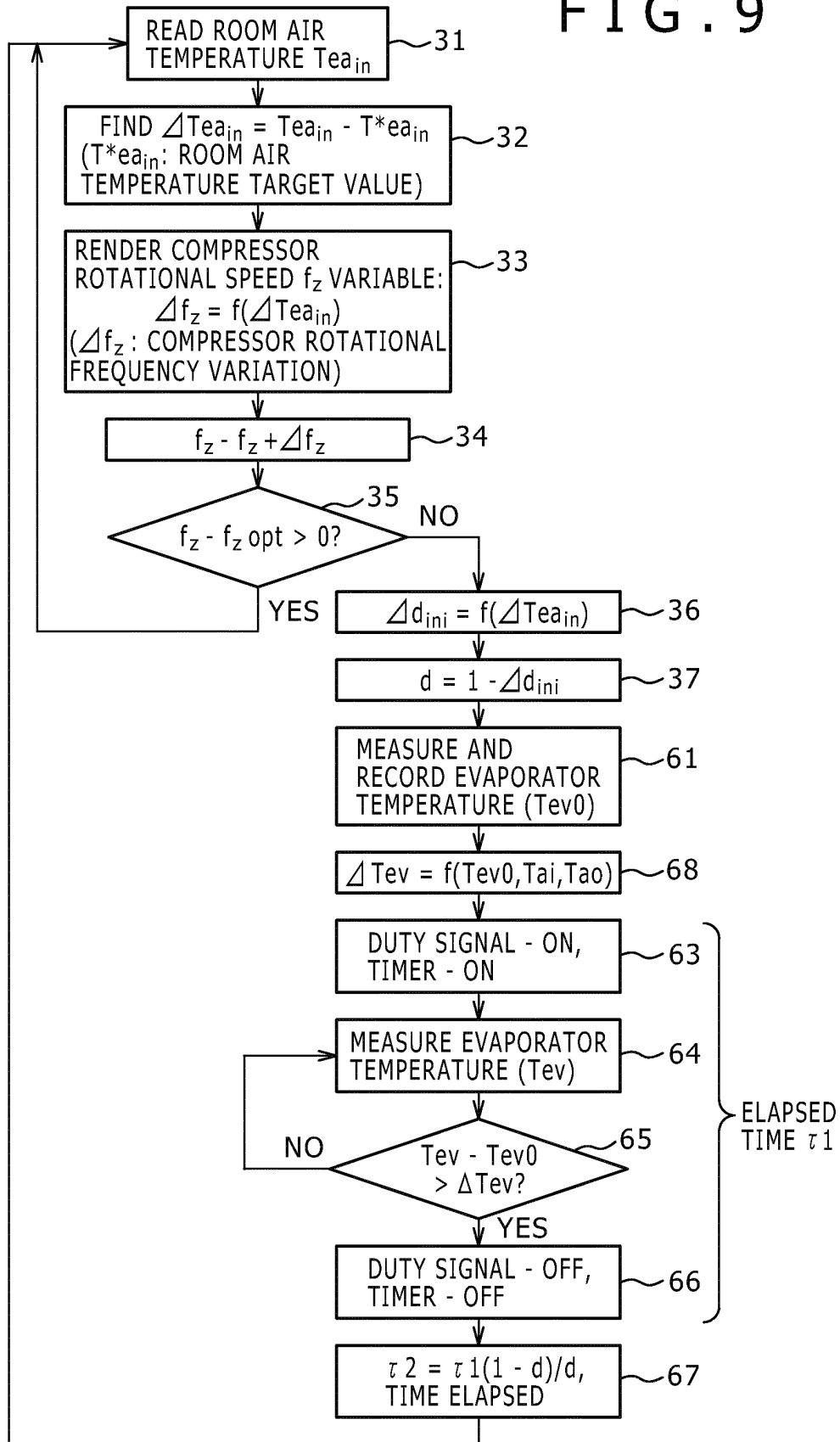


FIG. 10

