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

(11) EP 2 290 304 A1

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

(43) Date of publication:
02.03.2011 Bulletin 2011/09

(21) Application number: **09729048.0**

(22) Date of filing: **31.03.2009**

(51) Int Cl.:
F25B 1/00 *(2006.01)* ***F25B 29/00*** *(2006.01)*

(86) International application number:
PCT/JP2009/056655

(87) International publication number:
WO 2009/123190 (08.10.2009 Gazette 2009/41)

(84) Designated Contracting States:
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 SE SI SK TR
Designated Extension States:
AL BA RS

(30) Priority: **31.03.2008 PCT/JP2008/056370**

(71) Applicant: **Mitsubishi Electric Corporation**
Tokyo 100-8310 (JP)

(72) Inventors:
- **HOKAZONO, Keisuke**
  Tokyo 100-8310 (JP)
- **HATOMURA, Takeshi**
  Tokyo 100-8310 (JP)
- **MORIMOTO, Hiroyuki**
  Tokyo 100-8310 (JP)

(74) Representative: **Pfenning, Meinig & Partner GbR**
Theresienhöhe 13
D-80339 München (DE)

(54) AIR CONDITIONER

(57) Provided is an air conditioner capable of improving a COP in simultaneous heating and cooling operation. The air conditioner is an air conditioning system in which an outdoor unit (100) and a plurality of indoor units (301 to 303) are connected through a branch controller (200), and a supercritical fluid is used, to thereby establish a single refrigerating cycle. The outdoor unit (100) and the branch controller (200) are connected through two pipes of a high-pressure pipe (400) and a low-pressure pipe (500). The branch controller (200) and each of the plurality of indoor units (301 to 303) are connected through two pipes of a high-pressure pipe (700) and a low-pressure pipe (800). The branch controller (200) includes a double-pipe heat exchanger (240) for heat exchange between a medium-pressure two-phase refrigerant and a low-pressure two-phase refrigerant. The medium-pressure two-phase refrigerant is relatively high in temperature and flows into the double-pipe heat exchanger after branching a refrigerant flowing from the outdoor unit toward the plurality of indoor units, and joining together a refrigerant decompressed by a first expansion valve (211) and a refrigerant flowing from the plurality of indoor units. The low-pressure two-phase refrigerant is relatively low in temperature and flows out of the double-pipe heat exchanger toward the outdoor unit after branching a refrigerant flowing out of the double-pipe heat exchanger toward the plurality of indoor units, and decompressing a part of the branched refrigerant by a second expansion valve (212).

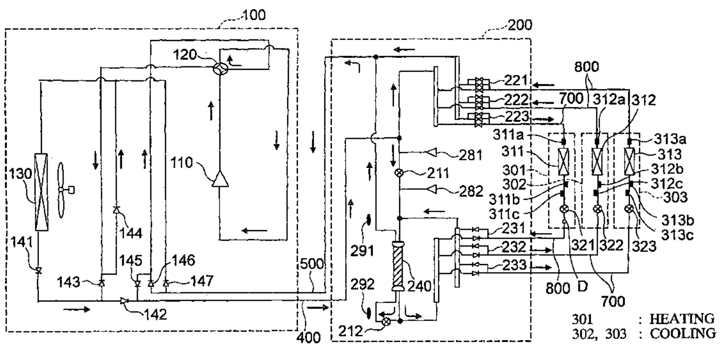

FIG. 1

EP 2 290 304 A1

Description

Technical Field

[0001] The present invention relates to an air conditioner in which an outdoor unit and a plurality of indoor units are connected through a branch controller, and a supercritical fluid is used, to thereby establish a single refrigerating cycle.

Background Art

[0002] There has conventionally been known a heat recovery air conditioner for simultaneous heating and cooling, which uses a supercritical fluid such as CO_2. In such an air conditioner, an outdoor unit and a branch kit are mostly connected through three pipes of a high-pressure pipe, a low-pressure pipe, and a high-temperature gas pipe. For the piping from the branch kit to the indoor unit, a two-pipe system is employed.

[0003] However, the pressure of the supercritical fluid is extremely high in a critical range, and hence the wall thickness of each connection pipe laid between the units greatly increases as compared with a conventional case of a refrigerant typified by chlorofluorocarbon. This fact may easily lead to expectations of increased cost of materials and enormously increased cost of processes on site, such as pipe bending.

[0004] Therefore, it is conceived that the branch kit for each indoor unit is incorporated into a single branch controller for the purpose of reduction in number of connection pipes.

[0005] Meanwhile, the air conditioner using the supercritical fluid has such a characteristic that the highest performance is obtained with a lower flow rate of the fluid by lowering temperature of the fluid to be conveyed to an indoor unit in cooling operation and raising temperature of the fluid to be conveyed to an indoor unit in heating operation. Accordingly, the efficiency (in this case, coefficient of performance (COP) expressed by taking performance of an air handling unit (unit: kW) as its numerator and power consumption (unit: kW) as its denominator) is enhanced as well. Thus, temperature of an inlet of the indoor unit, that is, temperature of an outlet of a heat-source side heat exchanger is basically lowered at the time of cooling and raised at the time of heating.

[0006] However, in the air conditioner that employs a two-pipe system to allow simultaneous heating and cooling, there is such a trade-off as described below in a case where the indoor unit in cooling operation and the indoor unit in heating operation simultaneously exist (are mixed).

[0007] · It is necessary to lower the temperature of the outlet of the heat-source side heat exchanger so as to supply a low-temperature fluid to the indoor unit in cooling operation.

· It is necessary to raise the temperature of the outlet of the heat-source side heat exchanger so as to supply a high-temperature fluid to the indoor unit in heating operation.

[0008] For example, in cooling-dominant operation of the conventional technology (the refrigerating cycle is a refrigerating cycle of simultaneous heating and cooling operation in a cooling cycle), both cooling and heating are inevitably controlled at a certain degree of temperature (for example, approximately 40°C to 50°C under a pressure of 10 MPa in the supercritical range of the Mollier chart) of the outlet of the heat-source side heat exchanger. Consequently, there is an insufficiency of an enthalpy difference to obtain high performance, and hence a flow rate of the fluid is increased (power consumption of a compressor is increased) for compensation therefor, which results in a lower COP.

[0009] Further, the efficiency of the air conditioner is conventionally evaluated based on the above-mentioned coefficient called COP in terms of only the efficiency with respect to 100% loads. In recent years, however, as to the loads in general offices, for example, cooling loads have been generated even in a season that necessitates heating, along with development of OA appliances and improvement in heat insulation performance of buildings. As a result, the frequency of the simultaneous heating and cooling operation is becoming higher throughout a year. Therefore, the efficiency improvement tends more increasingly to be evaluated based not only on the COP with respect to the 100% loads but also on a COP obtained at the time of the simultaneous heating and cooling operation.

Disclosure of the Invention

Problem to be solved by the Invention

[0010] As described above, the conventional air conditioner has the problem of decline in COP caused by the operation for satisfying both cooling and heating.

[0011] The present invention has been made in view of the points described above, and it is therefore an object of the present invention to provide an air conditioner capable of improving a COP in simultaneous heating and cooling operation.

Means for solving the Problem

[0012] An air conditioner according to the present invention is an air conditioning system in which an outdoor unit and a plurality of indoor units are connected through a branch controller, and a supercritical fluid is used, to thereby establish a single refrigerating cycle, the outdoor unit and the branch controller being connected through two pipes of a high-pressure pipe and a low-pressure pipe, the branch controller and each of the plurality of indoor units being connected through two pipes of a high-pressure pipe and a low-pressure pipe, in which the branch controller includes a double-pipe heat exchanger for heat exchange between a medium-pressure two-phase refrigerant and a low-pressure two-phase refrigerant, the medium-pressure two-phase refrigerant being relatively high in temperature and flowing into the double-pipe heat exchanger after branching a refrigerant flowing from the outdoor unit toward the plurality of indoor units, and joining together a refrigerant decompressed by a first expansion valve and a refrigerant flowing from the plurality of indoor units, the low-pressure two-phase refrigerant being relatively low in temperature and flowing out of the double-pipe heat exchanger toward the outdoor unit after branching a refrigerant flowing out of the double-pipe heat exchanger toward the plurality of indoor units, and decompressing a part of the branched refrigerant by a second expansion valve.

