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(54) **Air conditioner**

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(73) Proprietor: **DAIKIN INDUSTRIES, LTD.**
Osaka-shi, Osaka 530-8323 (JP)

(72) Inventors:
• **Kataoka, Hidehiko**
Kusatsu-shi
Shiga 525-0044 (JP)

- **Nishiura, Yoshihiro**
Kusatsu-shi
Shiga 525-0044 (JP)
- **Sakamoto, Shinichi**
Kusatsu-shi
Shiga 525-0044 (JP)

(74) Representative: **HOFFMANN EITLE**
Patent- und Rechtsanwälte
Arabellastraße 4
81925 München (DE)

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Description

TECHNICAL FIELD

5 **[0001]** The present invention relates to an air conditioner, and particularly to an air conditioner provided with a receiver for collecting surplus refrigerant.

BACKGROUND ART

10 **[0002]** A refrigerant circuit of an air conditioner is formed of a refrigerant pipe structure connecting an accumulator, a compressor, a four-way valve and an outdoor heat exchanger, which are arranged in an outdoor unit, as well as an indoor heat exchanger arranged in an indoor unit, and provides a circulation path of refrigerant.

15 **[0003]** When the refrigerant circuit of the air conditioner operate for cooling, the four-way valve controls a refrigerant circulating direction such that the outdoor heat exchanger functions as a condenser, and the indoor heat exchanger functions as an evaporator. In a heating operation, the four-way valve controls the refrigerant circulating direction such that the outdoor heat exchanger functions as an evaporator, and the indoor heat exchanger functions as a condenser.

20 **[0004]** In the above air conditioner, it is desired in view of ease of installation that the refrigerant circuit on the outdoor unit side is filled with an appropriate amount of refrigerant containing an additional amount of refrigerant, which is required for a connection pipe of a maximum length. However, the appropriate amount of refrigerant, which is required during an operation, significantly changes depending on an operation mode, a capacity of the indoor unit and a length of the connection pipe used for actual installation. Therefore, surplus refrigerant may be present in the refrigerant circuit, and this may cause abnormal rising of a pressure when it remains in the condenser, and may lower the operation efficiency.

25 **[0005]** For deal with such surplus refrigerant, it has been proposed to arrange a receiver between the outdoor heat exchanger and a liquid shut-off valve for collecting the surplus refrigerant.

30 **[0006]** In a conventional receiver circuit, a receiver is arranged in a pipe portion on a liquid pipe side between the outdoor heat exchanger and the liquid shut-off valve, and is located in series thereto. Therefore, the amount of refrigerant in the receiver increases or decreases merely in accordance with a degree of opening of a motor-operated valve located downstream from the receiver. This motor-operated valve corresponds to each room motor-operated valve in a cooling operation, and corresponds to a main pressure reducing valve, and the degree of opening thereof is controlled based on a degree of super-heating. Therefore, it is difficult to control or adjust precisely the amount of liquid refrigerant in the receiver, and particularly, it is impossible to control the amount of liquid refrigerant in the receiver when a special operation such as an operation with a low outside-air temperature or with an excessive load.

35 **[0007]** For example, when a cooling operation is performed when the outside-air temperature is extremely low, the outdoor heat exchanger has an excessively large capacity so that the pressure on the high-pressure side lowers. Therefore, a difference between the high and low pressures of the compressor decreases, which lowers reliability of the compressor. If the surplus refrigerant, which is collected in the receiver, can be transferred to the outdoor heat exchanger, a portion of the outdoor heat exchanger can be filled with the surplus refrigerant. Thereby, the capacity of the outdoor heat exchanger lowers, and the pressure on the high-pressure side rises so that a sufficient pressure difference can be ensured.

40 **[0008]** Further, JP 60-114669 A discloses an air-conditioner having the features of the preamble of claim 1. JP 2000-146322 A discloses an air-conditioner using a temperature sensor and a pressure sensor downstream of the evaporator to calculate the super-cooling degree based on the measured values. Based on the calculated super-cooling degree the circulating flow rate of the refrigerant is changed.

45 DISCLOSURE OF THE INVENTION

50 **[0009]** An object of the invention is to control surplus refrigerant in accordance with operation situations such as a cooling operation at a lower outside-air temperature and an operation with an excessive load.

55 **[0010]** An air conditioner according to the invention is defined in claim 1. Embodiments are named in the dependent claims.

60 **[0011]** The presence/absence of the surplus refrigerant is determined in the heating operation by comparing a high-pressure corresponding saturation temperature with a target value of the high-pressure corresponding saturation temperature.

65 **[0012]** In the heating operation, the target temperature of the high-pressure corresponding saturation temperature may be determined from an amount of circulated refrigerant and a room temperature, or may be determined from an operation frequency of the compressor and the room temperature.

70 **[0013]** The target value of the high-pressure corresponding saturation temperature may be corrected based on a deviation in super-heat control or target discharge pipe temperature control.

[0014] The target value of the high-pressure corresponding saturation temperature may be equal to or lower than a sum of a high-pressure saturation temperature and a predetermined value, and may be equal to or higher than a sum of a room temperature and a predetermined value.

[0015] Further, the air conditioner may be configured such that the gas pipe motor-operated valve operates in the heating operation to open when the high-pressure corresponding saturation temperature is lower than the target temperature, and to close when the high-pressure corresponding saturation temperature is higher than the target temperature.

BRIEF DESCRIPTION OF THE DRAWINGS

[0016]

Fig. 1 schematically shows a refrigerant circuit of an air conditioner employing an embodiment of the invention;

Figs. 2 to 5 are control flowcharts depicting control of surplus refrigerant in cooling operation;

Figs. 6 to 9 are control flowcharts depicting control of surplus refrigerant in cooling operation;

Figs. 10 to 13 are control flowcharts depicting control of surplus refrigerant in heating operation;

Figs. 14 to 17 are control flowcharts depicting control of surplus refrigerant according to the present invention;

Fig. 18 is a control block diagram;

Fig. 19 is a control block diagram of a compressor drive circuit;

Fig. 20 is a flowchart showing a manner of estimating a high-pressure corresponding saturation temperature; and

Fig. 21 shows a table for calculating a saturation temperature.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

(Schematic Structure)

[0017] An air conditioner employing a first embodiment of the invention has a refrigerant circuit shown in Fig. 1.

[0018] An outdoor unit 100 includes an outdoor refrigerant circuit provided with a compressor 101, a four-way valve 102, an outdoor heat exchanger 103 and an accumulator 105. A discharge-side pressure protection switch 108 is arranged on the discharge side of the compressor 101 for detecting abnormal rising of the discharge pressure. An intake-side pressure sensor 110 is arranged on the intake side of the compressor 101 for detecting an intake pressure.

[0019] An oil separator 107 is arranged on the discharge side of the compressor 101 for separating and returning lubricating oil contained in the refrigerant to the accumulator 105. The oil separator 107 is provided with a discharge pipe thermistor 109 for detecting a temperature on the discharge side of the compressor 101.

[0020] An oil return pipe 197 of the oil separator 107 is provided with a discharge bypass circuit 194, which is branched from the oil return pipe 197, and is connected to an inlet side of the accumulator 105. This discharge bypass circuit 194 is provided with a heat exchanger pipe portion 196 arranged within the accumulator 105 and a discharge-intake motor-operated valve 142 for capacity control. The oil return pipe 197 of the oil separator 107 is provided with a capillary 141, which is also connected to the intake side of the accumulator 105.

[0021] The outdoor unit 100 is also provided with an outside-air thermistor 111 for detecting an outside-air temperature, an outdoor heat exchange thermistor 112 for detecting an outlet temperature of the outdoor heat exchanger 103, and an heat exchange intermediate thermistor 113 for detecting an heat exchanger intermediate temperature. The outdoor unit 100 is further provided with a fan 106 for performing heat exchange between the outside air taken into the unit 100 and the refrigerant flowing within the outdoor heat exchanger 103, and a fan motor 104 for driving the fan 106.

[0022] The refrigerant pipe extending from the outdoor unit 100 toward the indoor unit is provided with a liquid pipe connection port 114 connected to the outdoor heat exchanger 103, and a gas pipe connection port 115 connected to the outdoor heat exchanger 103 via the four-way valve 102. The ports 114 and 115 are internally provided with a liquid pipe shut-off valve 116 and a gas pipe shut-off valve 117, respectively.

[0023] The outdoor unit 100 includes a receiver 121 for temporarily storing surplus refrigerant liquid, which flows from the outdoor heat exchanger 103 serving as a condenser in the cooling operation. The receiver 121 is provided with a liquid pipe connection pipe 122 and a gas pipe connection pipe 123. The liquid pipe connection pipe 122 is connected

to a liquid-side pipe portion 131 between the outdoor heat exchanger 103 and the liquid pipe shut-off valve 116. The gas pipe connection pipe 123 is connected to a gas-side pipe portion 132 between the four-way valve 102 and the gas pipe shut-off valve 117.

[0024] The liquid pipe connection pipe 122 of the receiver 121 is provided with a liquid pipe motor-operated valve (EVL) 128 having a pressure reducing function and a refrigerant shut-off function. The gas pipe connection pipe 123 is provided with a gas pipe motor-operated valve (EVG) 129.

[0025] An auxiliary heat exchanger 133 is arranged between the gas pipe motor-operated valve 129 and a connection to the gas-side pipe portion 132. A sub-cool heat exchanger 134 is arranged in a liquid-side outlet of the outdoor heat exchanger 103.

[0026] A gas vent capillary 130 is directed toward a portion of the gas-side pipe portion 132 between the four-way valve 102 and the gas pipe shut-off valve 117 for collecting a refrigerant gas from the receiver 121.

[0027] A plurality of branch units 300A, 300B, ... are connected to each of the liquid pipe connection port 114 and the gas pipe connection port 115 of the outdoor unit 100. Since these branch units 300A, 300B, ... have the same structures, the description will now be given on only the branch unit 300A, and the others will not be described.

[0028] The branch unit 300A includes an outdoor side liquid pipe connection port 301 connected to the liquid pipe connection port 114 of the outdoor unit 100 and an indoor side gas pipe connection port 303 connected to the gas pipe connection port 115 of the outdoor unit 100. The branch unit 300A has liquid-side branches divided within the outdoor side connection port 301, and ends of these branches form a plurality of indoor side liquid pipe connection ports 302 equal in number to the indoor units connected thereto. The branch unit 300A also has gas-side branches divided within the outdoor side gas pipe connection port 303, and ends of these branches form indoor side gas pipe connection ports 304 equal in number to the indoor units connected thereto. In the following description, it is assumed that the indoor units connected to the ports are three in number, and indoor side liquid pipe connection ports 302A, 302B and 302C as well as indoor side gas pipe connection ports 304A, 304B and 304C are employed. A motor-operated valve 308 for bypass is arranged between the outdoor side liquid pipe connection port 301 and the outdoor side gas pipe connection port 303.