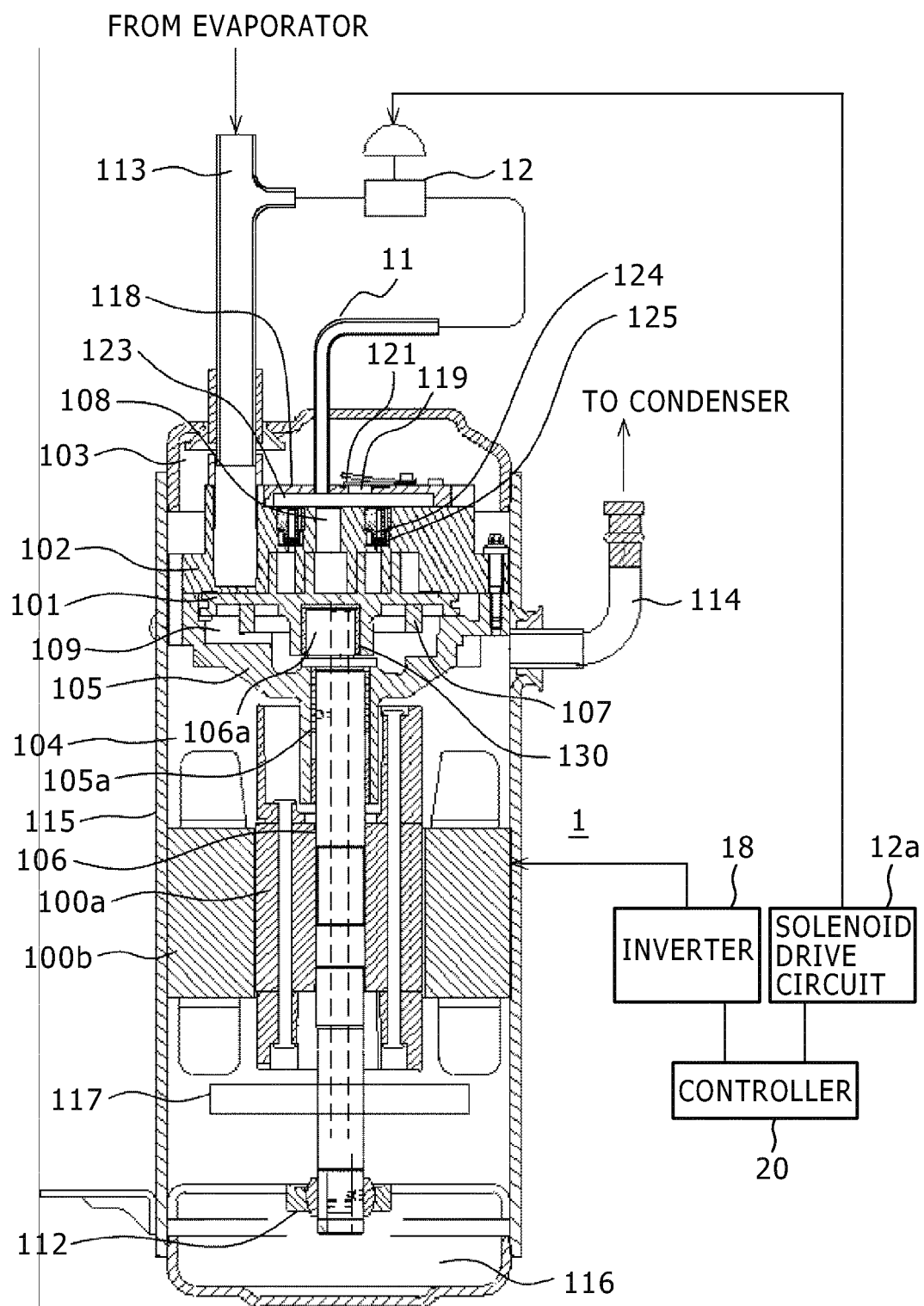


FIG. 11

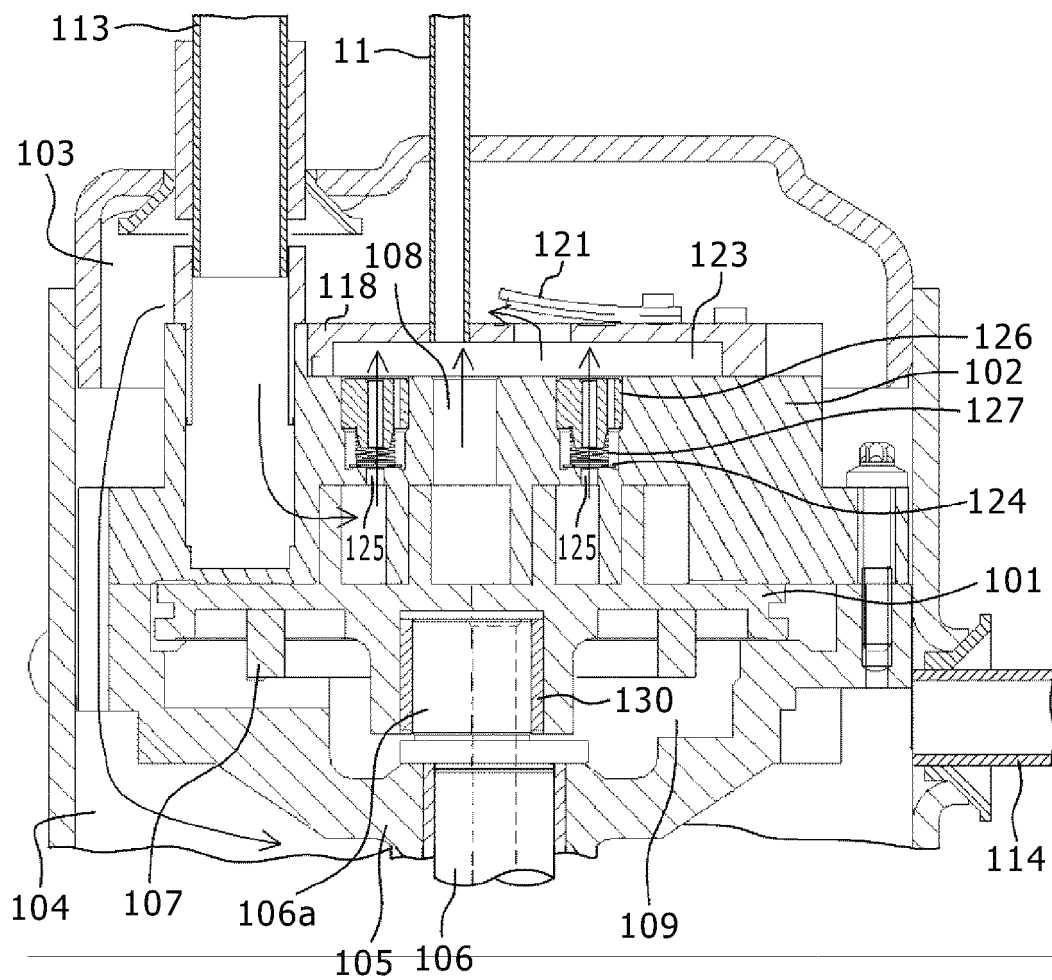