Effect of the Invention

[0013] According to the present invention, significant reduction in number of connection pipes is realized between the outdoor unit and the branch controller, and between the branch controller and each of the indoor units. At the same time, a COP in simultaneous heating and cooling operation is improved because a large enthalpy difference is secured on the side of the indoor units in cooling operation.

Brief Description of the Drawings

[0014] [FIG 1] A diagram of a refrigerant circuit in cooling-dominant operation of an air conditioner according to a first embodiment of the present invention.

[FIG. 2] A Mollier chart to be used for description of the air conditioner according to the first embodiment of the present invention.
[FIG. 3] A diagram of a refrigerant circuit in heating-dominant operation of the air conditioner according to the first embodiment of the present invention.
[FIG. 4] A control flow chart for a first expansion valve 211 in cooling-dominant operation of an air conditioner according to a second embodiment of the present invention.
[FIG. 5] A control flow chart for the first expansion valve 211 in full-heating operation and heating-dominant operation of the air conditioner according to the second embodiment of the present invention.
[FIG. 6] A control flow chart for a second expansion valve 212 in full-cooling operation and cooling-dominant operation of an air conditioner according to a third embodiment of the present invention.
[FIG. 7] A control flow chart for the second expansion valve 212 in heating-dominant operation of the air conditioner according to the third embodiment of the present invention.
[FIG. 8] A Mollier chart to be used for description of the air conditioner according to the third embodiment of the present invention.

Best Mode for carrying out the Invention

First embodiment

[0015] FIG 1 is a diagram of a refrigerant circuit in cooling-dominant operation of an air conditioner according to a first embodiment of the present invention. In the air conditioner illustrated in FIG. 1, an outdoor unit 100 and a plurality of indoor units 301 to 303 are connected through a branch controller 200, and a single refrigerating cycle is established by using a supercritical fluid. The outdoor unit 100 mainly includes a compressor 110, a four-way valve 120, a heat-source side heat exchanger 130, and check valves 141 to 147. The indoor units 301 to 303 respectively include use-side (load-side) heat exchangers 311 to 313, and expansion valves 321 to 323 serving as restriction devices. Further, the branch controller 200 mainly includes a first expansion valve 211, a second expansion valve 212, check valves 231 to 233, channel switching valves 221 to 223, and a double-pipe heat exchanger 240. It should be noted that the double-pipe heat exchanger 240 may be a plate heat exchanger or a microchannel heat exchanger.
[0016] Here, two pipes of a high-pressure pipe 400 and a low-pressure pipe 500 connect the outdoor unit 100 and the branch controller 200, while two pipes of a high-pressure pipe 700 and a low-pressure pipe 800 similarly connect

the branch controller 200 and each of the indoor units 301 to 303. Cooling-dominant operation mainly involving cooling operation and partially involving heating operation is herein described. As to heating-dominant operation, channels are switched by means of the four-way valve 120 and the check valves 141 to 147.

[0017] It should be noted that, though FIG. 1 illustrates a high-pressure detection means 281, a medium-pressure detection means 282, a first temperature detection means 291, and a second temperature detection means 292 that are included in the branch controller 200, those components are used in a second embodiment described later but not necessary in the first embodiment.

[0018] First, with reference to FIG. 1, description is given of a flow of the refrigerant circuit in the cooling-dominant operation. The description is given herein by taking a case of CO_2 used as the supercritical fluid. A high-pressure and high-temperature fluid that is compressed by the compressor 110 passes through the four-way valve 120 and is subjected to heat exchange with the ambient air by the heat-source side heat exchanger 130. The fluid is cooled down to temperature which does not reach the ambient air temperature, for example, temperature at which the dryness of a Mollier chart (pressure p-enthalpy h) illustrated in FIG. 2 becomes approximately 0.5 (point B of FIG. 2). Then, the fluid of the outlet of the heat-source side heat exchanger 130 enters into a state of high pressure and medium temperature. The fluid that flows out of the heat-source side heat exchanger 130 then flows into the branch controller 200 through the high-pressure pipe 400, and is branched into the indoor units 302 and 303 in cooling operation and the indoor unit 301 in heating operation by the respective channel switching valves 221 to 223.

[0019] As to the refrigerant for the side of the indoor unit 301 in heating operation, the high-pressure and medium-temperature fluid that flows into the load-side heat exchanger 311 from a branch port through the channel switching valve 223 is further subjected to heat exchange with room temperature to become a high-pressure and medium-temperature fluid, which has temperature substantially equal to the room temperature (point C of FIG. 2). Then, the fluid is decompressed by the expansion valve 321 (point D of FIG. 2). The refrigerant that flows out of the indoor unit 301 in heating operation through the low-pressure pipe 800 passes through the check valve 231 of the branch controller 200 in the state of medium pressure and medium temperature, and joins another refrigerant at a point between the first expansion valve 211 and the double-pipe heat exchanger 240.

[0020] Meanwhile, the refrigerant for the side of the indoor units 302 and 303 in cooling operation flows from the branch port and is decompressed by the first expansion valve 211 down to medium pressure in the supercritical range, which is slightly lower than high pressure (point E of FIG. 2). The refrigerant then flows into a medium-temperature side of the double-pipe heat exchanger 240 in the state of medium pressure and medium temperature. Further, the medium-pressure and medium-temperature fluid that is decompressed by the expansion valve 321 of the indoor unit 301 in heating operation joins this fluid at this point, and then flows into the medium-temperature side of the double-pipe heat exchanger 240. In this case, a part of the fluid that flows out of the medium-temperature side of the double-pipe heat exchanger 240 is further branched at a branch port, and is decompressed by the second expansion valve 212 to become a low-pressure and low-temperature fluid having two phases of gas and liquid (point I of FIG. 2). Then, the fluid flows into a low-temperature side of the double-pipe heat exchanger 240.

[0021] The low-pressure and low-temperature fluid on the low-temperature side enters into a state of low pressure and medium temperature, and a high dryness (point H of FIG. 2) after being subjected to heat exchange with the medium-pressure and medium-temperature fluid on the medium-temperature side in the double-pipe heat exchanger 240. On the other hand, the medium-pressure and medium-temperature fluid on the medium-temperature side is further cooled to become a medium-pressure and medium-temperature fluid in a state of a low enthalpy (point D of FIG. 2). The medium-pressure and medium-temperature fluid that is further cooled (point D of FIG. 2) is then further decompressed by the load-side expansion valves 322 and 323 to become a low-pressure and low-temperature fluid having two phases of gas and liquid (point G of FIG. 2). Then, the fluid flows into the load-side heat exchangers 312 and 313 for heat exchange with the room temperature, to thereby enter into the state of low pressure and medium temperature, and a high dryness (point H of FIG. 2). Finally, the low-pressure and medium-temperature fluid that flows out of the low-temperature side of the double-pipe heat exchanger 240 and the low-pressure and medium-temperature fluid that flows out of the load-side heat exchangers 312 and 313 join each other, and return to the outdoor unit 100 side through the low-pressure pipe 500.

[0022] With this configuration, significant reduction in number of connection pipes is realized between the outdoor unit 100 and the branch controller 200, and between the branch controller 200 and each of the indoor units 301 to 303. At the same time, a COP in simultaneous heating and cooling operation is improved because the large enthalpy difference is secured on the side of the indoor units 302 and 303 in cooling operation.

[0023] Next, FIG. 3 is a diagram of a refrigerant circuit in heating-dominant operation of the air conditioner according to the first embodiment of the present invention. The air conditioner illustrated in FIG. 3 has the same configuration as the configuration of the first embodiment illustrated in FIG. 1.

[0024] With reference to FIG. 3, description is given of a flow of the refrigerant circuit in the heating-dominant operation. A high-pressure and high-temperature fluid that is compressed by the compressor 110 flows into the branch controller 200 through the four-way valve 120, the check valve 145, and the high-pressure pipe 400. Then, the high-pressure and high-temperature fluid is branched into the indoor unit 303 in cooling operation and the indoor units 301 and 302 in heating operation at the respective channel switching valves 221 to 223 of the branch controller 200. Further, the first expansion valve 211 is fully closed to block the flow.