[0029] The branches, which extend from the outdoor side liquid pipe connection port 301 within the branch unit 300A to the indoor side liquid pipe connection ports 302A - 302C, respectively, are provided with motor-operated valves 305A - 305C for reducing the pressures of the refrigerant flowing therethrough as well as liquid pipe thermistors 306A - 306C for detecting temperatures of the refrigerant flowing therethrough, respectively. The branches, which extend from the outdoor side gas pipe connection port 303 within the branch unit 300A to the indoor side gas pipe connection ports 304A - 304C, respectively, are provided with gas pipe thermistors 307A - 307C for detecting the temperatures of the refrigerant flowing therethrough, respectively.

[0030] Each of the branch units 300A, 300B, ... is connected to a plurality of outdoor units 200. It is assumed in the structure shown in Fig. 1 that three outdoor units can be connected to each of the branch units 300A, 300B, ... so that outdoor units 200A - 200C are connected to the branch unit 300A, and outdoor units 200D - 200F are connected to the branch unit 300B. Each of the indoor units 200A - 200F can be used in the indoor unit of either a multi-unit type or a pair-unit type. In the following description, the indoor unit 200A is of the pair-unit type.

[0031] The indoor unit 200A is provided with an indoor heat exchanger 201, which is connected to a refrigerant pipe extending toward the outdoor unit through a liquid pipe connection port 204 and a gas pipe connection port 205. The indoor unit 200A is provided with a room temperature thermistor 202 for detecting a room temperature and an indoor heat exchange thermistor 203 for detecting a temperature of the indoor heat exchanger 201.

[0032] If the indoor units connected to the branch units 300A and 300B is of the multi-unit type, the liquid-side pipe portion may be provided with a liquid pipe thermistor for detecting the temperature of the refrigerant flowing therethrough, in which case the liquid pipe thermistors in the branch units 300A and 300B may be eliminated.

(Surplus Refrigerant Control in Cooling Operation)

[0033] In the cooling operation, the surplus refrigerant is controlled in accordance with a liquid pipe temperature. This will be described below with reference to a flowchart of Fig. 2.

[0034] In a step S11, it is determined whether a surplus refrigerant control sampling time TSCSET has elapsed or not. When it is determined that an elapsed time counted by a timer has reached the surplus refrigerant control sampling time TSCSET, the processing moves to a step S12.

[0035] In the step S12, the target liquid pipe temperature is calculated.

[0036] The processing of calculating the target liquid pipe temperature will now be described with reference to a flowchart of Fig. 3.

[0037] In a step S21, a variable DOATD is calculated in accordance with the following formula using target liquid pipe temperature calculation coefficients KSCC1, KSCC2, KSCC3 and EDOSC, a target frequency FMK of the compressor 101, a discharge pipe temperature deviation EDO and others.

$$DOATD = KSCC1 \times FMK + KSCC2 - (EDO - EDOSC) \times KSCC3$$

5 [0038] In a step S22, it is determined whether the variable DQATD is larger than a lower limit DQMTMIN of the target liquid pipe temperature or not. When it is determined that the variable DOATD is not larger than the lower limit DOATDMIN of the target liquid pipe temperature, the processing moves to a step S23. In the step S23, the value of variable DOATD is set to the lower limit DOATDMIN of the target liquid pipe temperature.

10 [0039] In a step S24, it is determined whether the variable DOATD is equal to or smaller than ((upper limit DOATDMAX of target liquid pipe temperature) - (outside air temperature DOA)) or not. When it is determined that the variable DOATD is larger than ((upper limit DQATDMAX of target liquid pipe temperature) - (outside air temperature DOA)), the processing moves to a step S25. In the step S25, the value of variable DQATD is set to ((upper limit DQATDMAX of target liquid pipe temperature) - (outside air temperature DOA)).

15 [0040] In a step S26, a target liquid pipe temperature DELSET is calculated. In this embodiment, the following formula is employed for this calculation.

$$DELSET = DOA \text{ (outside air temperature)} + DOATD \text{ (variable)}$$

20 [0041] In a step S13, a liquid pipe temperature deviation ΔDEL is calculated from the target liquid pipe temperature DELSET and the outdoor heat exchanger outlet temperature DEL in accordance with the following formula.

$$25 \quad \Delta DEL = DELSET - DEL$$

[0042] In a step S14, processing is performed to calculate an amount or degree, by which the motor-operated valve is to be operated.

30 [0043] This processing of calculating the motor-operated valve operation amount is shown in a flowchart of Fig. 4.

[0044] In a step S31, a motor-operated valve operation amount POSC is calculated in accordance with the following formula using coefficients KOSCA1 and KOSCA, which are coefficients for calculating the motor-operated valve operation amount, the liquid pipe temperature deviation ΔDEL , the last liquid pipe temperature deviation $\Delta DELZ$ and others.

$$35 \quad POSC = KOSCA1 \times ((\Delta DEL - \Delta DELZ) + \Delta DEL / KOSCA)$$

40 [0045] In a step S15, the motor-operated valve is operated in accordance with the operation amount determined in the step S14.

[0046] Processing of operating the motor-operated valve is shown in a flowchart of Fig. 5.

[0047] It is assumed that a degree EVL of opening of the liquid pipe motor-operated valve 128 is defined by ((current degree EVL) + (motor-operated valve operation amount POSC)). At the same time, a degree EVG of opening of the gas pipe motor-operated valve 129 is controlled to attain ((current degree EVG) + (operation amount POSC) $\times$ KPOSC1).

45 [0048] The liquid pipe temperature during the cooling operation can be determined from a value detected by the outdoor heat exchange thermistor 112 arranged at the vicinity of the outlet of the outdoor heat exchanger 103. Based on the liquid pipe temperature, it is possible to control the amount of the surplus refrigerant in the receiver 121.

[0049] Therefore, even when the cooling operation is performed while an outside-air temperature is low, the amount of the surplus refrigerant is controlled to ensure a difference between high and low pressures in the compressor 101.

50 (Control of Surplus Refrigerant in Cooling Operation)

[0050] Referring to a flowchart of Fig. 6, description will now be given on control of surplus refrigerant performed with a high-pressure corresponding saturation temperature in the cooling operation. It is assumed that the same refrigerant circuit as that in the first embodiment is used for controlling the surplus refrigerant in the cooling operation.

55 [0051] In a step S51, it is determined whether a surplus refrigerant control sampling time TSCSET has elapsed or not. When it is determined that an elapsed time counted by the timer has reached the surplus refrigerant control sampling time TSCSET, the processing moves to a step S52.

[0052] In the step S52, the target high-pressure corresponding saturation temperature is calculated.

[0053] The processing of calculating the target high-pressure corresponding saturation temperature will now be described with reference to a flowchart of Fig. 7.

[0054] In a step S61, the variable DOATD is calculated in accordance with the following formula using the target high-pressure corresponding saturation temperature calculation coefficients KSCC1, KSCC2, KSCC3 and EDOSC, the target frequency EMK of the compressor 101, the discharge pipe temperature deviation EDO and others.

$$DOATD = KSCC1 \times EMK + KSCC2 - (EDO - EDOSC) \times KSCC3$$

[0055] In a step S62, it is determined whether the variable DOATD is larger than a lower limit DOATDMIN of the target high-pressure corresponding saturation temperature or not. When it is determined that the variable DOATD is not larger than the lower limit DOATDMIN of the target high-pressure corresponding saturation temperature, the processing moves to a step S63. In the step S63, the value of variable DOATD is set to the lower limit DOATDMIN of the target high-pressure corresponding saturation temperature.

[0056] In a step S64, it is determined whether the variable DOATD is equal to or smaller than the upper limit DOATDMAX of the target high-pressure corresponding saturation temperature or not. When it is determined that the variable DOATD is larger than the upper limit DOATDMAX of the target high-pressure corresponding saturation temperature, the processing moves to a step S65. In the step S65, the value of variable DOATD is set to the upper limit DOATDMAX of the target high-pressure corresponding saturation temperature.

[0057] In a step S66, the target high-pressure corresponding saturation temperature TDSSET is calculated. In this embodiment, the following formula is employed for this calculation.

$$TDSSET = DOA \text{ (outside air temperature)} + DOATD \text{ (variable)}$$

[0058] In a step S53, a high-pressure corresponding saturation temperature deviation ΔTDS is calculated from the high-pressure corresponding saturation temperature TDSSET and the target high-pressure corresponding saturation temperature TDS in accordance with the following formula.

$$\Delta TDS = TDSSET - TDS$$

[0059] In a step S54, processing is performed to calculate an amount or degree, by which the motor-operated valve is to be operated.

[0060] This processing of calculating the motor-operated valve operation amount is shown in a flowchart of Fig. 8.

[0061] In a step S71, the motor-operated valve operation amount POSC is calculated in accordance with the following formula using the coefficients KOSCA1 and KOSCA, which are coefficients for calculating the motor-operated valve operation amount, the high-pressure corresponding saturation temperature deviation TDS, the last high-pressure corresponding saturation temperature deviation TDSZ and others.

$$POSC = KOSCA1 \times ((\Delta TDS - \Delta TDSZ + \Delta TDSZ / KOSCA))$$

[0062] In a step S55, the motor-operated valve is operated in accordance with the operation amount determined in the step S54.

[0063] Processing of operating the motor-operated valve is shown in a flowchart of Fig. 9.

[0064] It is assumed that the degree EVL of opening of the liquid pipe motor-operated valve 128 is defined by ((current degree EVL) + (motor-operated valve operation amount POSC)). At the same time, the degree EVG of opening of the gas pipe motor-operated valve 129 is controlled to attain ((current degree EVG) + (operation amount POSC) $\times$ KPOSC1).

[0065] Referring to a flowchart of Fig. 10, description will now be given on control of surplus refrigerant performed with a liquid pipe temperature in the heating operation.

[0066] In a step S91, it is determined whether a surplus refrigerant control sampling time TSCSET has elapsed or not. When it is determined that an elapsed time counted by the timer has reached the surplus refrigerant control sampling

time TSCSET, the processing moves to a step S92.

[0067] In the step S92, the target liquid pipe temperature is calculated.

[0068] The processing of calculating the target liquid pipe temperature will now be described with reference to a flowchart of Fig. 11.

[0069] In a step S101, the variable DOATD is calculated in accordance with the following formula using target liquid pipe temperature calculation coefficients KSCC1 and KSCC2, the target frequency EMK of the compressor 101 and others.

$$DOATD = KSCC1 \times FMK + KSCC2$$

[0070] In a step S102, it is determined whether the variable DQATD is larger than the lower limit DOATIMIN of the target liquid pipe temperature or not. When it is determined that the variable DOATD is not larger than the lower limit DQATDMIN of the target liquid pipe temperature, the processing moves to a step S103. In the step S103, the value of variable DOATD is set to the lower limit DQATDMIN of the target liquid pipe temperature.

[0071] In a step S104, it is determined whether the variable DOATD is equal to or smaller than ((upper limit DQATDMAX of target liquid pipe temperature) - room temperature DA) or not. When it is determined that the variable DOATD is larger than ((upper limit DQATDMAX of target liquid pipe temperature) - room temperature DA), the processing moves to a step S105. In the step S105, the value of variable DOATD is set to ((upper limit DQATDMAX of target liquid pipe temperature) - room temperature DA).