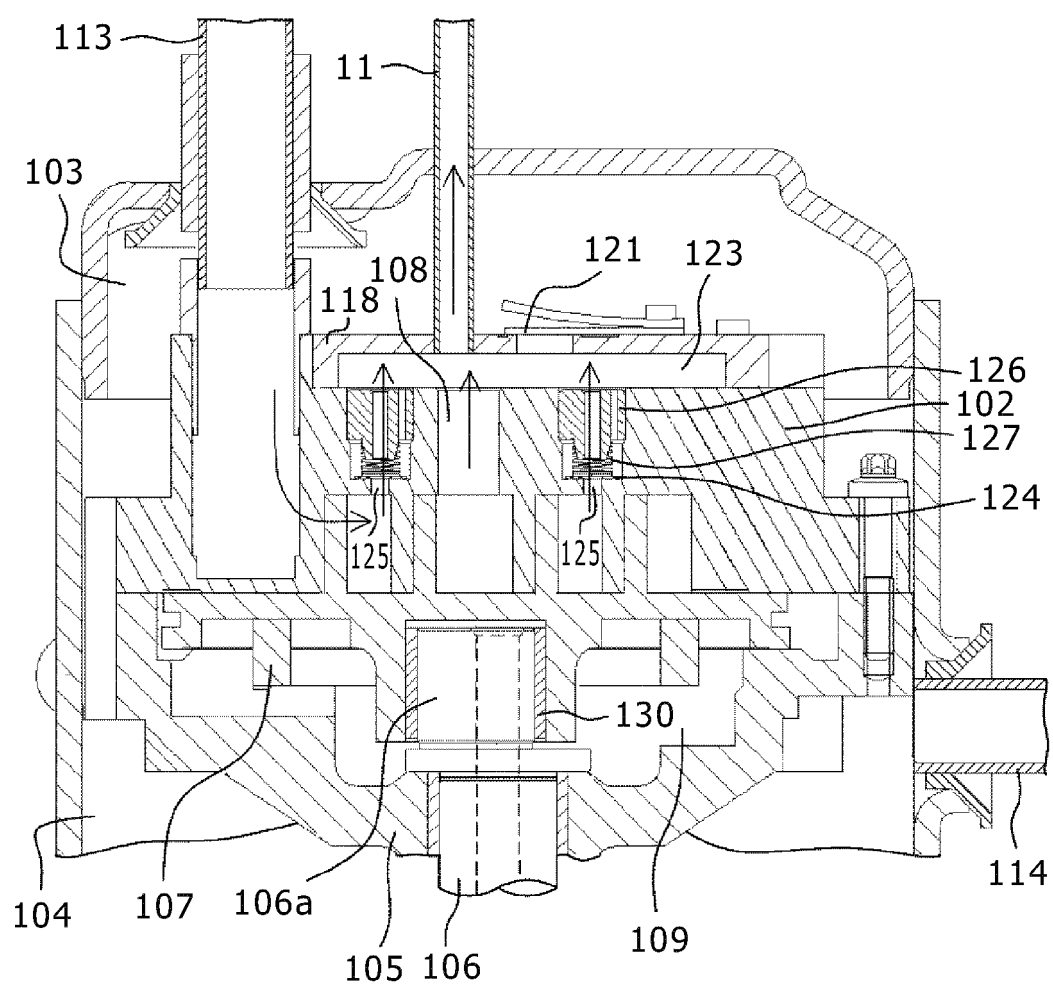