[0025] The refrigerant for the side of the indoor units 301 and 302 in heating operation flows into the load-side heat exchangers 311 and 312 from the branch port of the branch controller 200 through the channel switching valves 222 and 223 and the high-pressure pipes 700, and the high-pressure and medium-temperature fluid is further subjected to heat exchange with the room temperature to become a high-pressure and medium-temperature fluid, which has temperature substantially equal to the room temperature (point C of FIG. 2). Then, the fluid is decompressed by the expansion valves 321 and 322 to become a medium-pressure and medium-temperature fluid (point D of FIG. 2). The refrigerant that is decompressed by the expansion valves 321 and 322 then flows into the branch controller 200 through the low-pressure pipes 800, and passes through the check valves 231 and 232 in the state of medium pressure and medium temperature to join another refrigerant at the point between the first expansion valve 211 and the double-pipe heat exchanger 240.

[0026] Meanwhile, the refrigerant for the side of the indoor unit 303 in cooling operation flows into the load-side expansion valve 323 through the following path. The fluid that flows into the medium-pressure and medium-temperature side of the double-pipe heat exchanger 240 from the indoor units 301 and 302 in heating operation through the low-pressure pipes 800 and the check valves 231 and 232 is further branched at the branch port, and a part of the fluid is further decompressed by the second expansion valve 212 to become a low-pressure and low-temperature fluid (point I of FIG. 2). The fluid then flows into the low-temperature side of the double-pipe heat exchanger 240. Then, the fluid is subjected to heat exchange with the medium-pressure and medium-temperature fluid on the medium-temperature side by the double-pipe heat exchanger 240. As a result, the low-pressure and low-temperature fluid on the low-temperature side enters into the state of low pressure and medium temperature, and a high dryness (point H of FIG. 2), while the medium-pressure and medium-temperature fluid on the medium-temperature side is further cooled to become a medium-pressure and medium-temperature fluid in a state of a low enthalpy (point D of FIG. 2).

[0027] The medium-pressure and medium-temperature fluid that is further cooled (point D of FIG. 2) is then further decompressed by the load-side expansion valve 323 to become a low-pressure and low-temperature fluid. Then, the fluid flows into the load-side heat exchanger 313 for heat exchange with the room temperature, to thereby enter into the state of low pressure and medium temperature, and a high dryness (point H of FIG. 2). Finally, the low-pressure and medium-temperature fluid that flows out of the low-temperature side of the double-pipe heat exchanger 240 and the low-pressure and medium-temperature fluid that flows out of the load-side heat exchanger 313 join each other, and return to the outdoor unit 100 side through the low-pressure pipe 500, the heat-source side heat exchanger 130, and the four-way valve 120.

[0028] As described above, according to the first embodiment, the single outdoor unit 100 and the single branch controller 200 are connected through two pipes, and the branch controller 200 and each of the plurality of indoor units 301 to 303 are connected through two pipes. Accordingly, significant reduction in number of connection pipes is realized between the branch controller 200 and each of the indoor units 301 to 303, and at the same time, the COP in simultaneous heating and cooling operation is improved because the large enthalpy difference is secured on the side of the indoor units 302 and 303 in cooling operation. In addition, power-save operation is realized also in the cooling-dominant operation mainly involving cooling and partially involving heating operation.

Second embodiment

[0029] A configuration of the second embodiment is the same as the configurations of the first embodiment illustrated in FIGS. 1 and 3. Further, in FIGS. 1 and 3, the high-pressure detection means 281, the medium-pressure detection means 282, the first temperature detection means 291, and the second temperature detection means 292 are provided to the branch controller 200, which are unnecessary in the first embodiment.

[0030] The flow of the refrigerant of the second embodiment is the same as that of the first embodiment. Hereinbelow, a control method for the first expansion valve 211 is described. First, Table 1 shows overviews of control in each control mode (full cooling, cooling dominant, full heating, or heating dominant).

[Table 1]

[0031] List of overviews of expansion valve control

Control mode	First expansion valve 211	Second expansion valve 212
Full cooling	Fully opened	After initial degree of opening, control according to temperature difference ΔT

(continued)

Control mode	First expansion valve 211	Second expansion valve 212
Cooling dominant	After initial degree of opening, control according to pressure difference ΔP	↑
Full heating	Fully closed	Fully opened
Heating dominant	After initial degree of opening, control according to pressure difference ΔP	After fully closed for initial degree of opening, control according to ΔP

[0032] In the case of the full-cooling operation (hereinafter, abbreviated as "full cooling") in which all the indoor units 301 to 303 perform cooling operation, the first expansion valve 211 is fully opened and the flow rate is controlled based on loads only by the expansion valves 321 to 323 of the indoor units 301 to 303.

[0033] In FIG. 1, there are provided temperature sensors 311a, 312a, and 313a for detecting temperatures on the upper side of the load-side heat exchangers 311 to 313, respectively. There are also provided temperature sensors 311b, 312b, and 313b for detecting temperatures and pressure sensors 311c, 312c, and 313c for detecting pressures between the load-side heat exchangers 311 to 313 and the load-side expansion valves 321 to 323, respectively.

[0034] In the case of the full-cooling operation, temperature differences between the temperature sensors 311a, 312a, and 313a, and the temperature sensors 311b, 312b, and 313b are calculated, respectively, and calculation results thereof are each set as a degree of superheat. The degree of opening of each of the load-side expansion valves 321, 322, and 323 is adjusted so that the degree of superheat becomes a predetermined value, for example, approximately 2°C.

[0035] In the case of the "cooling-dominant" operation in which the simultaneous heating and cooling operation is performed in the state of the cooling cycle, pressure difference control, which is described later, is performed by using a pressure difference between the high-pressure detection means 281 and the medium-pressure detection means 282. Also in the cooling-dominant operation, when the load-side heat exchangers operate as evaporators, the above-mentioned degree of superheat is detected and the degree of opening of each of the load-side expansion valves is adjusted so that the degree of superheat becomes a predetermined value.

[0036] Further, in the case of the full-heating operation (hereinafter, abbreviated as "full heating") in which all the indoor units 301 to 303 perform heating operation, the first expansion valve 211 is fully closed basically and the flow rate is controlled based on loads only by the expansion valves 321 to 323 of the indoor units. Further, the pressure difference control, which is described later, is performed by using the pressure difference between the high-pressure detection means 281 and the medium-pressure detection means 282.

[0037] The temperature sensors 311b, 312b, and 313b for detecting temperatures and the pressure sensors 311c, 312c, and 313c for detecting pressures are provided between the load-side heat exchangers 311 to 313 and the load-side expansion valves 321 to 323, respectively.

[0038] In the case of the full-heating operation, pressure values obtained through detection by the pressure sensors 311c, 312c, and 313c are used for calculating a saturation temperature T_{sc}. The calculated saturation temperature is set as a condensation temperature T_c. A relational expression between the saturation temperature T_{sc} and a pressure P as in Expression (1) needs to be prepared in advance:

$$T_{sc}=f(P) \qquad (1)$$

[0039] It should be noted that, as illustrated in FIG. 8 which is referenced later, in a case where carbon dioxide is used as a refrigerant, the high-pressure side operates above the critical point, and hence there is no phase change. In other words, there exists no saturation temperature T_{sc}. In view of the above, an experiment for the refrigerating cycle, or the like is conducted to set a virtual saturation temperature based on balanced pressure and intake air temperature. For example, it is assumed that, when the pressure is 100 kgf/cm^2, the saturation temperature is 45°C. Expression (1) according to this embodiment is an arithmetic expression by which the virtual saturation temperature T_{sc} is calculated.

[0040] A virtual condensation temperature T_c is calculated by using Expression (1) based on the pressure values obtained by the pressure sensors 311c, 312c, and 313c. A difference between a temperature T_L obtained by each of the temperature sensors 311b, 312b, and 313b and the virtual condensation temperature T_c ($Tc-T_L$) is obtained, and this value is set as a degree of subcooling SC. The degree of opening of each of the load-side expansion valves 321, 322, and 323 is adjusted so that the degree of subcooling SC becomes a predetermined value, for example, approximately 5°C.

[0041] Further, similarly to the case of the full heating, in the case of the "heating-dominant" operation in which the simultaneous heating and cooling operation is performed in the state of the heating cycle, the first expansion valve 211 is fully closed basically, and the pressure difference control, which is described later, is performed by using the pressure difference between the high-pressure detection means 281 and the medium-pressure detection means 282.

[0042] Next, with reference to control flow charts illustrated in FIGS. 4 and 5, the control method is described in detail. First, in the "full-cooling" operation, the first expansion valve 211 is fully opened constantly and the flow rate is controlled based on loads only by the expansion valves 321 to 323 of the indoor units.

