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DESCRIPTION

Technical Field

[0001] The present invention relates to a refrigeration cycle apparatus using an internal heat exchanger; more particularly to a refrigerant control for stably securing performance.

Background Art

[0002] Descriptions will be given to prior art as follows.

[0003] Conventionally, a hot water supply apparatus is proposed as a built-in refrigeration cycle apparatus such as:

a hot water supply apparatus comprising a refrigeration cycle including a compressor, a hot water supply heat exchanger, an electronic expansion valve, and a heat source side heat exchanger whose heat source is an external air, and a hot water supply cycle including a hot water supply heat exchanger and a hot water supply tank,

wherein since ability control means that uses an ability-variable type compressor and ability-controls the compressor in response to changes in external environment conditions of the heat source side heat exchanger is attached, expansion valve opening degree control means for controlling an opening degree of an electronic expansion valve so as to make a discharge temperature of a compressor to be a target value in response to changes in external environment conditions (an external temperature, for example) of the heat source side heat exchanger and rotation speed control means for controlling a rotation speed of the compressor to be a target value in response to changes in the external environment conditions of the heat source side heat exchanger are attached, an opening of the electronic expansion valve is controlled so as to make the discharge temperature of the compressor becomes a target value in response to changes in the external environment conditions (an external temperature, for example) of the heat source side heat exchanger, and the rotation speed of the compressor is controlled to be a target value in response to changes in the external environment conditions of the heat source side heat exchanger, an optimal operation condition can be obtained in which a hot water supply ability and a hot water supply load further match, and a coefficient of performance (COP) can be improved and down-sizing of elements such as an heat exchanger becomes possible. (For example, refer to Patent Document 1 or US2003/0061827).

[0004] A water heater is also proposed such as:

a water heater for heating a hot water supply fluid in a supercritical heat pump cycle where a refrigerant pressure in a high pressure side becomes equal to or more than the critical pressure of the refrigerant comprising:

a compressor,

a radiator that performs heat exchange between a refrigerant discharged from the compressor and a hot water supply fluid and is configured so that a refrigerant flow and the hot water supply fluid flow opposes,

a decompressor for decompressing the refrigerant flowing out of the radiator, and

an evaporator that makes the refrigerant that flows out of the compressor evaporate, makes the refrigerant absorb a heat to discharge it into a suction side of the compressor,

wherein a refrigerant pressure of a high-pressure side is controlled so that a temperature difference (ΔT) between the refrigerant that flows out of the radiator and the hot water supply fluid that flows therein becomes a predetermined temperature difference (ΔT_0). (For example, refer to Patent Document 2) In this example of the prior art, a heat exchange efficiency of the radiator can be enhanced to improve efficiency of a heat pump.

[0005]

[Patent Document 1] Japanese Patent Gazette No. 3501369 (pp.6; Fig. 1)

[Patent Document 2] Japanese Patent Gazette No.3227651 (pp.1 - 3; Fig. 2)

Summary of Invention

Problems to be Solved by the Invention

[0006] Both of the above examples of the prior art control refrigerant conditions so that a discharge temperature of the compressor or a temperature difference (ΔT) between the refrigerant that flows out of the radiator and the hot water supply fluid that flows therein becomes a target value to achieve an efficient operation. However, there was a problem that in the vicinity where an efficiency (COP) of the refrigeration cycle becomes maximum, a control based only on an inlet side (the above discharge temperature) of the radiator or an outlet side (the above temperature difference ΔT) is difficult to achieve stable and efficient operation conditions because changes in the discharge temperature or the temperature difference ΔT are small. In addition, since an operation in which an internal heat exchanger exists in the refrigerant circuit is not considered, there was a problem that to control to achieve stable and efficient operation conditions is difficult.

[0007] The present invention is made to solve the above problems in the prior art. The object is to obtain a refrigeration cycle apparatus capable of stably achieving efficient operation conditions by controlling operation values based on standard conditions of the radiator and outlet conditions of the radiator to be a target value.

Means for Solving the Problems

[0008] In order to solve the above problems, the refrigeration cycle apparatus according to the present invention includes at least a compressor, a radiator, decompression means capable of changing an open degree, a heat absorber, an internal heat exchanger that performs heat exchange between a refrigerant at an outlet of the radiator and the refrigerant at the outlet of the heat absorber. The refrigeration cycle apparatus is characterized in that at least first refrigerant conditions detection means for detecting standard conditions of the radiator and second refrigerant conditions detection means for detecting refrigerant conditions between an outlet of the radiator and a high-pressure side inlet of an internal heat exchanger are provided, and an opening degree of decompression means is controlled so that a calculation value calculated based on an output of the first refrigerant conditions detection means and the output of the second refrigerant conditions detection means becomes a target value.