[0072] In a step S106, a target liquid pipe temperature DLSET is calculated. In this embodiment, the following formula is employed for this calculation.

$$DLSET = DA \text{ (room temperature)} + DOATD \text{ (variable)}$$

[0073] In a step S93, a liquid pipe temperature deviation ΔDL is calculated. For the indoor heat exchanger 201 having the lowest liquid pipe temperature in the indoor unit 200, which is operating, the liquid pipe temperature deviation ΔDL is calculated from a representative value DL of the liquid pipe temperature and the target liquid pipe temperature DLSET in accordance with the following formula.

$$\Delta DL = DLSET - DL$$

[0074] In a step S94, processing is performed to calculate an amount or degree, by which the motor-operated valve is to be operated.

[0075] This processing of calculating the motor-operated valve operation amount is shown in a flowchart of Fig. 12.

[0076] In a step S111, the motor-operated valve operation amount POSC is calculated in accordance with the following formula using the coefficients KOSCA1 and KOSCA, which are coefficients for calculating the motor-operated valve operation amount, the liquid pipe temperature deviation ΔDL , the last liquid pipe temperature deviation ΔDLZ and others.

$$POSC = KOSCA1 \times ((\Delta DL - \Delta DLZ) + \Delta DL/KOSCA)$$

[0077] In a step S95, the motor-operated valve is operated in accordance with the operation amount determined in the step S94.

[0078] Processing of operating the motor-operated valve is shown in a flowchart of Fig. 13.

[0079] It is assumed that the degree EVG of opening of the gas pipe motor-operated valve 129 is defined by ((current degree EVG) + (motor-operated valve operation amount POSC)). Likewise, the degree EVL of opening of the liquid pipe motor-operated valve 128 is controlled to attain ((current degree EVL) + (operation amount POSC) $\times$ KPOSC1).

[0080] In the heating operation, the lowest liquid pipe temperature in the indoor unit 200, which is being operating, is used as the representative value of the liquid pipe temperature, and the amount of the surplus refrigerant in the receiver 121 can be controlled with this representative value of the liquid pipe temperature.

[0081] Referring to a flowchart of Fig. 14, description will now be given on control of surplus refrigerant performed

with a high-pressure corresponding saturation temperature in the heating operation.

[0082] In a step S131, it is determined whether a surplus refrigerant control sampling time TSCSET has elapsed or not. When it is determined that an elapsed time counted by the timer has reached the surplus refrigerant control sampling time TSCSET, the processing moves to a step S132.

[0083] In the step S132, the target high-pressure corresponding saturation temperature is calculated.

[0084] The processing of calculating the target high-pressure corresponding saturation temperature will now be described with reference to a flowchart of Fig. 15.

[0085] In a step S141, the variable DQATD is calculated in accordance with the following formula using the target high-pressure corresponding saturation temperature calculation coefficients KSCC1 and KSCC2, the target frequency FMK of the compressor 101 and others.

$$DQATD = KSCC1 \times FMK + KSCC2$$

[0086] In a step S142, it is determined whether the variable DQATD is larger than the lower limit DQATIMIN of the target high-pressure corresponding saturation temperature or not. When it is determined that the variable DQATD is not larger than the lower limit DQATIMIN of the target high-pressure corresponding saturation temperature, the processing moves to a step S143. In the step S143, the value of variable DQATD is set to the lower limit DQATIMIN of the target high-pressure corresponding saturation temperature.

[0087] In a step S144, it is determined whether the variable DQATD is equal to or smaller than the upper limit DOATDMAX of the target high-pressure corresponding saturation temperature or not. When it is determined that the variable DQATD is larger than the upper limit DQATDMAX of the target high-pressure corresponding saturation temperature, the processing moves to a step S145. In the step S145, the value of variable DOATD is set to the upper limit DOATIMAX of the target high-pressure corresponding saturation temperature.

[0088] In a step S146, a target high-pressure corresponding saturation temperature TDSSET is calculated. In this embodiment, the following formula is employed for this calculation.

$$TDSSET = DA \text{ (room temperature)} + DQATD \text{ (variable)}$$

[0089] In a step S133, the high-pressure corresponding saturation temperature deviation ΔTDS is calculated. The high-pressure corresponding saturation temperature deviation ΔTDS is calculated from the high-pressure corresponding saturation temperature TDS and the target high-pressure corresponding saturation temperature TDSSET in accordance with the following formula.

$$\Delta TDS = TDSSET - TDS$$

[0090] In a step S134, processing is performed to calculate an amount or degree, by which the motor-operated valve is to be operated.

[0091] This processing of calculating the motor-operated valve operation amount is shown in a flowchart of Fig. 16.

[0092] In a step S151, the motor-operated valve operation amount POSC is calculated in accordance with the following formula using the coefficients KOSCAL and KO6CA, which are coefficients for calculating the motor-operated valve operation amount, the high-pressure corresponding saturation temperature deviation ΔTDS , the last high-pressure corresponding saturation temperature deviation $\Delta TDSZ$ and others.

$$POSC = KOSCAL \times ((\Delta TDS - \Delta TDSZ) + \Delta TDS / KOSCA)$$

[0093] In a step S135, the motor-operated valve is operated in accordance with the operation amount determined in the step S134.

[0094] Processing of operating the motor-operated valve is shown in a flowchart of Fig. 17.

[0095] It is assumed that the degree EVG of opening of the gas pipe motor-operated valve 129 is defined by ((current degree EVG) + (motor-operated valve operation amount POSC)). Likewise, the degree EVL of opening of the liquid pipe

motor-operated valve 128 is controlled to attain $((\text{current degree EVL}) + (\text{operation amount POSC}) \times \text{KPOSC1})$.

(Estimation of High-Pressure Corresponding Saturation Temperature)

- 5 **[0096]** Description will now be given on a manner of estimating the high-pressure corresponding saturation temperature in the second embodiment already described. A control block for this estimation is shown in Fig. 18.
- [0097]** A control portion 501 is formed of a microprocessor including a CPU, a ROM, a RAM and others, and is connected to a ROM 502 storing an operation control program and various parameters as well as a RAM 503 for temporarily storing work variables.
- 10 **[0098]** The control portion 501 is also connected to various sensor elements including the intake-side pressure sensor 110, discharge pipe thermistor 109, outside air thermistor 111, outdoor heat exchange thermistor 112 and heat exchange intermediate thermistor 113, which are arranged within an outdoor unit 100, for receiving values detected by these sensor and thermistors. Further, the discharge-side pressure switch 108 is connected to the control portion 501.
- [0099]** An indoor-side communication interface 504 for transmitting various data between the indoor unit 200 and the branch unit 300 is connected to the control portion 501.
- 15 **[0100]** Further, the control portion 501 is connected to a compressor drive circuit 505 for controlling the operation frequency of the compressor 101 and a fan motor drive circuit 506 for controlling a frequency of the fan motor 104.
- [0101]** The control portion 501 is further connected to the liquid and gas pipe motor-operated valves 128 and 129 arranged on the opposite sides of the receiver 121, and the discharge-intake motor-operated valve 142 arranged on the discharge bypass circuit 194 of the compressor 101.
- 20 **[0102]** The compressor drive circuit 505 is provided with an active filter circuit, which will be described later, and a secondary voltage sensor 507 and a secondary current sensor 508 of the active filter circuit are connected to the control portion 501.
- [0103]** Fig. 19 is a block diagram showing control of the compressor drive circuit 505 in Fig. 18.
- 25 **[0104]** The compressor drive circuit 505 includes a rectifier circuit 512 connected to a commercial power supply 511, an active filter circuit 513 and an inverter circuit 514.
- [0105]** The rectifier circuit 512 is formed of a diode bridge including four diodes, and performs full-wave rectification on an AC power supplied from the commercial power supply 511.
- [0106]** The active filter circuit 513 includes a reactor 521, a diode 522, a capacitor 523, a switching element 524 and active filter drive means 525 for performing switching control of the switching element 524.
- 30 **[0107]** The active filter 513 includes a primary voltage sensor 526 for detecting a primary voltage, a primary current sensor 527 for detecting a primary current, a secondary voltage sensor 507 for detecting a secondary voltage and a secondary current sensor 508 for detecting a secondary current. The active filter drive means 525 performs the switching control on the switching element 524 so that the secondary voltage detected by the secondary voltage sensor 507 may become equal to a predetermined voltage. At the same time, the active filter drive means 525 controls the current value detected by the primary current sensor 527 so that it matches in phase with the primary voltage detected by the primary voltage sensor 526. This significantly increases a power factor, and thus increases the accuracy of calculating the power consumption based on the secondary voltage detected by the secondary voltage sensor 507 and the secondary current detected by the secondary current sensor 508.
- 35 **[0108]** The inverter circuit 514 produces a pulse signal of a constant voltage from an output signal of a predetermined voltage sent from the active filter circuit 513. The output frequency of the inverter circuit 514 in this operation is equal to the operation frequency of the compressor determined based on the current operation situations. Therefore, the compressor drive motor 531 is driven in accordance with the output frequency of the inverter circuit 514.
- [0109]** The power consumption of the compressor 101 is calculated from the secondary voltage value and the secondary current value of the active filter 513 of the compressor drive circuit 505, and a value of the high-pressure corresponding temperature is estimated based on the power consumption thus calculated. A manner for this estimation will now be described with reference to a flowchart of Fig. 20.
- 40 **[0110]** In a step S171, an input voltage VIN and an input current IIN applied to the inverter circuit 514 are detected. The input voltage VIN and the input current IIN applied to the inverter circuit 514 can be obtained from the secondary voltage sensor 507 and the secondary current sensor 508, which detect the secondary voltage and current of the active filter 513, respectively.
- 45 **[0111]** In a step S172, power consumption INPUT of the compressor 101 is calculated based on the secondary voltage VIN and the secondary current IIN of the active filter 513. Since the switching element 524 is controlled so that the active filter drive means 525 of the active filter 513 may have an appropriate power factor, it can be considered that the power factor is equal to one. Therefore, the power consumption of the compressor can be determined by the following formula.
- 55

$$\text{INPUT} = \text{VIN} \times \text{IIN} \times 1 \text{ (power factor)}$$

[0112] In a step S173, processing is performed to determine an output frequency FOUT for driving the compressor 101 as well as an intake pressure value LP thereof. In this processing, the output frequency FOUT can be specifically determined from the output frequency of the inverter 514 driving the compressor drive motor 531. The intake pressure value LP can be specifically determined from the detected value of the intake pressure sensor 110.

[0113] In a step S174, a high pressure value is determined based on the power consumption INPUT, output frequency FOUT and intake pressure value LP. This determination is performed in accordance with the following formula, which uses high pressure estimation constants KHPLL, KHPFF, KHPII, KHPLF, KHPFI, KHPLI, KHPL, KHPF, KHPI and KHPC as well as a high pressure correction value HPHOSEI.

$$\begin{aligned} \text{HP} = & \text{KHPLL} \times \text{LP}^2 + \text{KHPFF} \times \text{FOUT}^2 + \text{KHPII} \times \text{INPUT}^2 + \text{KHPLF} \times \text{LP} \\ & \times \text{FOUT} + \text{KHPFI} \times \text{FOUT} \times \text{INPUT} + \text{KHPLI} \times \text{LP} \times \text{INPUT} + \text{KHPL} \times \text{LP} + \\ & \text{KHPF} \times \text{FOUT} + \text{KHPI} \times \text{INPUT} + \text{KHPC} + \text{HPHOSEI} \end{aligned}$$

[0114] In a step S157, the high-pressure corresponding saturation transmission TDS is calculated based on the high pressure value HP calculated in the step S174. For this, the following formula can be used.

$$\text{TDS} = \text{A} \times \text{HP} + \text{B}$$

where coefficients A and B for calculating the high-pressure corresponding saturation temperature are determined by the high pressure value HP in accordance with a table shown in Fig. 21.