[0043] In the "cooling-dominant" operation, as illustrated in FIG. 4, start up of the compressor 110, or the like triggers the first expansion valve 211 to start with an initial degree of opening L0 that is set in advance (Step S41). When a predetermined period U has elapsed from the start (Step S42), the degree of opening of the first expansion valve 211 is controlled according to comparison between the pressure difference ΔP obtained based on detection values of the high-pressure detection means 281 and the medium-pressure detection means 282, and set values P1 and P2 ($P1<P2$) that are set in advance.

[0044] For example, in a case where $\Delta P>P2$, the degree of opening of the first expansion valve 211 is increased by a predetermined degree of opening that is set in advance (Steps S43→S44). In a case where $P1\leq\Delta P\leq P2$, the degree of opening of the first expansion valve 211 is maintained at the current degree of opening (Steps S43→S45→S46). In a case where $\Delta P<P1$, the degree of opening of the first expansion valve 211 is decreased by a predetermined degree of opening that is set in advance (Steps S43→S45→S47→S48).

[0045] Through the control described above, it is possible to secure the pressure difference necessary to cause the refrigerant to flow at a flow rate according to the load on the side of the indoor unit in heating operation, and to decrease the pressure into low pressure due to an excess pressure difference, which results in suppression of decline in COP.

[0046] Further, in the "full-heating" operation, the first expansion valve 211 is fully closed constantly and the flow rate is controlled based on loads only by the expansion valves 321 to 323 of the indoor units.

[0047] In the "heating-dominant" operation, as illustrated in FIG. 5, start up of the compressor 110, or the like triggers the first expansion valve 211 to start with the fully closed state (Step S51). When the predetermined period U has elapsed from the start (Step S52), the degree of opening of the first expansion valve 211 is controlled according to comparison between the pressure difference ΔP obtained based on detection values of the high-pressure detection means 281 and the medium-pressure detection means 282, and the set values P1 and P2 ($P1<P2$) that are set in advance.

[0048] For example, in a case where $\Delta P>P2$, the degree of opening of the first expansion valve 211 is increased by a predetermined degree of opening that is set in advance (Steps S53→S54). In a case where $P1\leq\Delta P\leq P2$, the degree of opening of the first expansion valve 211 is maintained at the current degree of opening (Steps S53→S55→S56). In a case where $\Delta P<P1$, the degree of opening of the first expansion valve 211 is decreased by a predetermined degree of opening that is set in advance (Steps S53→S55→S57→S58).

[0049] Through the control described above, it is possible to secure the pressure difference necessary to cause the refrigerant to flow at a flow rate according to the load on the side of the indoor unit in heating operation, and to secure the pressure difference necessary to cause the refrigerant to flow at a flow rate according to the load on the side of the indoor unit in cooling operation by approximating the pressure of the inlet of the indoor unit in cooling operation to the low pressure due to an excess pressure difference, which results in suppression of decline in COP. In addition, the change in pressure may be suppressed, and hence the refrigerant may stably be conveyed to the indoor unit, to thereby realize power-save operation and comfort.

Third embodiment

[0050] As in the second embodiment described above, a configuration of a third embodiment is the same as the configurations of the first embodiment illustrated in FIGS. 1 and 3. Further, in FIGS. 1 and 3, the high-pressure detection means 281, the medium-pressure detection means 282, the first temperature detection means 291, and the second temperature detection means 292 are provided to the branch controller 200, which are unnecessary in the first embodiment.

[0051] The flow of the refrigerant of the third embodiment is the same as that of the first embodiment. Hereinbelow, a control method for the second expansion valve 212 is described. First, Table 1 shows overviews of control in each control mode (full cooling, cooling dominant, full heating, or heating dominant).

[0052] In the case of the "full-cooling" operation in which all the indoor units 301 to 303 perform cooling operation, for the second expansion valve 212, temperature difference (superheat) control, which is described later, is performed by using the first temperature detection means 291 and the second temperature detection means 292, and the flow rate is controlled based on loads by the expansion valves 321 to 323 on the side of the indoor units 301 to 303. The same control as in the case of the "full cooling" applies to the case of the "cooling-dominant" operation in which the simultaneous heating and cooling operation is performed in the state of the cooling cycle.

[0053] Further, in the case of the "full-heating" operation in which all the indoor units 301 to 303 perform heating operation, the second expansion valve 212 is fully opened and the flow rate is controlled based on loads by the expansion valves 321 to 323 of the indoor units. Then, the refrigerant subjected to heat exchange with the load side flows into the low-pressure line of the outdoor unit through the second expansion valve 212.

[0054] In the case of the "heating-dominant" operation in which the simultaneous heating and cooling operation is performed in the state of the heating cycle, the second expansion valve 212 is fully closed basically, and the pressure difference control, which is described later, is performed by using the pressure difference between the high-pressure detection means 281 and the medium-pressure detection means 282.

[0055] Next, with reference to control flow charts illustrated in FIGS. 6 and 7, the control method is described in detail. First, in the "full-cooling" operation, as illustrated in FIG. 6, start up of the compressor 110, or the like triggers the second expansion valve 212 to start with the initial degree of opening L0 that is set in advance (Step S61). When the predetermined period U has elapsed from the start (Step S62), a temperature difference ΔT (degree of superheat) is calculated based on detection values of the first temperature detection means 291 and the second temperature detection means 292, and the degree of opening of the second expansion valve 212 is controlled according to comparison between the temperature difference ΔT and values T1 and T2 (T1<T2) that are set in advance.

[0056] For example, in a case where $\Delta T > T2$, the degree of opening of the second expansion valve 212 is increased by a predetermined degree of opening that is set in advance (Steps S63→S64). In a case where $T1 \leq \Delta T \leq T2$, the degree of opening of the second expansion valve 212 is maintained at the current degree of opening (Steps S63→S65→S66). In a case where $\Delta T < T1$, the degree of opening of the second expansion valve 212 is decreased by a predetermined degree of opening that is set in advance (Steps S63→S65→S67→S68).

[0057] Through the control described above, it is possible to secure the enthalpy difference necessary to lower the temperature of the refrigerant in the inlet on the side of the indoor unit in cooling operation, to thereby obtain satisfactory performance, which results in suppression of decline in COP. In addition, also in the cooling-dominant operation mainly involving cooling and partially involving heating operation, a lower-temperature refrigerant may be conveyed to the indoor unit in cooling operation, and power-save operation can accordingly be realized.

[0058] Further, in the "full-heating" operation, the second expansion valve 212 is fully opened constantly and the flow rate is controlled based on loads by the expansion valves 321 to 323 of the indoor units. Then, the refrigerant subjected to heat exchange with the load side flows into the low-pressure line of the outdoor unit through the second expansion valve 212.

[0059] In the "heating-dominant" operation, as illustrated in FIG. 7, start up of the compressor 110, or the like triggers the second expansion valve 212 to start with the fully closed state (Step S71). When the predetermined period U has elapsed from the start (Step S72), the degree of opening of the second expansion valve 212 is controlled according to comparison between the pressure difference ΔP obtained based on detection values of the high-pressure detection means 281 and the medium-pressure detection means 282, and the set values P1 and P2 (P1<P2) that are set in advance.

[0060] For example, in a case where $\Delta P > P2$, the degree of opening of the second expansion valve 212 is decreased by a predetermined degree of opening that is set in advance (Steps S73→S74). In a case where $P1 \leq \Delta P \leq P2$, the degree of opening of the second expansion valve 212 is maintained at the current degree of opening (Steps S73→S75→S76). In a case where $\Delta P < P1$, the degree of opening of the second expansion valve 212 is increased by a predetermined degree of opening that is set in advance (Steps S73→S75→S77→S78).

[0061] Through the control described above, it is possible to secure the pressure difference necessary to cause the refrigerant to flow at a flow rate according to the load on the side of the indoor unit in heating operation, and to approximate the pressure of the inlet of the indoor unit in cooling operation to the low pressure due to an excess pressure difference (approximation of medium pressure to low pressure), which results in suppression of decline in COP caused by the fact that the pressure difference necessary to cause the refrigerant to flow at a flow rate according to the load on the side of the indoor unit in cooling operation cannot be secured.

[0062] At the time of the cooling-dominant operation, the heat-source side heat exchanger 130 operates as a condenser (radiator). In the cooling-dominant operation, the cooling load exceeds the heating load, and hence a part of the heat radiation capability needs to be supplemented by the heat-source side heat exchanger 130. Therefore, the heat exchanger capacity needs to be increased and decreased by adjusting the fan speed and dividing the heat-source side heat exchanger 130.