Effect of the Invention

[0009] According to the present invention, the expansion valve opening degree is controlled so that the COP becomes maximum based on standard conditions of the radiator and refrigerant conditions of the radiator outlet part, so that a refrigerant cycle apparatus capable of stably achieving efficient operation can be obtained.

Brief Description of Drawings

[0010]

[Fig. 1] Fig. 1 is a diagram showing a configuration of a refrigeration cycle apparatus of the present invention.

[Fig. 2] Fig. 2 is a diagram showing an operation behavior on a P-h diagram of the present invention.

[Fig. 3] Fig. 3 is a diagram showing a temperature distribution of a refrigerant and water in a water heat exchanger of the present invention.

[Fig. 4] Fig. 4 is a diagram showing cycle conditions against an expansion valve opening degree of the present invention.

[Fig. 5] Fig. 5 is a diagram showing changes in each calculation value, heating ability, and COP against an expansion valve

opening degree of the present invention.

[Fig. 6] Fig. 6 is a diagram showing changes in other calculation value, heating ability, and COP against an expansion valve opening degree of the present invention.

[Fig. 7] Fig. 7 is a diagram showing a control flowchart of the present invention.

Descriptions of Codes and Symbols

[0011]

- 1
compressor
- 2
radiator (water heat exchanger)
- 3
expansion valve
- 4
heat absorber (evaporator)
- 5
internal heat exchanger
- 20
hot water supply side pump
- 21
hot water storage tank
- 22
use side pump
- 23, 24, 25
on-off valve
- 29
blower
- 30, 31, 32, 33, 41, 42, 52
temperature detection means
- 35, 51
pressure detection means
- 40
controller
- 50
heat source apparatus
- 60
hot water storage apparatus

Best Mode for carrying Out the Invention

[0012] Fig. 1 shows a configuration diagram of the refrigerant cycle apparatus according to the invention. In the figure, the refrigerant cycle apparatus according to the present embodiment is a hot water supply apparatus using carbon dioxide (hereinafter, CO₂ as a refrigerant, composed of a heat source apparatus 50, a hot water storage apparatus 60, and a controller 40 for controlling these. It shows an example of the hot water supply apparatus, however, it is not limited thereto. The apparatus may be an air conditioner. In the same way, the refrigerant is not limited to carbon dioxide but an HFC refrigerant may be used.

[0013] The heat source apparatus 50 is composed of a compressor 1 for compressing the refrigerant, a radiator 2 (hereinafter, referred to "water heat exchanger") for taking out heat of a high-temperature high-pressure refrigerant compressed in the

compressor 1, an internal heat exchanger 5 for further cooling the refrigerant output from the water heat exchanger 2, a decompressor 3 (hereinafter, referred to "expansion valve") that decompresses the refrigerant and whose opening degree can be changed, an heat-absorber 4 (hereinafter, referred to "evaporator") for evaporating the refrigerant decompressed in the expansion valve 3, and an internal heat exchanger 5 for further heating the refrigerant flowed out of the evaporator 4. That is, the internal heat exchanger 5 is a heat exchanger that heat-exchanges the refrigerant at an outlet of the water heat exchanger 2 with the refrigerant at the outlet of the evaporator 4. A blower 29 is provided for sending air on an outer surface of the evaporator 4. There are also provided first temperature detection means 30 for detecting a discharge temperature of the compressor 1, second temperature detection means 31 for detecting an outlet temperature of the water heat exchanger 2, fifth temperature detection means 32 for detecting an inlet refrigerant temperature of the evaporator 4, and sixth temperature detection means 33 for detecting a suction temperature of the compressor 1. In addition, the first temperature detection means 30 and the second temperature detection means 31 correspond to a first refrigerant conditions detection means and second refrigerant conditions detection means respectively in an example of control in Fig. 7 to be described later.

[0014] A hot water storage apparatus 60 is connected with the water heat exchanger 2, which is a radiator, via piping, being composed of a heat source side pump 20, a hot water storage tank 21, a use side pump 22, and on-off valves 23, 24, 25. Here, on-off valves 23, 24, 25 may be a simple valve only for switching operation or an opening variable valve. When a water level of the hot water storage tank 21 drops, the on-off valves 24, 25 are closed, the on-off valve 23 is opened, and hot water storage operation is performed in which supplied water is heated up to a predetermined temperature. When a heat dissipation loss is large and the temperature in the hot water storage tank 21 decreases such as in winter, the on-off valves 23, 25 are closed, the on-off valve 24 is opened, and circulation heating operation is performed in which low-temperature hot water in the hot water storage tank 21 is re-boiled. At the time of using the hot water supply, the on-off valves 23, 24 are closed, the on-off valve 25 is opened, the use side pump 22 starts operation to transfer stored hot water to the use side. At an inlet side of the water heat exchanger 2, third temperature detection means 41 is attached for detecting an inlet temperature of a medium (water) to be heated. At an outlet side of the water heat exchanger 2, fourth temperature detection means 42 is attached for detecting the outlet temperature of the medium (water) to be heated.