Industrial Applicability

[0115] According to the invention, the amount of surplus refrigerant collected into the receiver can be controlled in accordance with operation situations, and the degree of super cooling or super heating within the outdoor heat exchanger can be controlled. Therefore, an appropriate amount of refrigerant can be used in the refrigerant circuit system for performing efficient driving.

Claims

1. An air conditioner comprising a refrigerant circuit formed of an outdoor unit refrigerant circuit including at least a compressor (101), a four-way valve (102) and an outdoor heat exchanger (103) arranged in an outdoor unit (100), and an indoor heat exchanger (201) arranged in an indoor unit (200) and connected to said outdoor unit refrigerant circuit by a liquid-side pipe and a gas-side pipe, wherein
a bypass circuit forming a bypass between said liquid-side and gas-side pipes (131, 132) is provided with a receiver (121) for collecting liquid refrigerant, and refrigerant on-off means (128, 129) arranged in a liquid pipe connection pipe (122) connecting said receiver (121) to said liquid-side pipe (131) and in a gas pipe connecting pipe (123) connecting said receiver (121) to said gas-side pipe (132),
said refrigerant on-off means (128, 129) is controlled based on determination of presence/absence of surplus refrigerant performed by determining a current heat exchanging performance of said outdoor heat exchanger (103), and thereby an amount of the refrigerant in said receiver (121) is controlled, **characterized in that** the presence/absence of the surplus refrigerant is determined in the heating operation by comparing a high-pressure corresponding saturation temperature with a target value of the high-pressure corresponding saturation temperature.
2. The air conditioner according to claim 1, wherein the target temperature of the high-pressure corresponding saturation temperature is determined from an amount of circulated refrigerant and a room temperature in the heating operation.

3. The air conditioner according to claim 1, wherein the target temperature of the high-pressure corresponding saturation temperature is determined from an operation frequency of said compressor (101) and the room temperature in the room temperature.
- 5 4. The air conditioner according to claim 2 or 3, wherein the target value of said high-pressure corresponding saturation temperature is corrected based on a deviation in super-heat control or target discharge pipe temperature control.
- 10 5. The air conditioner according to claim 4, wherein the target value of said high-pressure corresponding saturation temperature is equal to or lower than a sum of a high-pressure saturation temperature and a predetermined value.
- 15 6. The air conditioner according to claim 4 or 5, wherein the target value of said high-pressure corresponding saturation temperature is equal to or higher than a sum of a room temperature and a predetermined value.
- 20 7. The air conditioner according to any one of the preceding claims, wherein said gas pipe motor-operated valve (129) operates in the heating operation to open when said high-pressure corresponding saturation temperature is lower than the target temperature, and to close when said high-pressure corresponding saturation temperature is higher than the target temperature.

Patentansprüche

- 25 1. Klimaanlage aufweisend: einen K hlmittelkreis, der aus einem Au eneinheit-K hlmittelkreis mit zumindest einem Kompressor (101), einem Vierwegeventil (102) und einem in einer Au eneinheit (100) angeordneten Au enw rmetauscher (103) ausgebildet ist, und einen in einer Inneneinheit (200) angeordneten und mit dem Au eneinheit-K hlmittelkreis mittels einem fl ssigkeitsseitigen Rohr und einem gasseitigen Rohr verbundenen Innenw rmetauscher (201), bei der
30 ein einen Nebenweg zwischen dem fl ssigkeitsseitigen und gasseitigen Rohr (131, 132) ausbildender Nebenwegkreis mit einem Empf nger (121) zum Sammeln eines fl ssigen K hlmittels und mit einem in einem den Empf nger (121) mit dem fl ssigkeitsseitigen Rohr (131) verbindenden Fl ssigkeitsrohrverbindungsrohr (122) und in einem den Empf nger (121) mit dem gasseitigen Rohr (132) verbindenden Gasrohrverbindungsrohr (123) angeordneten K hlmittel-An-Aus-Mittel (128, 129) versehen ist,
35 wobei das K hlmittel-An-Aus-Mittel (128, 129) basierend auf einer Bestimmung einer Anwesenheit/ Abwesenheit eines  bersch ssigen K hlmittels gesteuert wird, die durchgef hrt wird, indem eine gegenw rtige W rmetausch-Leistungsf higkeit des Au enw rmetauschers (103) bestimmt wird, und **dadurch** eine Gr  e des K hlmittels im Empf nger (121) gesteuert wird,
40 **dadurch gekennzeichnet, dass** die Anwesenheit/Abwesenheit des  bersch ssigen K hlmittels im W rmebetrieb durch Vergleichen einer Hochdruck-entsprechenden S ttigungstemperatur mit einem Zielwert der Hochdruck-entsprechenden S ttigungstemperatur bestimmt wird.
- 45 2. Klimaanlage gem   Anspruch 1, bei welcher die Zieltemperatur der Hochdruck-entsprechenden S ttigungstemperatur aus einer Gr  e des zirkulierten K hlmittels und einer Raumtemperatur im W rmebetrieb bestimmt wird.
- 50 3. Klimaanlage gem   Anspruch 1, bei welcher die Zieltemperatur der Hochdruck-entsprechenden S ttigungstemperatur aus einer Betriebsfrequenz des Kompressors (101) und der Raumtemperatur bei der Raumtemperatur bestimmt wird.
- 55 4. Klimaanlage gem   Anspruch 2 oder 3, bei welcher der Zielwert der Hochdruck-entsprechenden S ttigungstemperatur basierend auf einer Abweichung bei einer  berhitzungssteuerung oder einer Zielaussto rohrtemperatursteuerung korrigiert wird.
5. Klimaanlage gem   Anspruch 4, bei welcher der Zielwert der Hochdruck-entsprechenden S ttigungstemperatur gleich oder geringer ist als eine Summe einer Hochdruck-entsprechenden S ttigungstemperatur und einem vorbestimmten Wert.

6. Klimaanlage gemäß Anspruch 4 oder 5, bei welcher der Zielwert der Hochdruck-entsprechenden Sättigungstemperatur gleich oder höher ist als eine Summe einer Raumtemperatur und einem vorbestimmten Wert.

7. Klimaanlage gemäß einem der vorangegangenen Ansprüche, bei welcher das motorbetriebene Gasrohrventil (129) im Wärmebetrieb arbeitet, um sich zu öffnen, wenn die Hochdruck-entsprechende Sättigungstemperatur geringer ist als die Zieltemperatur, und um sich zu schließen, wenn die Hochdruck-entsprechende Sättigungstemperatur höher ist als die Zieltemperatur.

Revendications

1. Climatiseur comprenant un circuit réfrigérant formé d'un circuit réfrigérant d'une unité externe comprenant au moins un compresseur (101), une soupape à quatre voies (102) et un échangeur thermique externe (103) que l'on agence dans une unité externe (100), et un échangeur thermique interne (201) agencé dans une unité interne (200) et relié audit circuit réfrigérant de l'unité externe par un tuyau côté liquide et un tuyau côté gaz, dans lequel un circuit de dérivation formant une dérivation entre lesdits tuyaux côté liquide et côté gaz (131, 132) est doté d'un réservoir (121) pour collecter le réfrigérant liquide, et un moyen de marche/d'interruption du réfrigérant (128, 129) agencé dans un tuyau de liaison (122) du tuyau côté liquide reliant ledit réservoir (121) audit tuyau côté liquide (131) et dans un tuyau de liaison (123) du tuyau côté gaz reliant ledit réservoir (121) audit tuyau côté gaz (132), ledit moyen de marche/d'interruption du réfrigérant (128, 129) est commandé sur la base d'une détermination de présence/d'absence du surplus de réfrigérant exécuté en déterminant une performance actuelle d'échange thermique dudit échangeur thermique externe (103), et ainsi une quantité du réfrigérant dans ledit réservoir (121) est régulée, **caractérisé en ce que**

la présence/absence du surplus de réfrigérant est déterminée dans l'opération de chauffage en comparant une température de saturation correspondant à une haute pression à une valeur cible de la température de saturation correspondant à une haute pression.

2. Climatiseur selon la revendication 1, dans lequel la température cible de la température de saturation correspondant à une haute pression est déterminée à partir d'une quantité du réfrigérant qui circule et d'une température ambiante dans l'opération de chauffage.

3. Climatiseur selon la revendication 1, dans lequel la température cible de ladite température de saturation correspondant à une haute pression est déterminée à partir d'une fréquence de fonctionnement dudit compresseur (101) et de la température ambiante dans la température ambiante.

4. Climatiseur selon la revendication 2 ou 3, dans lequel la valeur cible de ladite température de saturation correspondant à une haute pression est corrigée sur la base d'une déviation dans une commande de surchauffe ou une commande de température cible du tuyau de décharge.

5. Climatiseur selon la revendication 4, dans lequel la valeur cible de ladite température de saturation correspondant à une haute pression est inférieure ou égale à la somme d'une température de saturation à haute pression et d'une valeur prédéterminée.

6. Climatiseur selon la revendication 4 ou 5, dans lequel la valeur cible de ladite température de saturation correspondant à une haute pression est supérieure ou égale à la somme d'une température ambiante et d'une valeur prédéterminée.

7. Climatiseur selon l'une quelconque des revendications précédentes, dans lequel ladite soupape (129) actionnée par le moteur du tuyau côté gaz fonctionne dans l'opération de chauffage pour s'ouvrir lorsque ladite température de saturation correspondant à une haute pression est inférieure à la température cible, et pour se fermer lorsque ladite température de saturation correspondant à une haute pression est supérieure à la température cible.