[0063] According to the present invention, description is given of a method of adjusting the heat radiation capability without dividing the heat-source side heat exchanger 130. In the embodiments of the present invention, carbon dioxide is used as the refrigerant. The high-pressure side operates above the critical point of this refrigerant as illustrated in FIGS. 2 and 8. With this characteristic, the heat radiation capacity can be adjusted with ease.

[0064] As illustrated in FIG. 3, there are provided a pressure sensor 900 and a temperature sensor 901 between the heat-source side heat exchanger 130 and the check valve 141, and a temperature sensor 902 to the inlet of the heat-source side heat exchanger 130. In the supercritical state, when the temperature and the pressure are determined, the enthalpy is uniquely determined.

[0065] In FIG 8, "a" represents an enthalpy H_1 of the inlet of the heat-source side heat exchanger 130; "b", an enthalpy H_2 of the outlet of the heat-source side heat exchanger 130 (inlet of the load-side heat exchanger in heating operation);

and "c", an enthalpy H_3 of the inlet of the heat-source side heat exchanger 130.

[0066] The load on the heating side may be grasped from the number of indoor units in heating operation and the capacity of each of the connected indoor units. This heating load is denoted by Q_c. Further, a refrigerant flow rate G_r may be obtained based on discharge pressure and suction pressure of the compressor. An enthalpy difference ΔH necessary to cover the heating load may be obtained from Expression (2):

$$Q_c/G_r=\Delta H=H_2-H_3 \qquad (2)$$

[0067] Further; the enthalpy H_3 of the outlet of the load-side heat exchanger (heating) may be determined by using the pressure sensor 311c. and the temperature sensor 311b provided to the outlet of the load-side heat exchanger. Expression (2) is converted into Expression (3):

$$H_2=Q_c/G_r+H_3 \qquad (3)$$

[0068] In other words, the enthalpy H_2 of the outlet of the heat-source side heat exchanger is determined. The enthalpy of the outlet of the heat-source side heat exchanger may be obtained by using the pressure sensor 900 and the temperature sensor 901. The enthalpy obtained from Expression (3) is set as a target enthalpy H_{2m}.

[0069] Further, the enthalpy measured by the pressure sensor 900 and the temperature sensor 901 is denoted by H_{2s}. A difference between the target enthalpy H_{2m} and the measured enthalpy H_{2s} (H_{2m}-H_{2s}) is calculated, and $H_{2m}-H_{2s}=\Delta H_s$ holds.

[0070] Then, a control means increases and decreases the rotation of the heat-source side fan (blower) so that $-epsH_2<\Delta H_s<epsH_2$ (it should be noted that $-epsH_2$ and $epsH_2$ denote a lower limit value and an upper limit value in an error range, respectively) becomes a predetermined value. Control is performed so that, in a case where $-epsH_2>\Delta H_s$, the number of rotations of the fan is increased, while in a case where $\Delta H_s>epsH_2$, the number of rotations of the fan is decreased.

[0071] It should be noted that, in order to obtain an enthalpy H, a physical property expression as in Expression (4) needs to be prepared in advance:

$$H=f(P,T) \qquad (4)$$

where P denotes pressure and T denotes temperature.

Claims

1. An air conditioner in which an outdoor unit (100) and a plurality of indoor units (301-303) are connected through a branch controller (200), and a supercritical fluid is used, to thereby establish a single refrigerating cycle,
the outdoor unit (100) and the branch controller (200) being connected through two pipes of a high-pressure pipe (400) and a low-pressure pipe (500),
the branch controller (200) and each of the plurality of indoor units (301-303) being connected through two pipes of a high-pressure pipe (700) and a low-pressure pipe (800),
wherein the branch controller (200) comprises a double-pipe heat exchanger (240) for heat exchange between a medium-pressure two-phase refrigerant and a low-pressure two-phase refrigerant,
the medium-pressure two-phase refrigerant being relatively high in temperature and flowing into the double-pipe heat exchanger (240) after branching a refrigerant flowing from the outdoor unit (100) toward the plurality of indoor units (301-303), and joining together a refrigerant decompressed by a first expansion valve (211) and a refrigerant flowing from the plurality of indoor units (301-303),
the low-pressure two-phase refrigerant being relatively low in temperature and flowing out of the double-pipe heat exchanger (240) toward the outdoor unit (100) after branching a refrigerant flowing out of the double-pipe heat exchanger (240) toward the plurality of indoor units (301-303), and decompressing a part of the branched refrigerant by a second expansion valve (212).

2. The air conditioner according to claim 1, wherein the double-pipe heat exchanger (240) comprises a plate heat exchanger or a microchannel heat exchanger.

3. The air conditioner according to claim 1, comprising pressure detection means (281, 282) provided to an outlet and an inlet of the first expansion valve (211), respectively,
wherein a degree of opening of the first expansion valve (211) is controlled so that a pressure difference between two pressure detection values of the pressure detection means (281, 282) is constant.

4. The air conditioner according to claim 1, comprising temperature detection means (291, 292) provided to an outlet of the second expansion valve (212) and an outlet of the double-pipe heat exchanger (240) on a low-pressure side, respectively,
wherein a degree of opening of the second expansion valve (212) is controlled so that a temperature difference between two temperature detection values of the temperature detection means (291, 292) is constant.

5. The air conditioner according to claim 1, comprising pressure detection means (281, 282) provided to an outlet and an inlet of the first expansion valve (211), respectively,
wherein a degree of opening of the second expansion valve (212) is controlled so that a pressure difference between two pressure detection values of the pressure detection means (281, 282) is constant.

6. The air conditioner according to claim 1, comprising:

a pressure detection means (900) and a temperature detection means (901) provided to an outlet of a heat-source side heat exchanger (130); and
a control means for calculating an enthalpy based on a value of the pressure detection means (900) and a value of the temperature detection means (901),
wherein the air conditioner increases and decreases rotation speed of a heat-source side blower so that a target enthalpy is obtained.

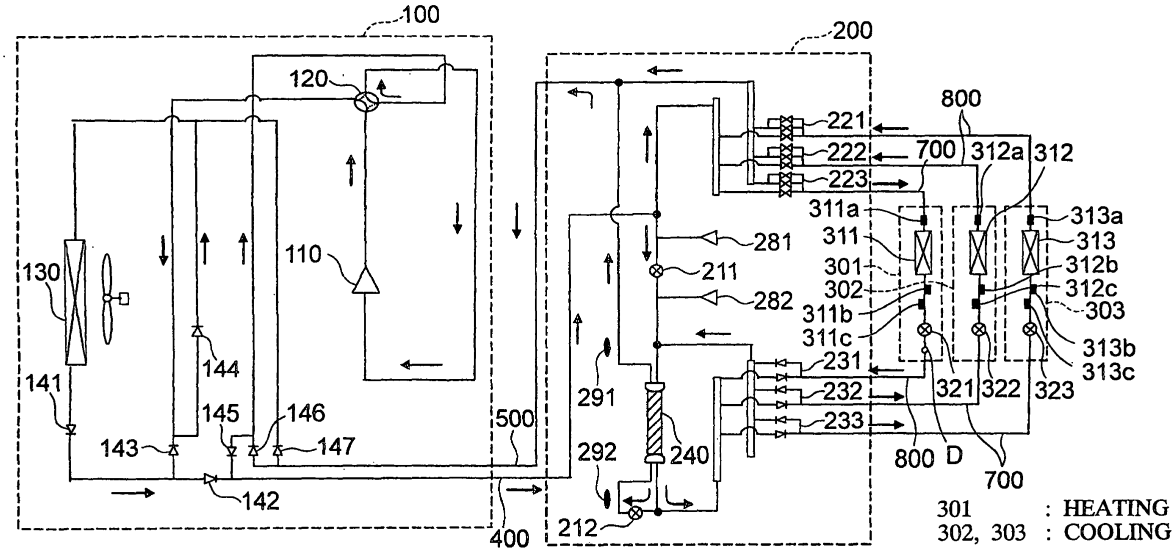
100
200
120
110
130
141
142
143
144
145
146
147
500
400
800
700
221
222
223
211
212
240
231
232
233
281
282
291
292
311a
311
301
302
311b
311c
312a
312
312b
312c
313a
313
303
313b
313c
321
322
323
D
301 : HEATING
302, 303 : COOLING