[0015] A controller 40 performs calculation using detected values from first temperature detection means 30, second temperature detection means 31, fifth temperature detection means 32, sixth temperature detection means 33, third temperature detection means 41, and fourth temperature detection means 42 to control an opening degree of the expansion valve 3, a rotation speed of the compressor 1, and the rotation speed of the hot water supply side pump 20, respectively.

[0016] Fig. 2 is a P-h diagram describing cycle conditions during hot water storage operation in the refrigeration cycle apparatus shown in Fig. 1. In Fig. 2, solid lines denote refrigerant conditions at a certain expansion valve opening degree and A, B, C, D, and E denote refrigerant conditions in the hot water storage operation. At the time of the hot water storage operation, a high-temperature high-pressure refrigerant (A) discharged from the compressor 1 flows into the water heat exchanger 2. In the water heat exchanger 2, the refrigerant heats supplied water while dissipating heat to water circulating the hot water storage circuit to decrease the own temperature. A refrigerant (B) flowed out of the water heat exchanger 2 dissipates heat in the internal heat exchanger 5 to further decrease (C) the temperature, being decompressed (D) by the expansion valve 3 to turn into a low-temperature low-pressure refrigerant. The low-temperature low-pressure refrigerant absorbs heat from the air in the evaporator 4 to evaporate (E). The refrigerant flowed out of the evaporator 4 is heated in the internal heat exchanger 5 to turn into a gas (F) and sucked by the compressor 1 to form a refrigeration cycle.

[0017] Here, the expansion valve 3 is controlled so that a suction superheat degree of the compressor 1 becomes a target value (for example, 5 to 10 °C). Specifically, based on a detection value of fifth temperature detection means 32 detecting an inlet refrigerant temperature of the evaporator 4, a temperature decrease amount due to a pressure loss in the evaporator 4 and the internal heat exchanger 5 is corrected, an evaporation temperature (ET) is estimated, a suction superheat degree SH_s is calculated by the following formula using a detection value (T_s) of sixth temperature detection means 33 detecting a suction temperature of the compressor 1.

$$SH_s = T_s - ET$$

[0018] Using the above formula, an opening degree of the expansion valve 3 is controlled so that SH_s becomes a target value. An example is given in which an evaporation temperature (ET) is estimated based on the detection value of the fifth temperature detection means 32, however, it is not limited thereto. Pressure detection means (second pressure detection means) 51 (refer to Fig. 1) is installed between a low-pressure side outlet of the internal heat exchanger 5 and the inlet of the compressor 1, and from the detection value, a refrigerant saturation temperature may be obtained. A suction superheat degree control precedes other high efficiency operation control because a function to prevent liquid return of the compressor 1 precedes a function to efficiently

operate the water heat exchanger 2 from the viewpoint of securing reliability of the equipment.

[0019] Next, operation on the P-h diagram in the case when the opening degree of the expansion valve 3 is made smaller is denoted by broken lines in Fig. 2. When the opening degree of expansion valve 3 is made smaller, the refrigerant flow amount flowing from the expansion valve 3 to the evaporator 4 decreases and the suction superheat degree of the compressor 1 temporarily increases. In addition, since the refrigerant shifts to a high pressure side, the pressure on the high pressure side increases and a discharge temperature becomes high. At the same time, a water heat exchanger output temperature decreases so that a temperature difference in the becomes constant. When the water heat exchanger output temperature decreases, a heat exchange amount in the internal heat exchanger 5 decreases, and as a result, the suction superheat degree becomes almost the same state as that of before the opening degree of the expansion valve 3 is made smaller to indicate a constant value. That is, a change in opening degree of the expansion valve 3 is absorbed by the heat exchange amount of the internal heat exchanger 5 (the heat exchange amount varies in response to the opening degree of the expansion valve 3) to make a change in the suction superheat degree small. Accordingly, control of the suction superheat degree of the compressor 1 alone cannot secure heating ability in the water heat exchanger 2 and efficiency is lowered. Therefore, new control is required in order to secure heating ability and improve operation efficiency.

[0020] Next, descriptions will be given to why a local maximal value occurs in performance (COP) using a temperature distribution in the water heat exchanger shown in Fig. 3.