Fig. 1

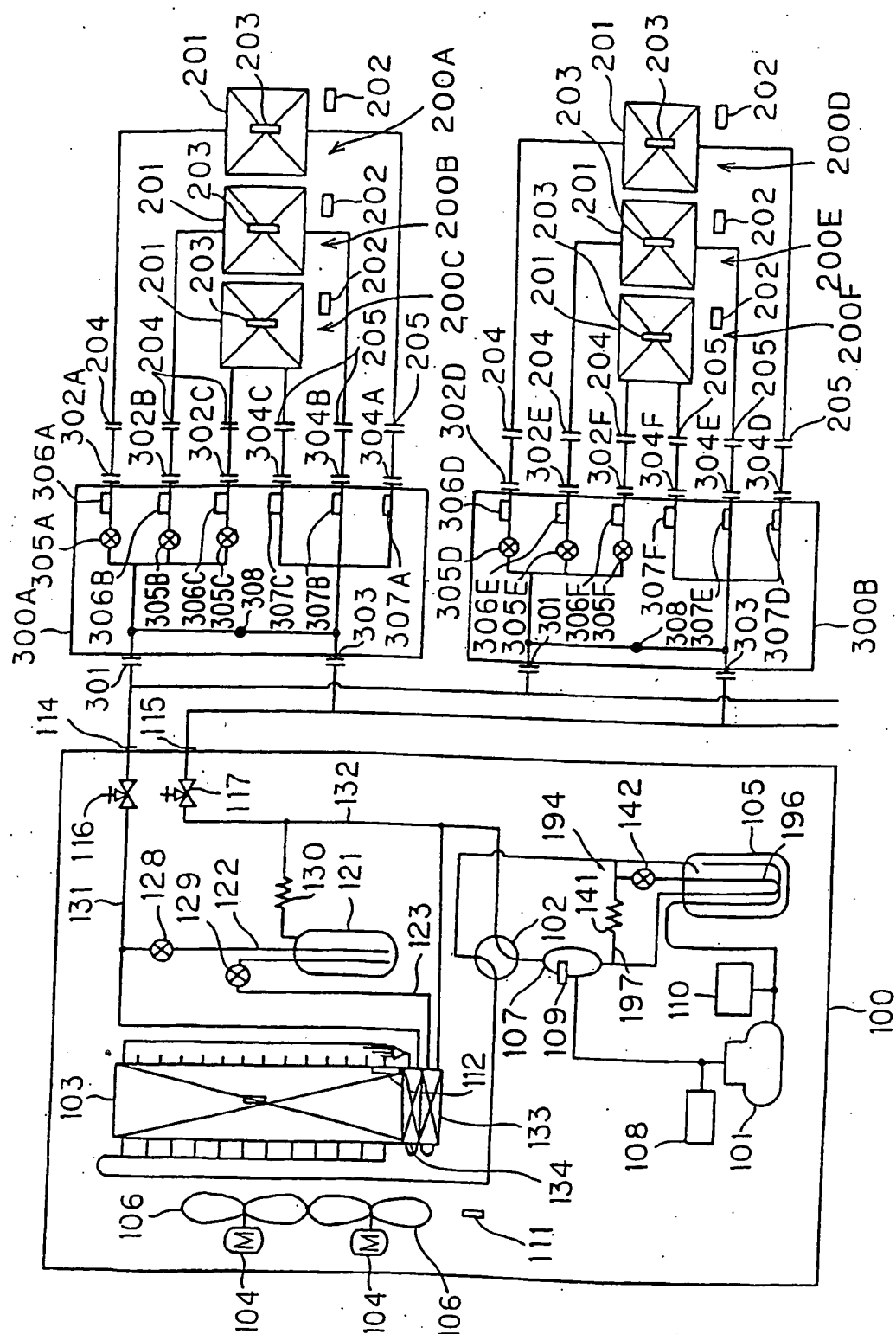


Fig. 2

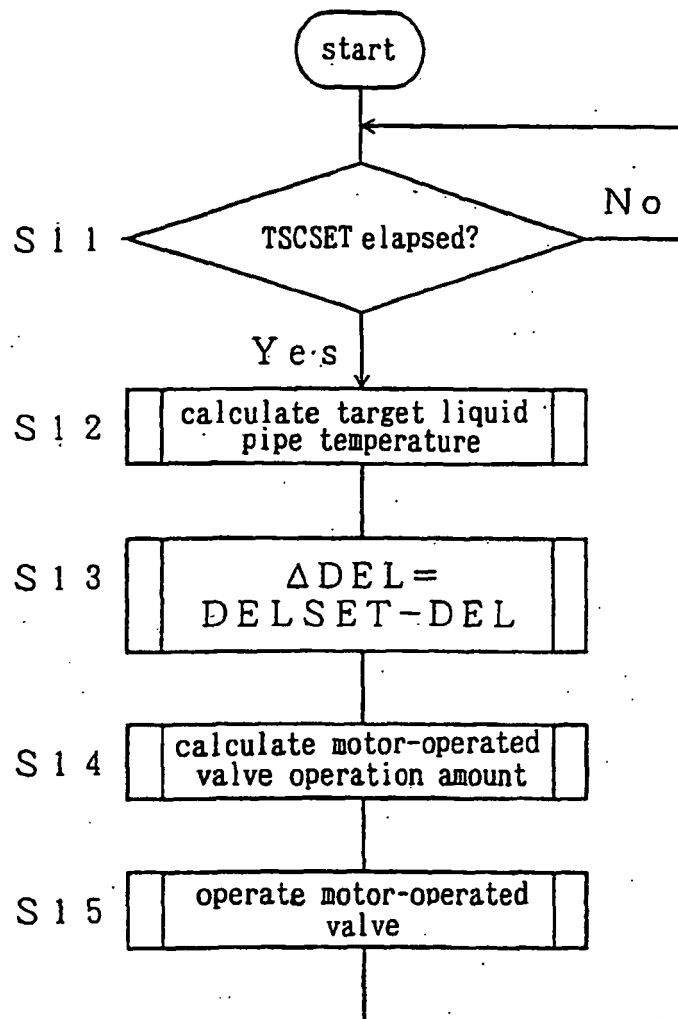


Fig. 3

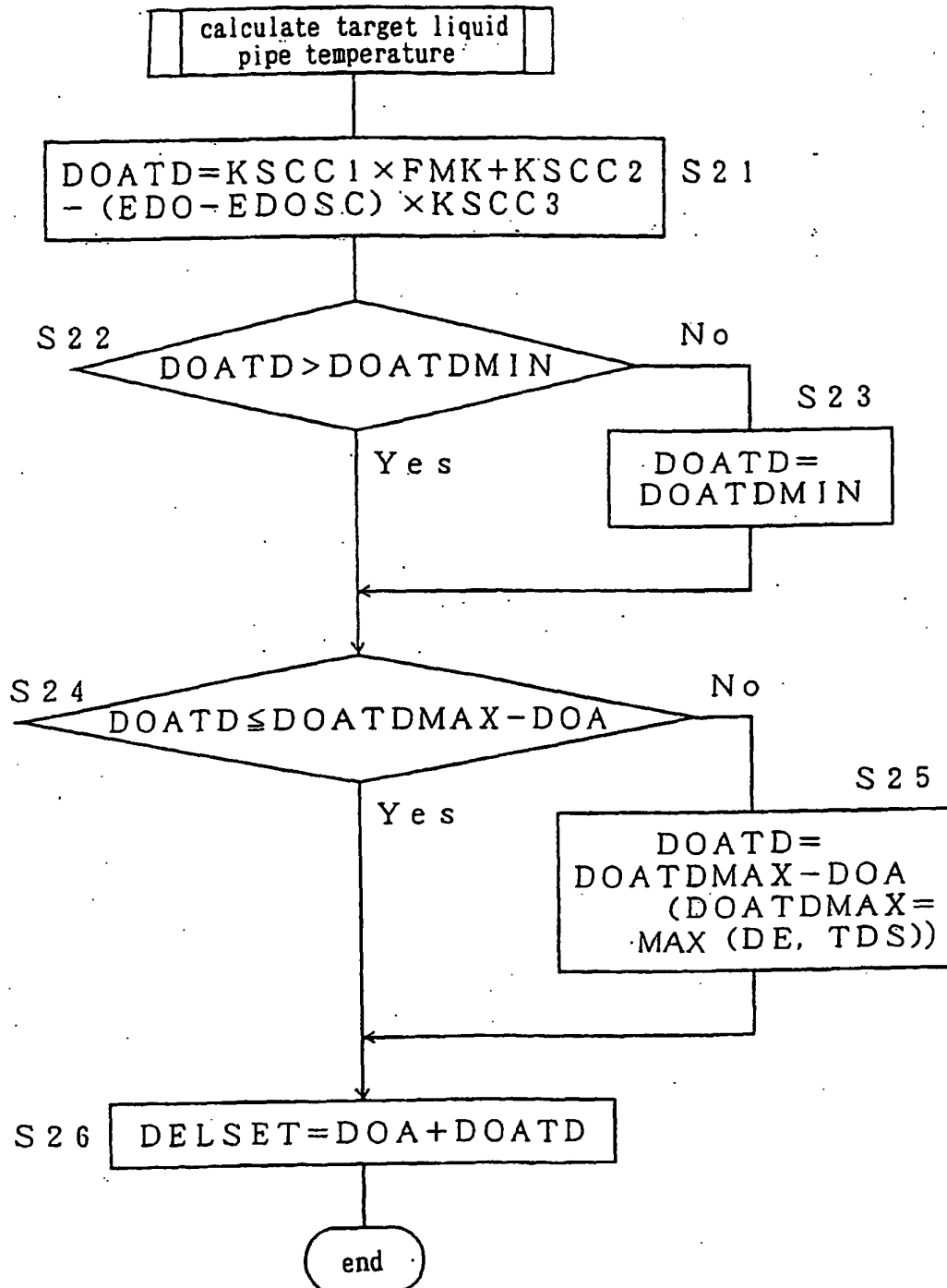


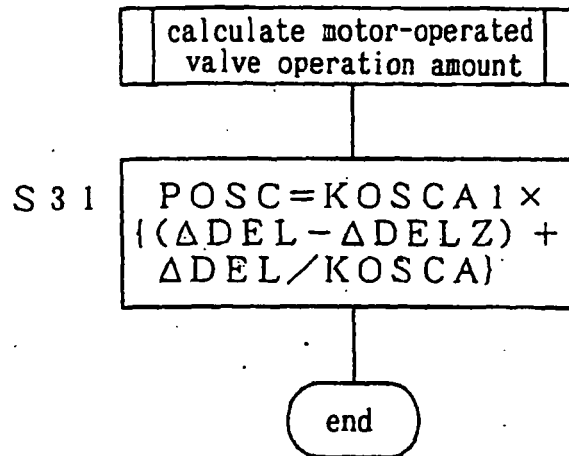
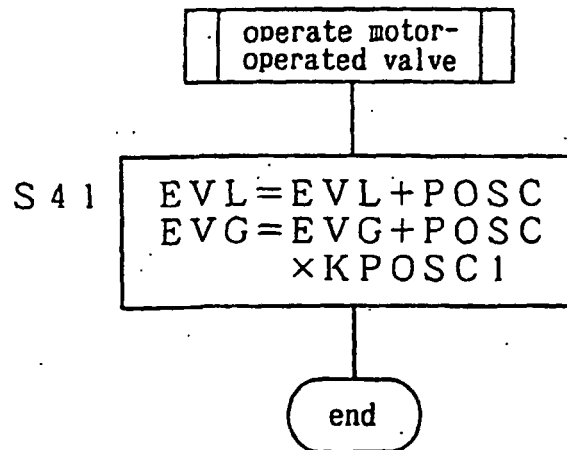
Fig. 4*Fig. 5*