FIG. 1

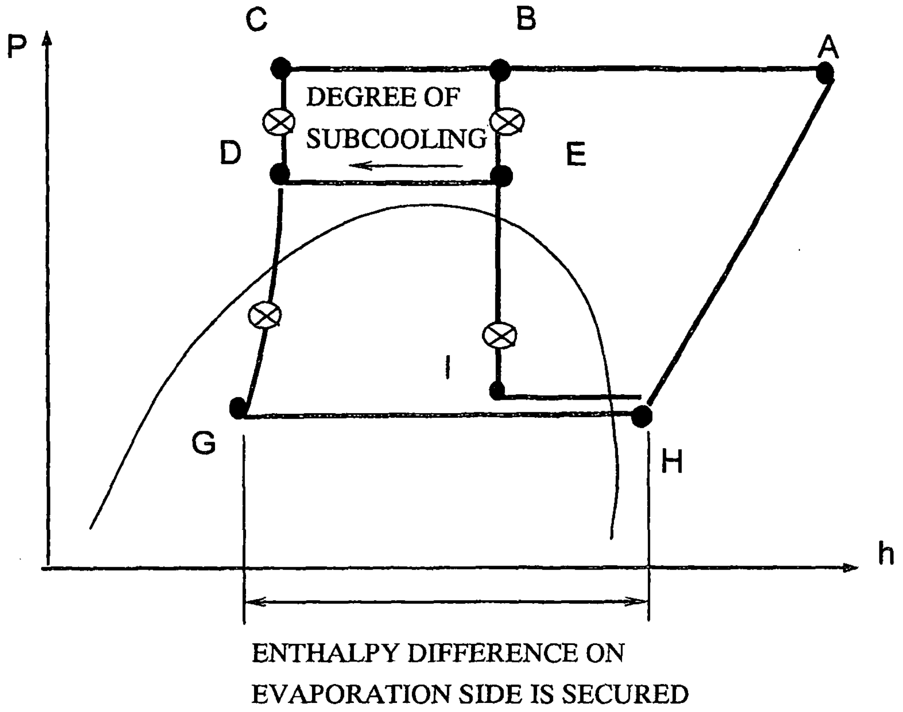
P
C
B
A
DEGREE OF
SUBCOOLING
D
E
I
G
H
h
ENTHALPY DIFFERENCE ON
EVAPORATION SIDE IS SECURED

FIG. 2

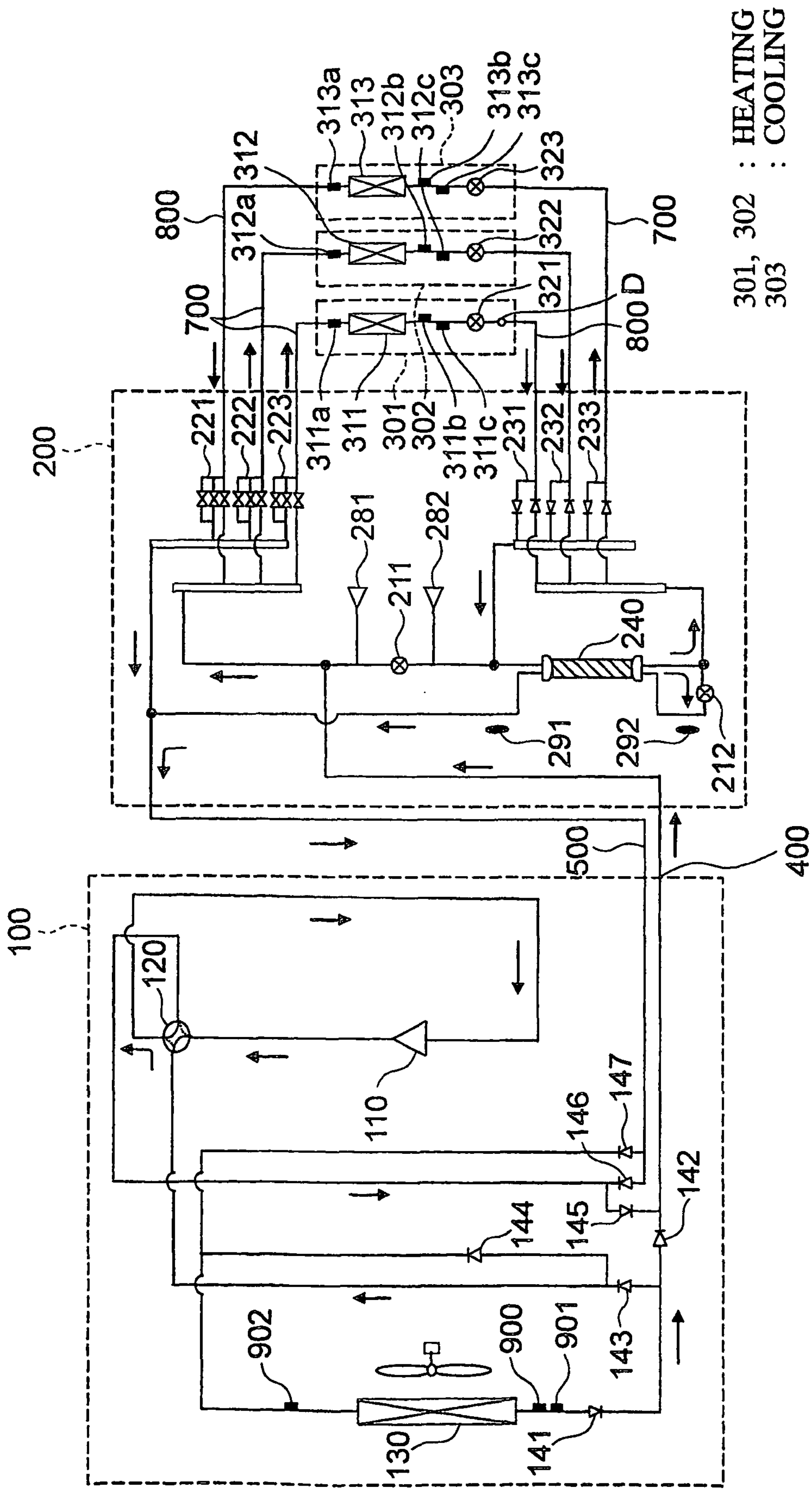
100
110
120
130
141
142
143
144
145
146
147
200
211
212
221
222
223
231
232
233
240
281
282
291
292
301
302
303
311
311a
311b
311c
312
312a
312b
312c
313
313a
313b
313c
321
322
323
400
500
700
800
800 D
900
901
902
301, 302 : HEATING
303 : COOLING

FIG. 3

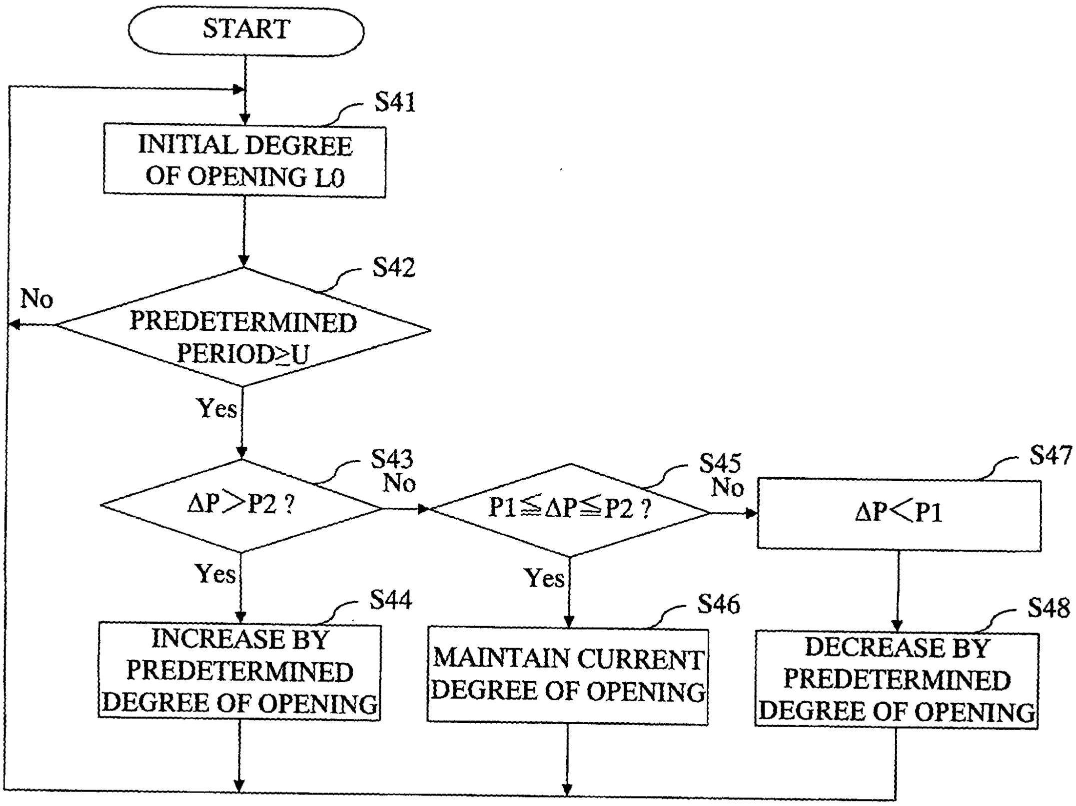
START
S41
INITIAL DEGREE OF OPENING L0
S42
No
PREDETERMINED PERIOD≥U
Yes
S43
No
ΔP>P2 ?
S45
No
P1≦ΔP≦P2 ?
S47
ΔP<P1
Yes
S44
INCREASE BY PREDETERMINED DEGREE OF OPENING
Yes
S46
MAINTAIN CURRENT DEGREE OF OPENING
S48
DECREASE BY PREDETERMINED DEGREE OF OPENING