[0021] Fig. 3 shows a refrigerant and water temperature distribution in the water heat exchanger 2. In the figure, thick solid lines show a change in refrigerant temperature, and a thin solid lines denote a change in water temperature. ΔT_1 denotes a temperature difference between the water heat exchanger inlet temperature and water outlet temperature, and ΔT_2 denotes a temperature difference between the water heat exchanger outlet temperature and water inlet temperature. ΔT_p is a temperature difference at a pinch point where the temperature difference between a refrigerant and water in the water heat exchanger 2 becomes minimum. ΔT denotes a temperature difference between the water heat exchanger inlet temperature and the water heat exchanger outlet temperature. As shown by a cycle state against the expansion valve opening degree in Fig. 4, when a discharge temperature is increased by decreasing the expansion valve 3 opening degree, under a condition when heating ability in the water heat exchanger 2 is almost constant, the outlet temperature of the water heat exchanger 2 decreases so that an average temperature difference of the refrigerant and water in the water heat exchanger 2 is maintained, and the temperature difference ΔT_p of pinch point also decreases. Further, as the refrigerant amount shifts to a high pressure side, a discharge pressure rises to increase an input and COP is lowered. To the contrary, when the expansion valve 3 opening degree is made large and the discharge temperature is lowered, the outlet temperature of the water heat exchanger 2 increases so that an average temperature difference between the refrigerant and water in the water heat exchanger 2 is maintained. The temperature difference ΔT_p at the pinch point also increases, however, a heating ability ratio becomes small and COP is lowered. Accordingly, as shown by broken lines in the figure, a suitable expansion opening degree exists that makes COP maximum.

[0022] Next, Fig. 5 shows changes in operation values obtained from the temperature of each part when the opening degree of the expansion valve 3 changes. In Fig. 5, the horizontal axis represents the opening degree (%) of the expansion valve 3, and the vertical axis represents the suction superheat degree, discharge temperature, temperature difference ΔT_2 between the outlet temperature of the water heat exchanger and water inlet temperature, heating ability ratio, COP ratio. The heating ability ratio and COP ratio show a ratio when a maximum value against the expansion valve opening degree is set as 100%, respectively. Against changes in the opening degree of the expansion valve 3, changes in the suction superheat degree can be regarded as almost a constant value, so that it is understood that changes in the heating ability ratio and the COP ratio cannot be judged by the suction superheat degree. When controlling the COP to be maximum based on the temperature difference ΔT_2 between the discharge temperature and the outlet temperature of the water heat exchanger and water inlet temperature, changes in the discharge temperature and temperature difference ΔT_2 are small in the vicinity of the expansion valve opening degree when the COP reaches maximum as shown by a dotted line in the figure, so that it is found that a high accuracy temperature measurement is required for controlling COP to be maximum.

[0023] Next, Fig. 6 shows changes in other operation values obtained from temperatures of each part when the opening degree of the expansion valve 3 is changed. In Fig. 6, the horizontal axis represents the opening degree (%) of the expansion valve 3. The vertical axis represents an outlet/inlet temperature difference ΔT_{hx} of the internal heat exchanger, a temperature difference ΔT between a discharge temperature and an outlet temperature of the water heat exchanger, a total temperature difference $\Sigma \Delta T$ of the above ΔT_1 and ΔT_2 , heating ability, and a COP ratio, respectively. Characteristics of Fig. 6 shows that operation can be performed in the vicinity where the COP becomes maximum by either controlling a heat exchange amount of the internal heat exchanger 5 based on the temperature difference ΔT_{hx} between the outlet and inlet of the internal heat exchanger or controlling the heat exchange amount of the water heat exchanger 2 based on the total temperature difference $\Sigma \Delta T$ of ΔT_1 and ΔT_2 of the water heat exchanger 2. Further, the temperature difference ΔT between the discharge temperature and the outlet temperature

of the water heat exchanger significantly changes in the vicinity of the expansion valve opening degree at which the COP becomes maximum, so that it is understood that a deviation from the maximum value of the COP could be controlled to be small based on the temperature difference ΔT . Here, only the case of the temperature difference ΔT is shown, however, the same effect can be expected by controlling based on the difference ($\Delta T_1 - \Delta T_2$) of the temperature differences ΔT_1 and ΔT_2 . Thus, it is possible to achieve an operation in the vicinity of the maximum efficiency by adopting a high-pressure side outlet temperature of the internal heat exchanger 5 for ΔT_{hx} , the discharge temperature for ΔT , and the discharge temperature and a water side outlet/inlet temperatures for $\Sigma \Delta T$.