FIG. 12



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2011/053839

A. CLASSIFICATION OF SUBJECT MATTER <i>F25B1/00(2006.01) i, F25B1/04(2006.01) i</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) <i>F25B1/00, F25B1/04</i>										
Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched <table border="0"> <tr> <td>Jitsuyo Shinan Koho</td> <td>1922-1996</td> <td>Jitsuyo Shinan Toroku Koho</td> <td>1996-2011</td> </tr> <tr> <td>Kokai Jitsuyo Shinan Koho</td> <td>1971-2011</td> <td>Toroku Jitsuyo Shinan Koho</td> <td>1994-2011</td> </tr> </table>			Jitsuyo Shinan Koho	1922-1996	Jitsuyo Shinan Toroku Koho	1996-2011	Kokai Jitsuyo Shinan Koho	1971-2011	Toroku Jitsuyo Shinan Koho	1994-2011
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.								
Y	JP 4-340058 A (Toshiba Corp.), 26 November 1992 (26.11.1992), claims; paragraphs [0001] to [0026]; fig. 1 to 6	1-16								
Y	JP 58-193055 A (Tokyo Shibaura Electric Co., Ltd.), 10 November 1983 (10.11.1983), claims; page 1, lower right column, line 2 to page 4, line 13; fig. 1 to 5	1-16								
Y	JP 62-126289 A (Toshiba Corp.), 08 June 1987 (08.06.1987), claims; page 1, line 19 to page 4, lower left column, line 17; fig. 1 to 3	1-16								
☒ Further documents are listed in the continuation of Box C. ☒ See patent family annex.										
<table border="0"> <tr> <td> * Special categories of cited documents: "A" document defining the general state of the art which is not considered to be of particular relevance "E" earlier application or patent but published on or after the international filing date "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 </td> <td> "I" 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 </td> </tr> </table>			* Special categories of cited documents: "A" document defining the general state of the art which is not considered to be of particular relevance "E" earlier application or patent but published on or after the international filing date "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	"I" 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						
* Special categories of cited documents: "A" document defining the general state of the art which is not considered to be of particular relevance "E" earlier application or patent but published on or after the international filing date "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	"I" 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 18 May, 2011 (18.05.11)		Date of mailing of the international search report 31 May, 2011 (31.05.11)								
Name and mailing address of the ISA/ Japanese Patent Office		Authorized officer								
Facsimile No.		Telephone No.								

INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2011/053839

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT		
Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	Microfilm of the specification and drawings annexed to the request of Japanese Utility Model Application No. 156804/1988 (Laid-open No. 77555/1990) (Mitsubishi Electric Corp.), 14 June 1990 (14.06.1990), page 8, line 10 to page 9, line 8; fig. 1 to 2	1-16
Y	JP 2008-180420 A (Toshiba Corp.), 07 August 2008 (07.08.2008), claim 4	2, 8, 11
Y	JP 2005-16884 A (Matsushita Electric Industrial Co., Ltd.), 20 January 2005 (20.01.2005), claims; paragraphs [0001] to [0036]; fig. 1 to 7	8-13
Y	JP 2003-28517 A (Matsushita Electric Industrial Co., Ltd.), 29 January 2003 (29.01.2003), claims; paragraphs [0001] to [0096]; fig. 1 to 7	3, 12
Y	JP 9-42783 A (Matsushita Electric Industrial Co., Ltd.), 14 February 1997 (14.02.1997), claims; paragraphs [0001] to [0034]; fig. 1 to 12	4, 5, 13
Y	JP 5-149608 A (Daikin Industries, Ltd.), 15 June 1993 (15.06.1993), claims; paragraphs [0001] to [0030]; fig. 1 to 6 (Family: none)	6, 7, 9, 10
Y	JP 6-26470 A (Toshiba Corp.), 01 February 1994 (01.02.1994), claims; paragraphs [0001] to [0052]; fig. 1 to 2	14-16
A	JP 8-334094 A (Coopland Corp.), 17 December 1996 (17.12.1996), entire text; all drawings	1-16
A	JP 11-324951 A (Mitsubishi Electric Corp.), 26 November 1999 (26.11.1999), entire text; all drawings	1-16

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

INTERNATIONAL SEARCH REPORT
 Information on patent family members

International application No.

PCT/JP2011/053839

JP 4-340058 A	1992.11.26	(Family: none)	
JP 58-193055 A	1983.11.10	US 4517812 A	1985.05.21
		GB 2121994 A	1984.01.04
		KR 1988-0004147 Y1	1988.11.18
JP 62-126289 A	1987.06.08	(Family: none)	
JP 2-77555 U	1990.06.14	(Family: none)	
JP 2008-180420 A	2008.08.07	(Family: none)	
JP 2005-16884 A	2005.01.20	(Family: none)	
JP 2003-28517 A	2003.01.29	(Family: none)	
JP 9-42783 A	1997.02.14	(Family: none)	
JP 5-149608 A	1993.06.15	(Family: none)	
JP 6-26470 A	1994.02.01	(Family: none)	
JP 8-334094 A	1996.12.17	US 5741120 A	1998.04.21
		EP 747597 A2	1996.12.11
		DE 69534835 T2	2006.08.31
		ES 2258766 T3	2006.09.01
		CN 1137614 A	1996.12.11
JP 11-324951 A	1999.11.26	(Family: none)	

REFERENCES CITED IN THE DESCRIPTION

This list of references cited by the applicant is for the reader's convenience only. It does not form part of the European patent document. Even though great care has been taken in compiling the references, errors or omissions cannot be excluded and the EPO disclaims all liability in this regard.

Patent documents cited in the description

- JP HEI81996334094 B [0009]
- JP HEI111999324951 B [0009]