FIG. 4

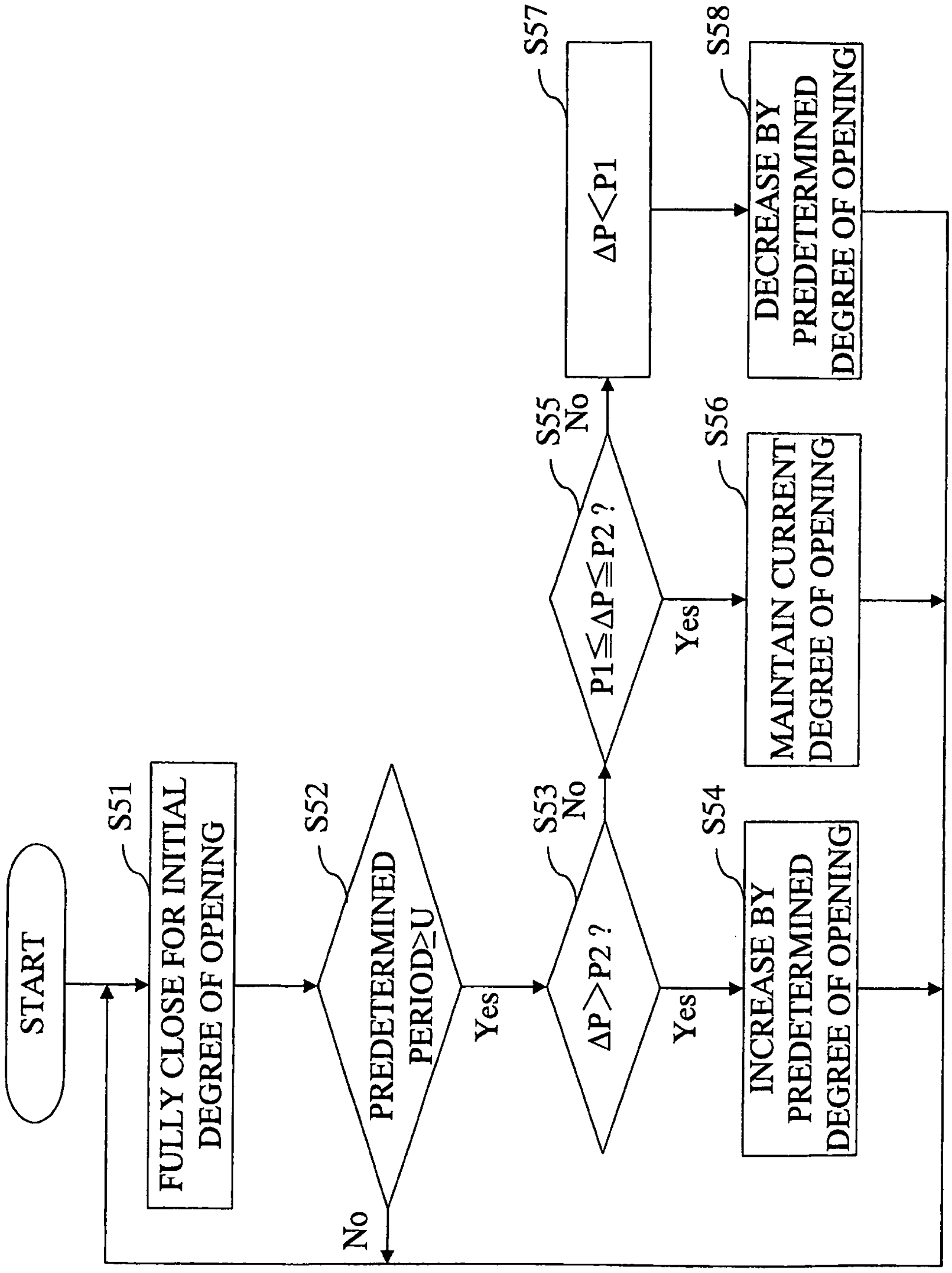
START
S51
FULLY CLOSE FOR INITIAL DEGREE OF OPENING
S52
PREDETERMINED PERIOD≥U
No
Yes
S53
ΔP>P2 ?
No
Yes
S54
INCREASE BY PREDETERMINED DEGREE OF OPENING
S55
P1≦ΔP≦P2 ?
No
Yes
S56
MAINTAIN CURRENT DEGREE OF OPENING
S57
ΔP<P1
S58
DECREASE BY PREDETERMINED DEGREE OF OPENING

FIG. 5

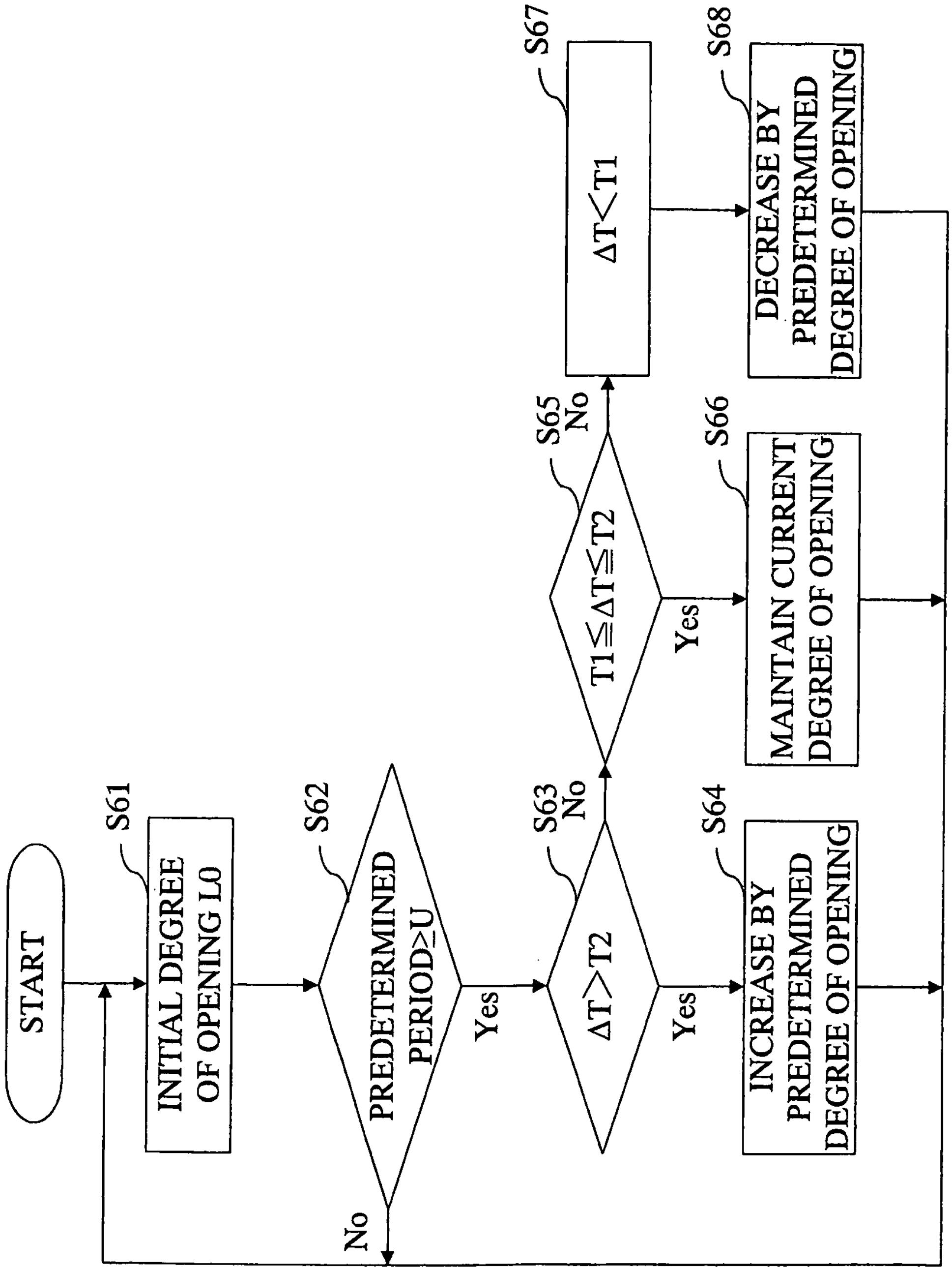
START
S61
INITIAL DEGREE OF OPENING L0
S62
PREDETERMINED PERIOD≥U
No
Yes
S63
ΔT>T2
No
Yes
S64
INCREASE BY PREDETERMINED DEGREE OF OPENING
S65
T1≦ΔT≦T2
No
Yes
S66
MAINTAIN CURRENT DEGREE OF OPENING
S67
ΔT<T1
S68
DECREASE BY PREDETERMINED DEGREE OF OPENING