Fig. 6

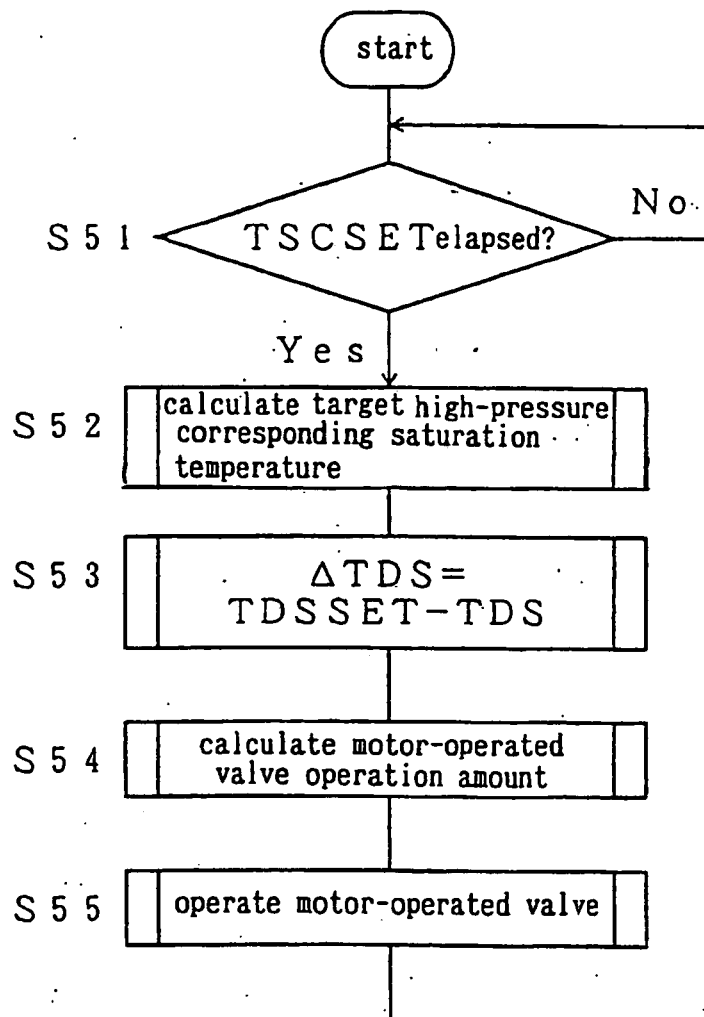


Fig. 7

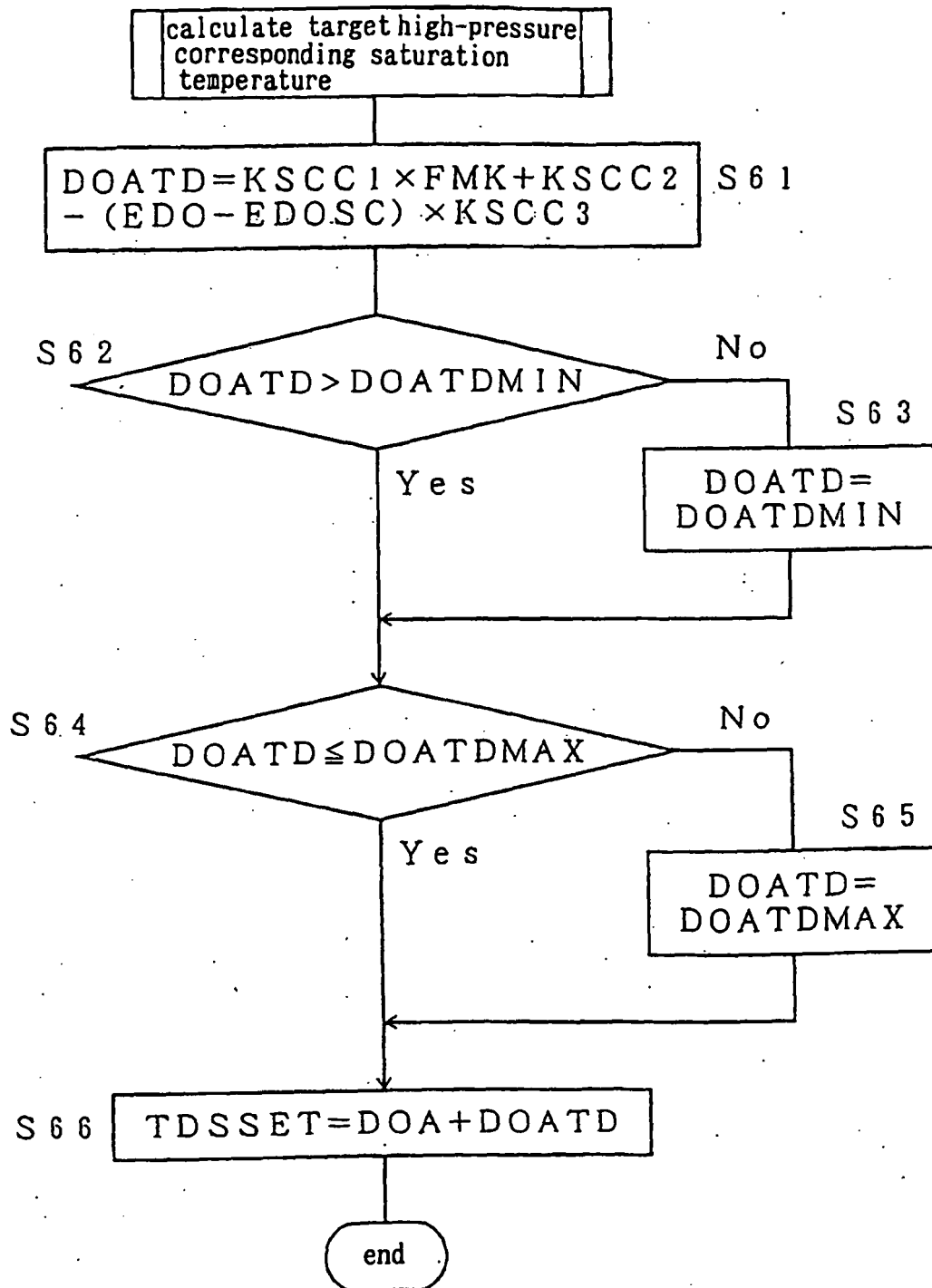


Fig. 8

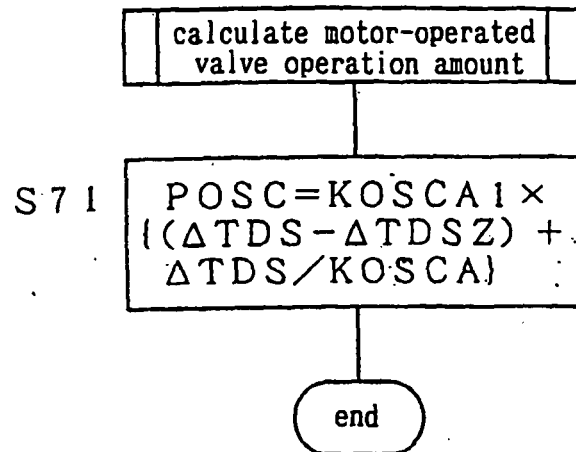


Fig. 9

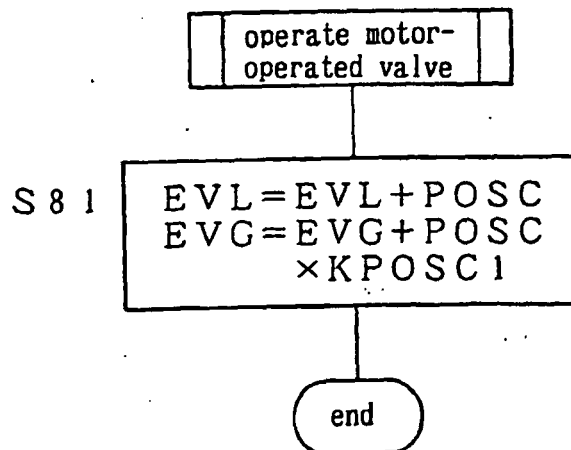


Fig. 10

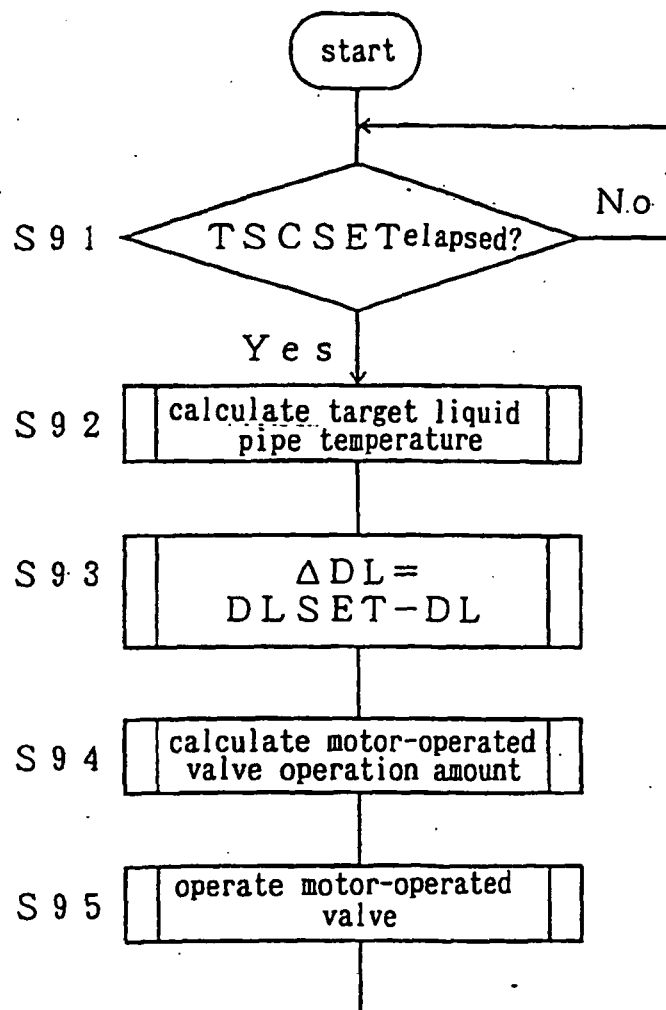


Fig. 11

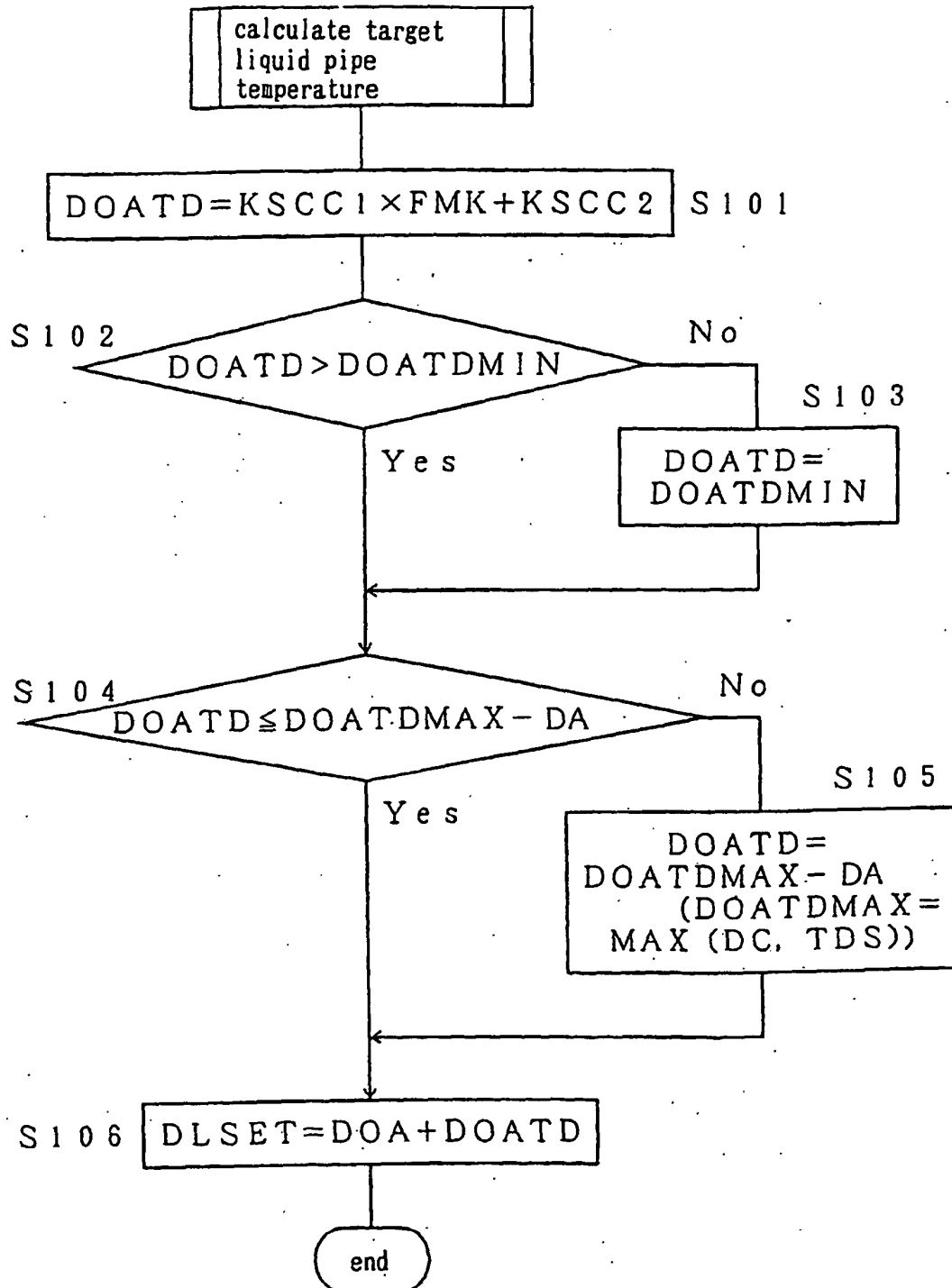


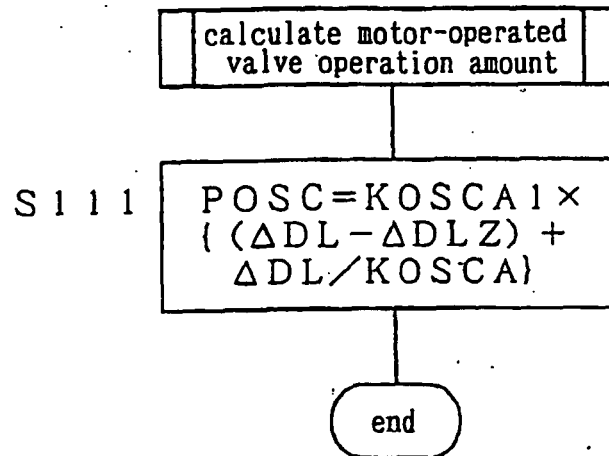
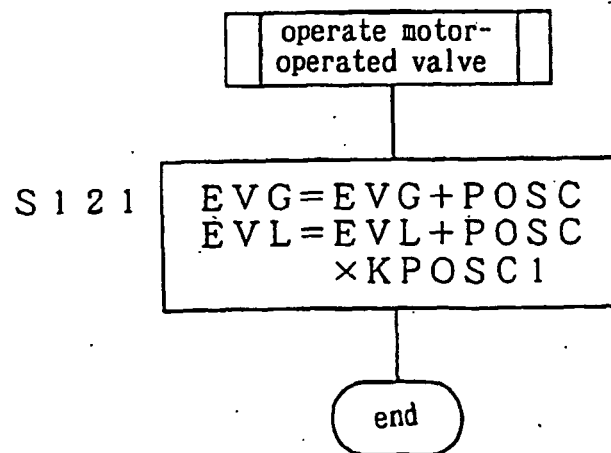
Fig. 12*Fig. 13*