[0024] As is understood from Fig. 6, a total temperature difference $\Sigma \Delta T$ of the temperature difference ΔT_1 between the water heat exchanger inlet temperature and water outlet temperature and the temperature difference ΔT_2 between the water heat exchanger outlet temperature and water inlet temperature becomes a minimum. The control based on such an index has a physical meaning and being reasonable. However, high-precision temperature detection is required because change in temperature is small in the vicinity where the COP becomes a maximum compared with the temperature difference ΔT . Further, from Fig. 3, it is considered that when the COP becomes a maximum value, a temperature difference ΔT_p at a pinch point is almost the same as that of ΔT_2 between the water heat exchanger outlet temperature and water inlet temperature. This is because a maximum performance is shown when two temperature differences that become minimum in the water heat exchanger 2 become equal without being biased to either of them when considering characteristics of the heat exchanger. Accordingly, it is allowable to control the expansion valve 3 so as to make ΔT_p and ΔT_2 to be equal.

[0025] Next, descriptions will be given to an example of a control operation of the refrigeration cycle apparatus of Fig. 1 in which an expansion valve opening degree is controlled so as to make a suction superheat degree and the above temperature difference ΔT to converge at target values.

[0026] Fig. 7 is a flowchart showing a control operation of the refrigeration cycle apparatus. With the present invention, for the purpose of giving a priority to reliability of products, the suction superheat degree (SHs) control of the compressor 1 precedes the temperature difference ΔT control for securing the heating ability.

[0027] Firstly, when the suction superheat degree (SHs) is smaller than a target value (SHm) by a preset convergence range ΔSH or less (S101), the expansion valve opening degree is lowered until the suction superheat degree (SHs) converges. Thus, when the suction superheat degree (SHs) is secured, the temperature difference ΔT is made to converge at the target value. Specifically, when the temperature difference ΔT is smaller than a target value (ΔT_m) by a preset convergence range δT or less (S102), the expansion opening degree is lowered and ΔT is made to converge. Thus, lower limit values of the suction superheat degree (SHs) and the temperature difference ΔT can be suppressed.

[0028] Next, when the suction superheat degree (SHs) is larger than the target value (SHm) by a preset convergence range ΔSH or more (S103), the expansion valve opening degree is increased until the suction superheat degree (SHs) converges. Thus, when the suction superheat degree (SHs) is converged, the temperature difference ΔT is made to converge at the target value. Thus, when the suction superheat degree (SHs) is converged, the temperature difference ΔT is made to converge at the target value. Specifically, when the temperature difference ΔT is larger than the target value (ΔT_m) by a preset convergence range δT or more (S104), the expansion opening degree is increased and ΔT is made to converge. Thus, upper limit values of the suction superheat degree (SHs) and the temperature difference ΔT can be suppressed. An example is shown in which a priority is given to control the suction superheat degree, however, it is not limited thereto when using a compressor which is resistant to liquid return. The same effect can be expected even when the priority order is exchanged. Through the above control, the suction superheat degree (SHs) and the temperature difference ΔT are converged at target values.

[0029] In the above, descriptions are given to an example in which the suction superheat degree (SHs) and the temperature difference ΔT are controlled to converge at target values (SHm, ΔT_m), however, it is allowable that, in place of the temperature difference ΔT , a total temperature difference $\Sigma \Delta T$ of ΔT_1 and ΔT_2 , a difference between ΔT_1 and ΔT_2 ($\Delta T_1 - \Delta T_2$), or ΔT_{hx} can be used to control them to converge at a target value, respectively. When using $\Sigma \Delta T$ and ($\Delta T_1 - \Delta T_2$), they are obtained by calculating detection temperatures by the first temperature detection means 30, the second temperature detection means 31, the third temperature detection means 41, and the fourth temperature detection means 42. When using ΔT_{hx} , internal heat exchanger outlet temperature detection means 52 is attached (refer to Fig. 1) between a high-pressure side outlet of the internal heat exchanger 5 and an inlet of the expansion valve 3, the temperature difference ΔT_{hx} is obtained from a detection temperatures by the second temperature detection means 31 and the internal heat exchanger outlet temperature detection means 52.

[0030] Since, in the invention, in addition to suction superheat degree control of the compressor, the expansion valve opening

degree is made to be controlled so that the COP becomes maximum based on a temperature difference $\Delta T_1 - \Delta T_2$ between the discharge temperature and the water heat exchanger outlet temperature, a high efficiency refrigeration cycle apparatus can be obtained.