FIG. 6

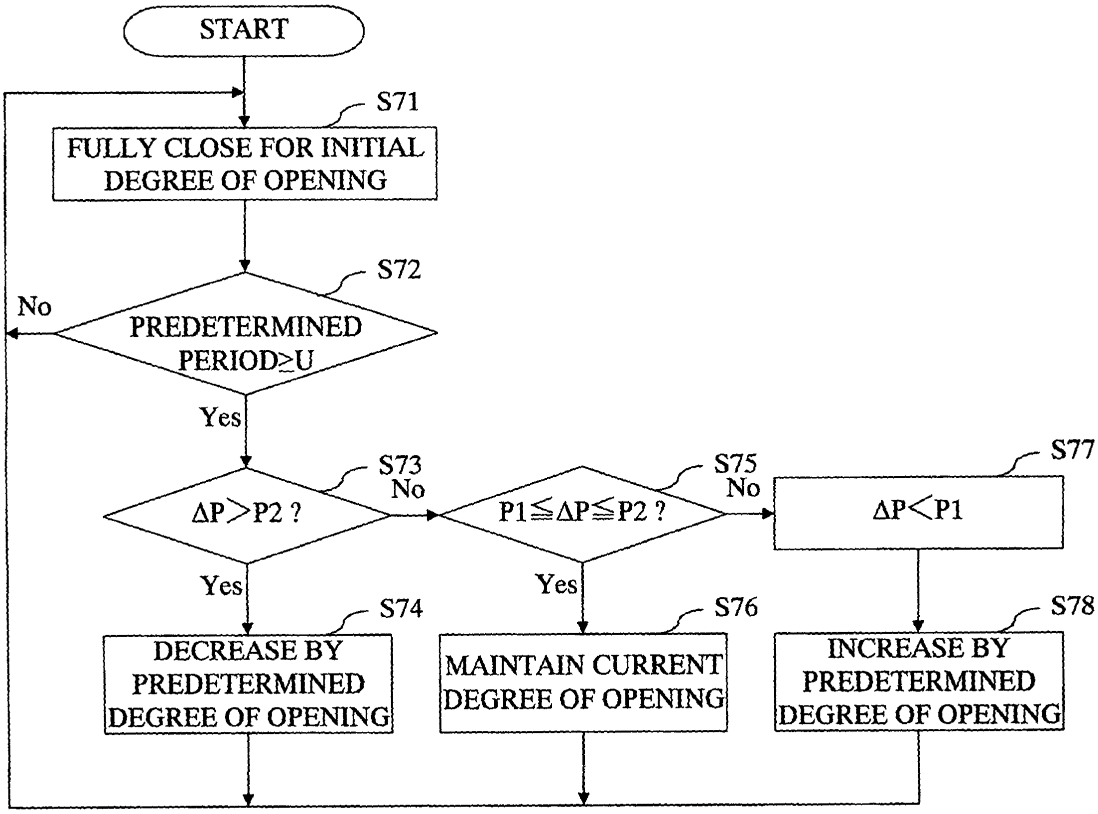
START
S71
FULLY CLOSE FOR INITIAL DEGREE OF OPENING
S72
No
PREDETERMINED PERIOD≥U
Yes
S73
No
ΔP>P2 ?
S75
No
P1≦ΔP≦P2 ?
S77
ΔP<P1
Yes
Yes
S74
DECREASE BY PREDETERMINED DEGREE OF OPENING
S76
MAINTAIN CURRENT DEGREE OF OPENING
S78
INCREASE BY PREDETERMINED DEGREE OF OPENING

FIG. 7

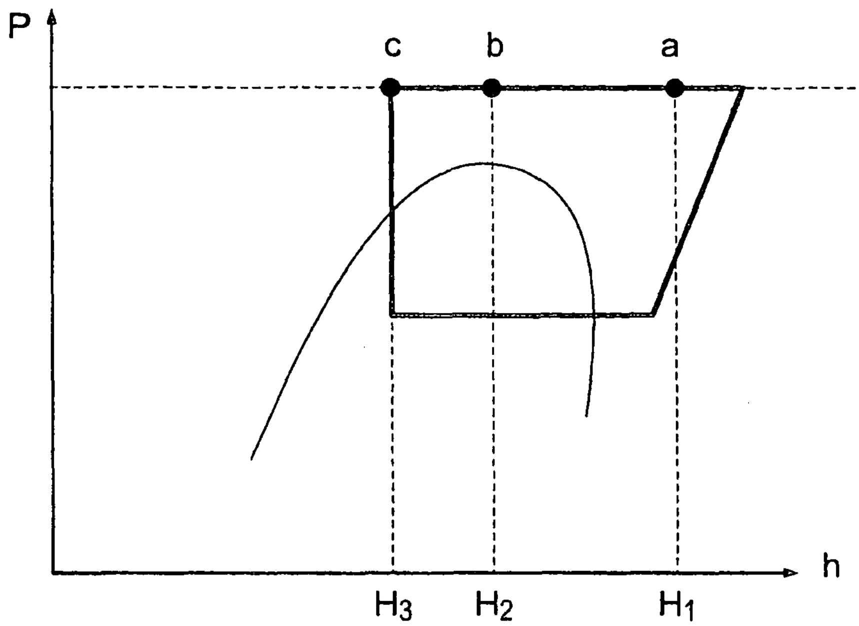

FIG. 8

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2009/056655

A. CLASSIFICATION OF SUBJECT MATTER
F25B1/00(2006.01)i, *F25B29/00*(2006.01)i

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

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
F25B1/00, F25B29/00

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

Jitsuyo Shinan Koho	1922-1996	Jitsuyo Shinan Toroku Koho	1996-2009
Kokai Jitsuyo Shinan Koho	1971-2009	Toroku Jitsuyo Shinan Koho	1994-2009

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X Y	JP 2000-346488 A (Mitsubishi Electric Corp.), 15 December, 2000 (15.12.00), Full text; Figs. 1 to 11 (Family: none)	1,2,4-6 3
Y	JP 2004-245480 A (Calsonic Kansei Corp.), 02 September, 2004 (02.09.04), Par. Nos. [0002], [0003] (Family: none)	3

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

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

Date of the actual completion of the international search	Date of mailing of the international search report
15 April, 2009 (15.04.09)	28 April, 2009 (28.04.09)
Name and mailing address of the ISA/ Japanese Patent Office Facsimile No.	Authorized officer Telephone No.

Form PCT/ISA/210 (second sheet) (April 2007)