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(54) **COOLING SYSTEM AND HEATING SYSTEM**

(57) A refrigerating system (100) and a heating system (200) are provided. The refrigerating system (100) includes a low back pressure rotary compressor (1), a four-way valve (2), an outdoor heat exchanger (3), an indoor heat exchanger (4), a throttling element (4) and a control valve assembly (6). A shell of the low back pressure rotary compressor (1) is formed with an upper gas suction pipe (S1), a middle gas suction pipe (S2) and an exhaust pipe (D). The control valve assembly (6) is connected with a gas suction valve port (21), the upper gas suction pipe (S1) and the middle gas suction pipe (S2) via a first pipeline (G), a second pipeline (F) and a third pipeline (E) respectively. When the refrigerating system is in a refrigerating mode, the control valve assembly (6) controls a gas suction flow of the third pipeline (E) to be larger than that of the second pipeline (F); when the refrigerating system is in a heating mode, the control valve assembly (6) controls the gas suction flow of the second pipeline (F) to be larger than that of the third pipeline (E).

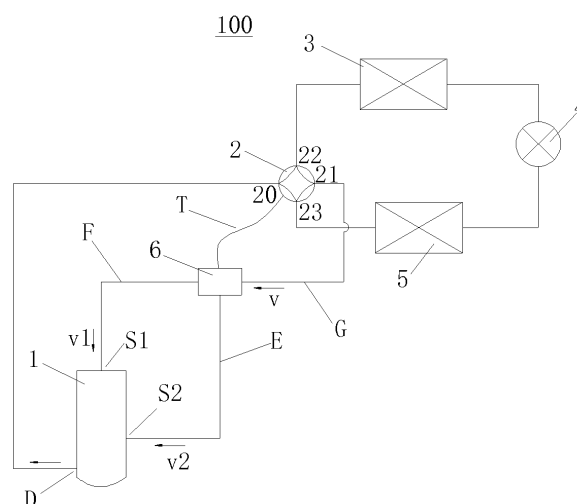


Fig. 9

Description**FIELD**

5 **[0001]** The present invention relates to a field of household appliances, and more particularly to a refrigerating system and a heating system.

BACKGROUND

10 **[0002]** Currently, the motor of a low back pressure rotary compressor is disposed in the shell, and during the operation of the compressor, the motor generates heat due to energy loss, but a low temperature and low pressure environment communicated with the gas suction pipe is present in the shell. Therefore, the sucked low temperature and low pressure gas exchanges heat with the motor, which cools the motor and hence guarantees the reliability of the motor on one hand, but deteriorates the performance of the refrigeration cycle since the sucked gas is heated on the other hand.

15 **[0003]** Regarding the refrigerating system provided with the low back pressure rotary compressor, the refrigerating performance and the heating performance in the refrigeration cycle are, in essence, differently affected by the heating of the sucked gas. Therefore, the refrigerating system should be designed according to characteristics when the refrigerating device is in a refrigerating mode or in a heating mode, so as to achieve the double objectives of improving the performance of the refrigerating system and meeting the cooling requirement of the motor.

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SUMMARY

[0004] The present invention seeks to solve at least one of the problems existing in the prior art.

25 **[0005]** Therefore, an objective of the present invention is to provide a refrigerating system that can achieve the double objectives of optimizing the performance of the refrigerating system and meeting the cooling requirement of the motor.

[0006] Another objective of the present invention is to provide a heating system that can guarantee the reliability of the motor furthest.

30 **[0007]** According to a first aspect of the present invention, a refrigerating system includes: a low back pressure rotary compressor including a shell, and a motor and a compressing mechanism disposed in the shell respectively, an upper gas suction pipe disposed at an upper part of the shell, a middle gas suction pipe disposed at a middle part of the shell, and an exhaust pipe disposed at the shell; a four-way valve being formed with an exhaust valve port, a gas suction valve port, an outdoor heat exchanger valve port and an indoor heat exchanger valve port, and the exhaust valve port being connected with the exhaust pipe; an outdoor heat exchanger defining an end connected with the outdoor heat exchanger valve port; an indoor heat exchanger defining an end connected with the indoor heat exchanger valve port, and the other end connected with the other end of the outdoor heat exchanger; a throttling element connected between the outdoor heat exchanger and the indoor heat exchanger in series; a control valve assembly being connected with the gas suction valve port, the upper gas suction pipe and the middle gas suction pipe via a first pipeline, a second pipeline and a third pipeline respectively, in which when the refrigerating system is in a refrigerating mode, the control valve assembly controls a gas suction flow of the third pipeline to be larger than that of the second pipeline; when the refrigerating system is in a heating mode, the control valve assembly controls the gas suction flow of the second pipeline to be larger than that of the third pipeline.

35 **[0008]** In the refrigerating system according to embodiments of the present invention, when the refrigerating system is in a refrigerating mode, the control valve assembly controls the gas suction flow of the third pipeline to be larger than that of the second pipeline; when the refrigerating system is in a heating mode, the control valve assembly controls the gas suction flow of the second pipeline to be larger than that of the third pipeline. Therefore, when the refrigerating system is in the refrigerating mode, the gas is mainly sucked via the middle gas suction pipe without going through the motor, thereby reducing the heating degree of the sucked gas by the motor and the deterioration of the performance due to heating. Meanwhile, a small amount of gas is sucked via the upper gas suction pipe and cools the motor after going through the motor so as to guarantee the reliability of the motor. When the refrigerating system is in the heating mode, the gas is mainly sucked via the upper gas suction pipe and cools the motor after going through the motor so as to maximize the reliability of the motor.