[0031] A refrigerant saturation temperature (ET) is obtained based on an output of the fifth temperature detection means 32 or pressure detection means, the suction superheat degree (SHs) is obtained by the detection temperature (Ts) of the sixth temperature detection means and the refrigerant saturation temperature (ET), and the expansion valve opening degree is controlled so that the suction superheat degree (SHs) becomes a target value, so that the superheat degree of the suction part of the compressor 1 is secured, liquid return to the compressor 1 can be prevented, and reliability can be secured. In the example of Fig. 1, descriptions are given to an example in which the fifth temperature detection means 32 is provided between the expansion valve 3 and the evaporator 4, it can be disposed at any position between the inlet of the evaporator 4 and a low-pressure side inlet of the internal heat exchanger 5.

[0032] In the present invention, when controlling the superheat degree and the above temperature difference $\Delta T_1 - \Delta T_2$, the control of the superheat degree precedes the control of the above temperature differences. From this point, the reliability of the compressor 1 is secured.

[0033] In the present invention, the radiator is composed of the water heat exchanger, so that a high efficiency hot water supply apparatus can be obtained.

[0034] In said previously mentioned refrigerant cycle apparatuses

sixth temperature detection means for detecting the refrigerant temperature between a low-pressure side outlet of said internal heat exchanger and an inlet of said compressor may be provided,

superheat degree of a compressor suction part may be calculated from a refrigerant saturation temperature at a detection point of said sixth temperature detection means and a detection temperature by said sixth temperature detection means, and

the opening degree of said decompression means may be controlled such that said superheat degree becomes the target value.

[0035] In said previously mentioned refrigerant cycle apparatus

second pressure detection means may be provided between the low-pressure side outlet of said internal heat exchanger and the inlet of said compressor and

said refrigerant saturation temperature may be calculated based on a detection value of said second pressure detection means.

[0036] In said refrigerant cycle apparatus fifth temperature detection means may be provided between the inlet of said heat absorber and the low-pressure side inlet of said internal heat exchanger and

said refrigerant saturation temperature may be calculated based on the detection temperature of said fifth temperature detection means.

[0037] In said refrigerant cycle apparatus a priority may be given to control said superheat degree over said temperature difference.

[0038] In any one of said previously mentioned refrigerant cycle apparatuses said radiator may be a heat exchanger that exchanges heat with water.

[0039] In any one of said previously mentioned refrigerant cycle apparatuses carbon dioxide may be used as a refrigerant.

REFERENCES CITED IN THE DESCRIPTION

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

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- [JP2501369B \[0005\]](#)
- [JP3227651B \[0005\]](#)

Kølekredsløbsapparat

Patentkrav

1. Kølekredsløbsapparat omfattende:

- 5 mindst en kompressor (1), en radiator (2), dekompressionsorgan (3) i stand til at ændre en åbningsgrad, en varmeoptager (4), en indvendig varmeveksler (5) der udfører varmeveksling mellem et kølemiddel ved et udløb af radiatoren (2) og kølemidlet ved et udløb af varmeoptageren (4), hvor
- 10 første temperaturdetektionsorgan (30) til at detektere en kølemiddeltemperatur mellem et udløb af kompressoren (1) og et indløb af radiatoren (2) og andet temperaturdetektionsorgan (31) til at detektere kølemiddeltemperaturen mellem udløbet af radiatoren (2) og et højtrykssideindløb af den indvendige varmeveksler (5), tredje
- 15 temperaturdetektionsorgan (41) til at detektere en indløbstemperatur af et medie som skal opvarmes og fjerde temperaturdetektionsorgan (42) til at detektere en udløbstemperatur af mediet som skal opvarmes er tilvejebragt, **kendetegnet ved at**
- 20 en åbningsgrad af dekompressionsorganet (3) styres således at en forskel ($\Delta T_1 - \Delta T_2$) mellem en anden temperaturforskel (ΔT_1) mellem en detektionstemperatur med det første temperaturdetektionsorgan (30) og detektionstemperaturen med det fjerde temperaturdetektionsorgan (42) og en tredje temperaturforskel (ΔT_2) mellem detektionstemperaturen med det andet temperaturdetektionsorgan (31) og detektionstemperaturen med det
- 25 tredje temperaturdetektionsorgan (41) bliver en målværdi.

2. Kølekredsløbsapparatet ifølge krav 1, hvor

- 30 sjette temperaturdetektionsorgan (33) til at detektere kølemiddeltemperaturen mellem et lavtrykssideudløb af den indvendige varmeveksler (5) og et indløb af kompressoren (1) er tilvejebragt,
- overhedningsgrad af en kompressorsugningsdel beregnes fra en

kølemiddelmætningstemperatur ved et detektionspunkt af det sjette temperaturdetektionsorgan (33) og en detektionstemperatur med det sjette temperaturdetektionsorgan (33), og
åbningsgraden af dekompressionsorganet (3) styres således at
5 overhedningsgraden bliver målværdien.