Fig. 14

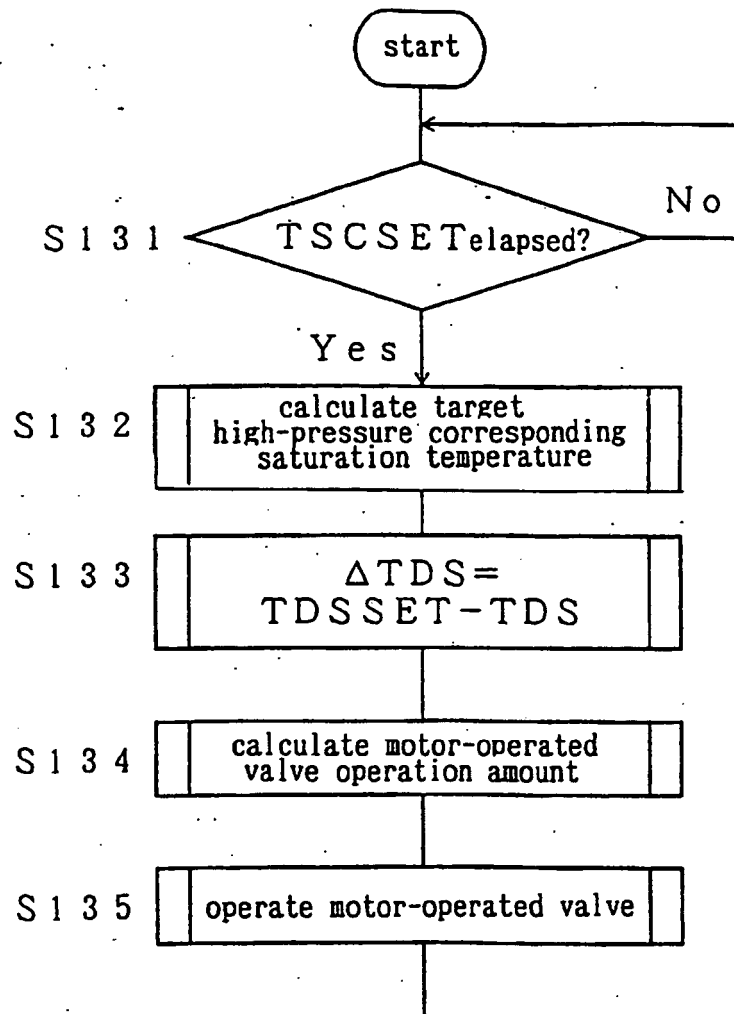


Fig. 15

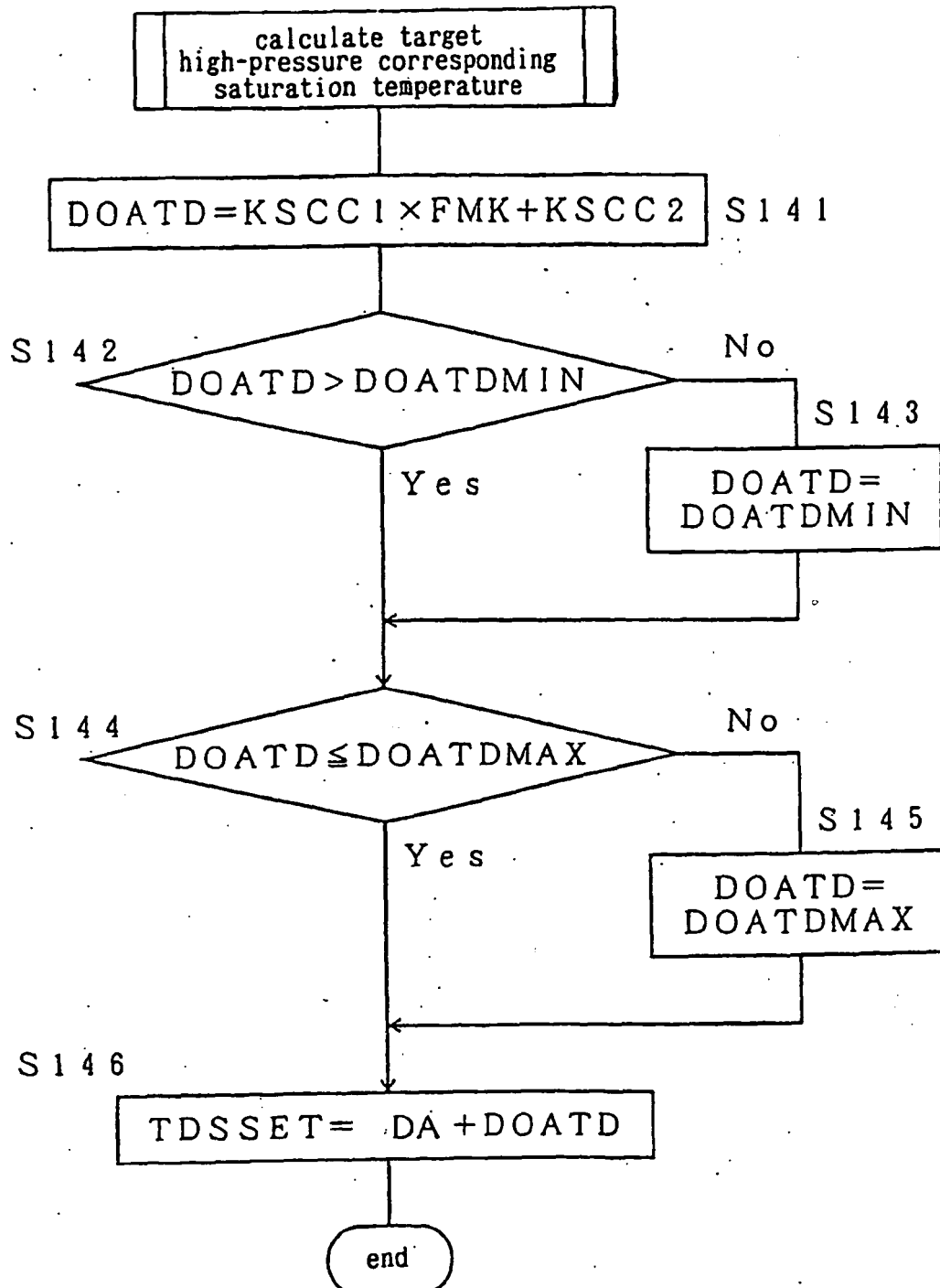


Fig. 16

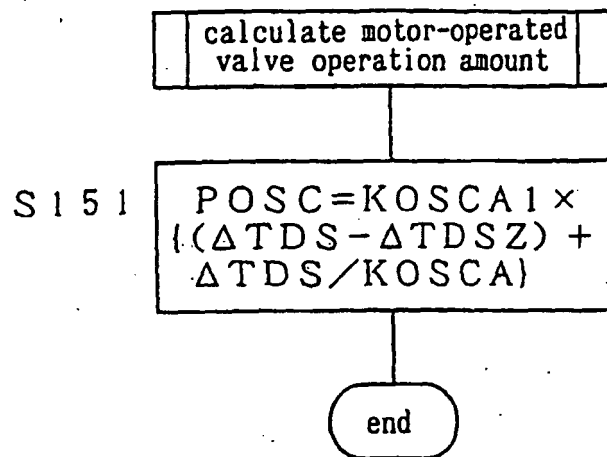


Fig. 17

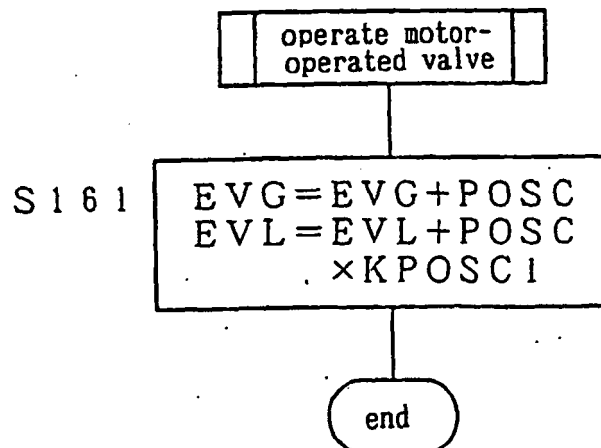


Fig. 18

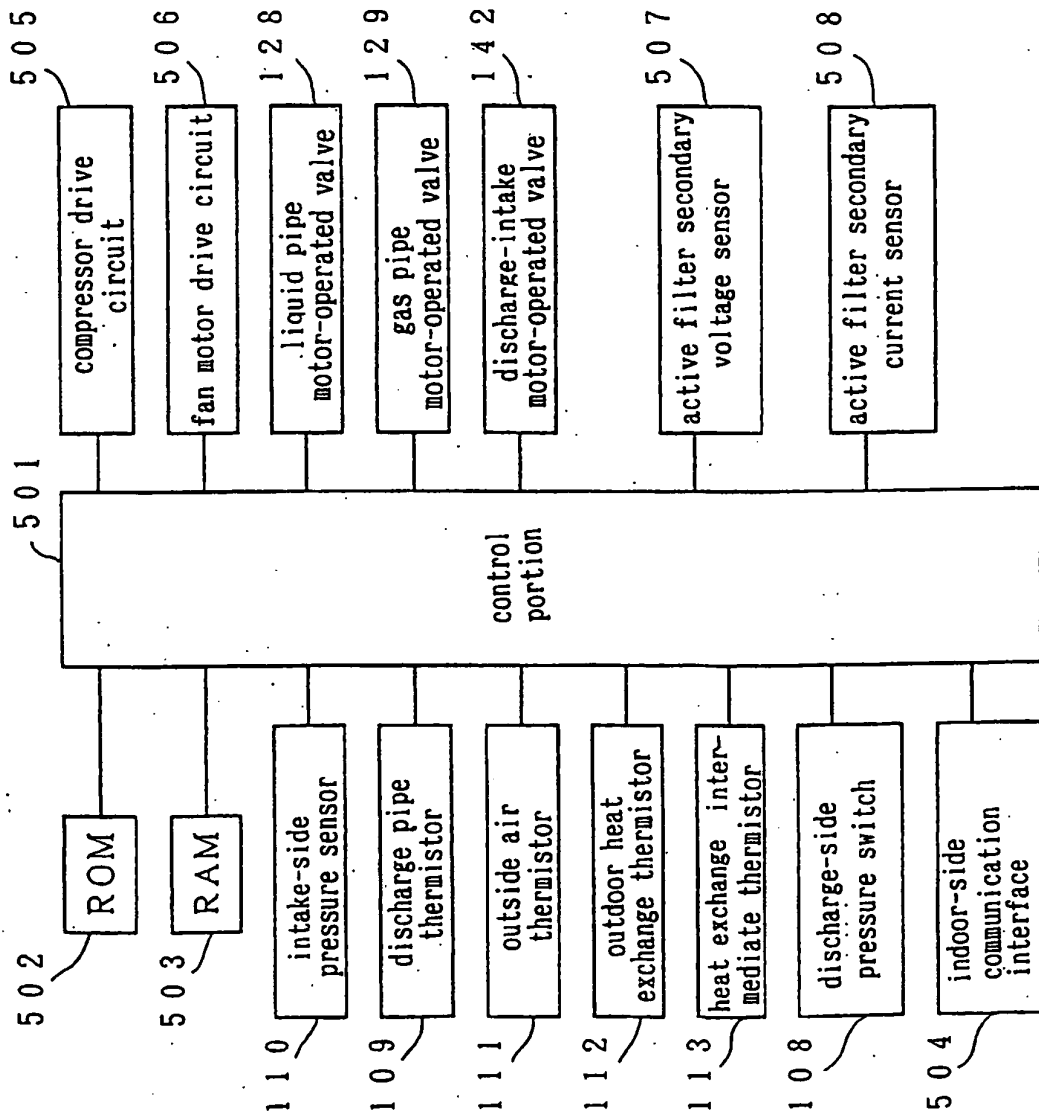


Fig. 19

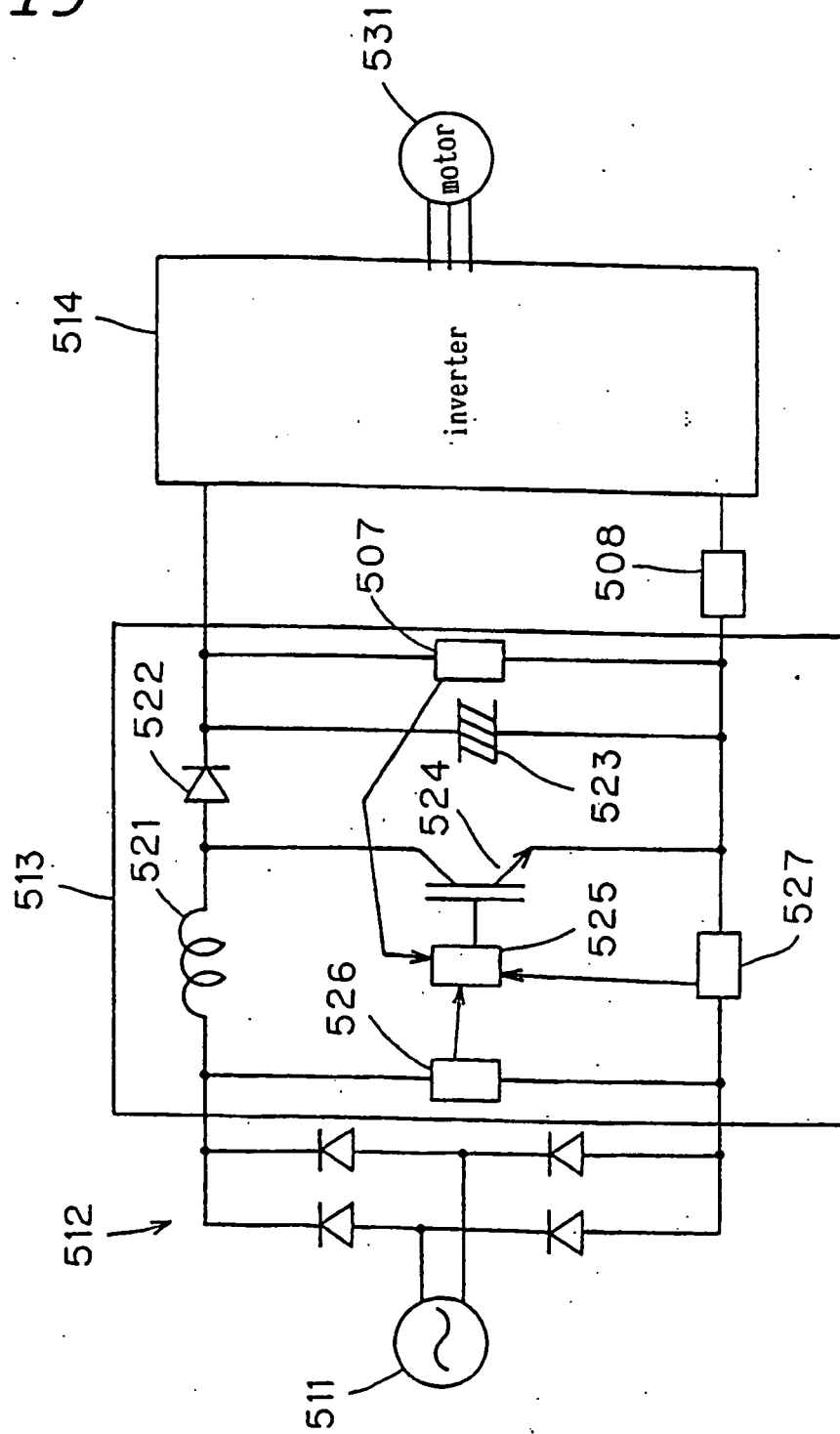


Fig. 20

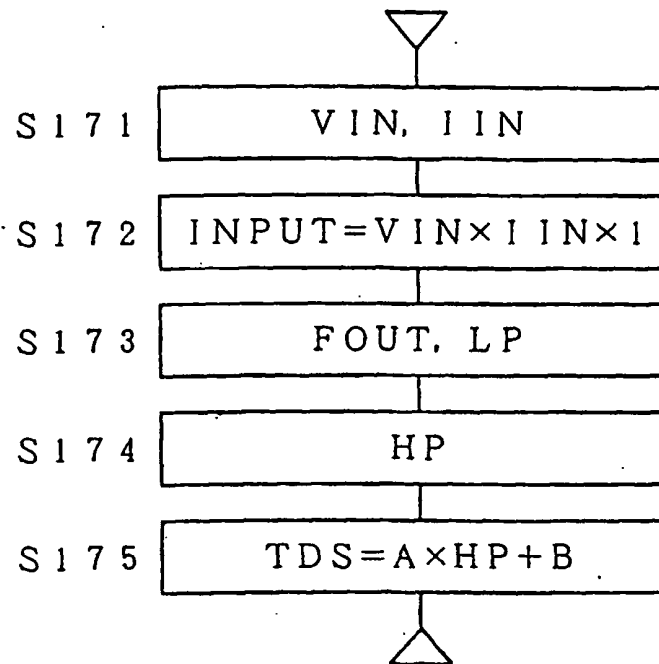


Fig. 21

HP	A	B
$HP \leq HP_1$	A 1	B 1
$HP_1 < HP \leq HP_2$	A 2	B 2
$HP_2 < HP \leq HP_3$	A 3	B 3
$HP_3 < HP \leq HP_4$	A 4	B 4
$HP_4 < HP \leq HP_5$	A 5	B 5
$HP_5 < HP \leq HP_6$	A 6	B 6
$HP_6 < HP \leq HP_7$	A 7	B 7
$HP_7 < HP \leq HP_8$	A 8	B 8
$HP_8 < HP$	A 9	B 9

REFERENCES CITED IN THE DESCRIPTION

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