3. Kølekredsløbsapparatet ifølge krav 2, hvor
andet trykdetektionsorgan (51) er tilvejebragt mellem lavtrykssideudløbet af den indvendige varmeveksler (5) og indløbet af kompressoren (1) og
10 kølemiddelmætningstemperaturen beregnes baseret på en detektionsværdi af det andet trykdetektionsorgan (51).

4. Kølekredsløbsapparatet ifølge krav 2, hvor
femte temperaturdetektionsorgan (32) er tilvejebragt mellem indløbet af
15 varmeoptageren (4) og lavtrykssideudløbet af den indvendige varmeveksler (5)
og
kølemiddelmætningstemperaturen beregnes baseret på detektionstemperaturen af det femte temperaturdetektionsorgan (32).

20 **5.** Kølekredsløbsapparatet ifølge et hvilket som helst af kravene 2 til 4, hvor en prioritet gives til styring af overhedningsgraden over temperaturforskellen.

6. Kølekredsløbsapparatet ifølge et hvilket som helst af kravene 1 til 5, hvor radiatoren (2) er en varmeveksler der veksler varme med vand.

25

7. Kølekredsløbsapparatet ifølge et hvilket som helst af kravene 1 til 6, hvor kuldioxid anvendes som et kølemiddel.

DRAWINGS

FIG. 1

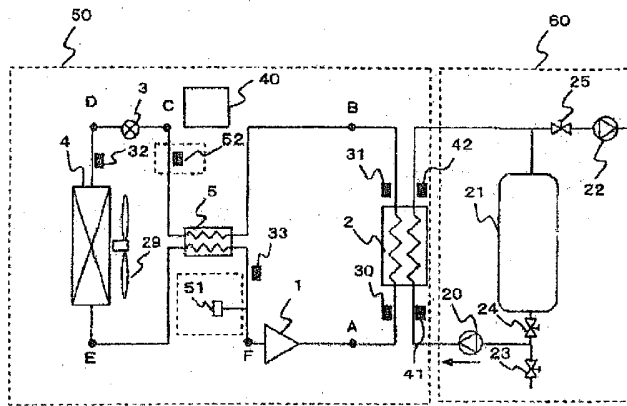


FIG. 2

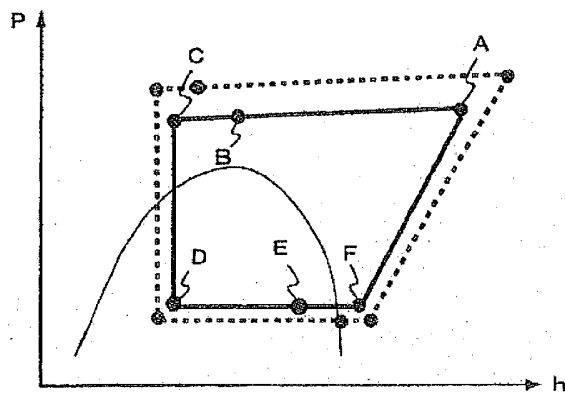


FIG. 3

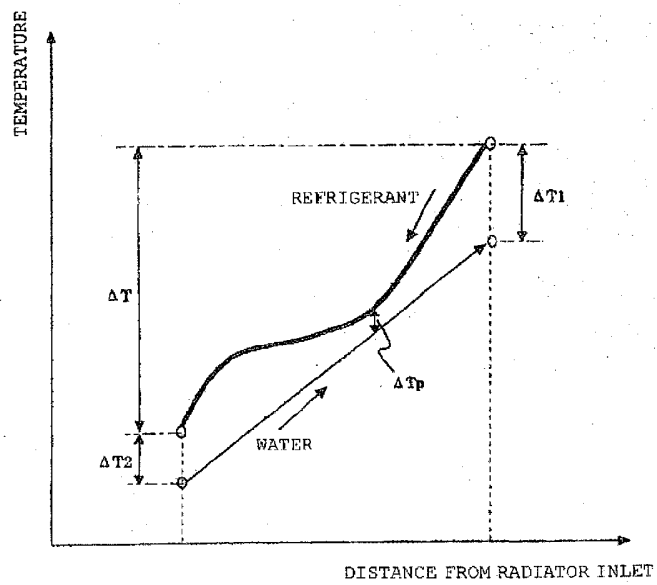


FIG. 4

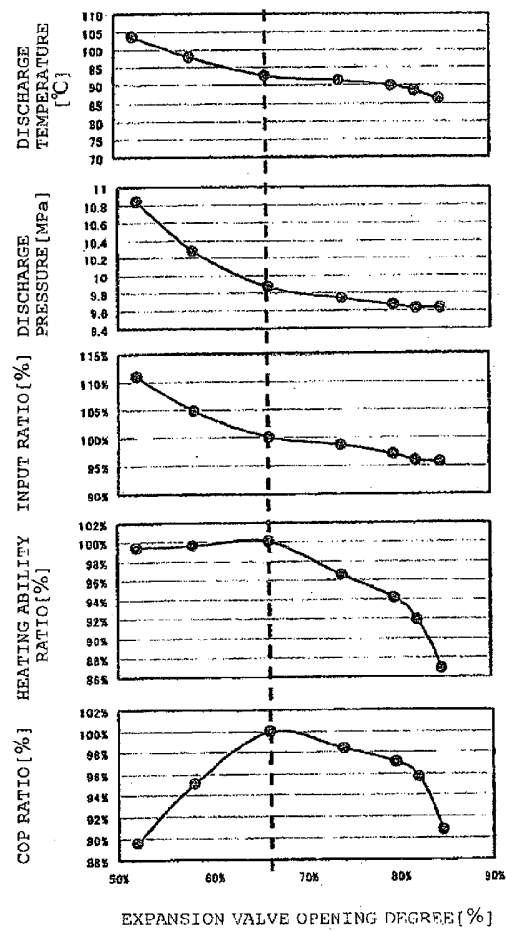


FIG. 5

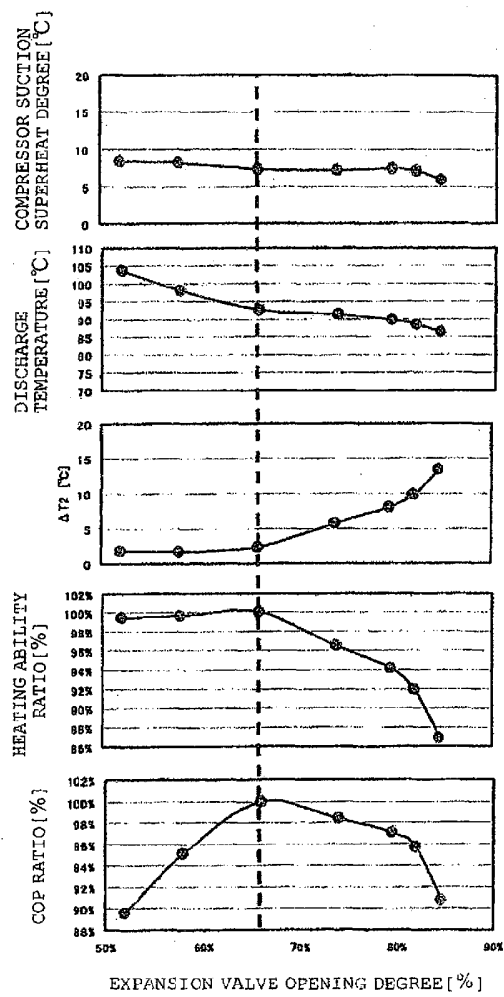


FIG. 6

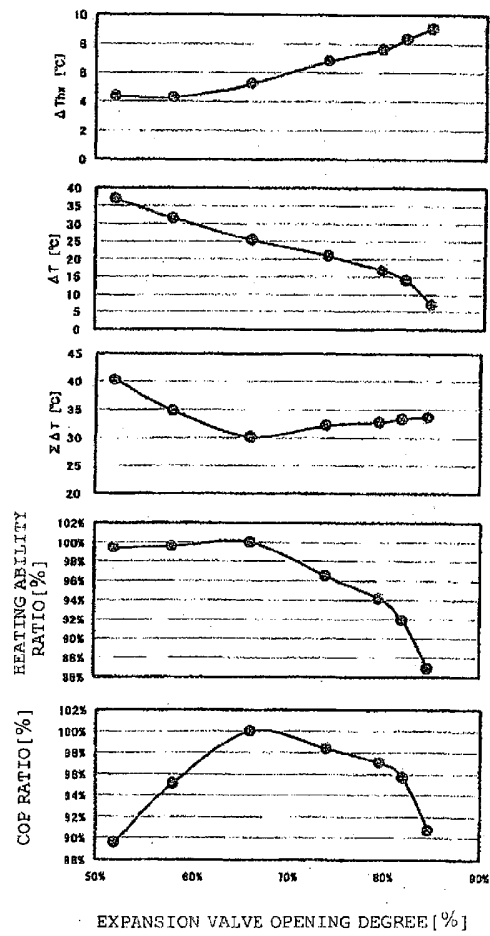


FIG. 7