50 **[0009]** In addition, the refrigerating system according to the present invention also has the additional technical features as follows:

55 Specifically, in the refrigerating mode, a ratio between the gas suction flow of the third pipeline and that of the first pipeline is greater than or equal to 0.6.

[0010] Further, in the refrigerating mode, the gas suction flow of the third pipeline is equal to that of the first pipeline.

[0011] Specifically, in the heating mode, a ratio between the gas suction flow of the second pipeline and that of the first pipeline is greater than or equal to 0.6.

[0012] Further, in the heating mode, the gas suction flow of the second pipeline is equal to that of the first pipeline.

[0013] Specifically, the control valve assembly includes a first control valve and a second control valve disposed at the second pipeline and the third pipeline respectively.

[0014] In some examples of the present invention, the control valve assembly is connected with the four-way valve to determine whether the refrigerating system is in the refrigerating mode or the heating mode in accordance with a flow direction of a refrigerant in the four-way valve.

[0015] In the specific embodiment of the present invention, when an evaporation temperature, a condensation temperature and a supercooling degree are constant, a heating quality or coefficient of performance (COP) of the refrigerant in the low back pressure rotary compressor improves as a superheat degree rises.

[0016] According to a second aspect of the present invention, a refrigerating system includes: a low back pressure rotary compressor including a shell, and a motor and a compressing mechanism disposed in the shell respectively, an upper gas suction pipe disposed at an upper part of the shell, a middle gas suction pipe disposed at a middle part of the shell, and an exhaust pipe disposed at the shell; an outdoor heat exchanger defining an end connected with the exhaust pipe; an indoor heat exchanger defining an end connected with the other end of the outdoor heat exchanger; a throttling element connected between the outdoor heat exchanger and the indoor heat exchanger in series; a control valve assembly being connected with the other end of the indoor heat exchanger, the upper gas suction pipe and the middle gas suction pipe via a first pipeline, a second pipeline and a third pipeline respectively, in which the control valve assembly controls a gas suction flow of the third pipeline to be larger than that of the second pipeline.

[0017] In the refrigerating system according to embodiments of the present invention, the control valve assembly controls the gas suction flow of the third pipeline to be larger than that of the second pipeline, and the gas is mainly sucked via the middle gas suction pipe without going through the motor, thereby reducing the heating degree of the sucked gas by the motor and the deterioration of the performance due to heating. Meanwhile, a small amount of gas is sucked via the upper gas suction pipe and cools the motor after going through the motor so as to guarantee the reliability of the motor, to furthest avoid the deterioration of the performance due to the heating of the sucked gas and to improve the performance of the compressor and the system.

[0018] In addition, the refrigerating system according to the present invention also has the additional technical features as follows:

Specifically, a ratio between the gas suction flow of the third pipeline and that of the first pipeline is greater than or equal to 0.6.

[0019] Further, the gas suction flow of the third pipeline is equal to that of the first pipeline.

[0020] In a specific embodiment of the present invention, when an evaporation temperature, a condensation temperature and a supercooling degree are constant, a heating quality or coefficient of performance of the refrigerant in the low back pressure rotary compressor increases as a superheat degree rises.

[0021] Specifically, the refrigerant can be one of R290, R134a and R410A.

[0022] Optionally, the control valve assembly is a three-way valve.

[0023] According to a third aspect of the present invention, a heating system includes: a low back pressure rotary compressor including a shell, and a motor and a compressing mechanism disposed in the shell respectively, an upper gas suction pipe disposed at an upper part of the shell, a middle gas suction pipe disposed at a middle part of the shell, and an exhaust pipe disposed at the shell; an indoor heat exchanger defining an end connected with the exhaust pipe; an outdoor heat exchanger defining an end connected with the other end of the indoor heat exchanger; a throttling element connected between the indoor heat exchanger and the outdoor heat exchanger in series; a control valve assembly being connected with the outdoor heat exchanger, the upper gas suction pipe and the middle gas suction pipe via a first pipeline, a second pipeline and a third pipeline respectively, in which the control valve assembly controls a gas suction flow of the second pipeline to be larger than that of the third pipeline.

[0024] In the heating system according to embodiments of the present invention, the control valve assembly controls the gas suction flow of the second pipeline F to be larger than that of the third pipeline E, so that the sucked gas goes through the motor to cool the motor so as to maximize the reliability of the motor.

[0025] In addition, the heating system according to the present invention also has the additional technical features as follows:

Specifically, a ratio between the gas suction flow of the second pipeline and that of the first pipeline is greater than or equal to 0.8.

[0026] Further, the gas suction flow of the second pipeline is equal to that of the first pipeline.

[0027] In a specific embodiment of the present invention, the refrigerant in the low back pressure rotary compressor is a mixed refrigerant containing R32 with a mass percent less than or equal to 50%.

[0028] Optionally, the control valve assembly is a three-way valve.

5 [0029] In some embodiments of the present invention, the heating system is a heat pump water heater, and the indoor heat exchanger is disposed in a water tank to heat water in the water tank.

[0030] Additional aspects and advantages of the present invention will be given in part in the following descriptions, become apparent in part from the following descriptions, or be learned from the practice of the embodiments of the present invention.

10 BRIEF DESCRIPTION OF THE DRAWINGS

[0031] These and other aspects and advantages of the present invention will become apparent and more readily appreciated from the following descriptions made with reference to the drawings, in which:

- 15 Fig. 1 is a refrigeration cycle diagram of a refrigerant at different superheat degrees;
 Fig. 2 is a diagram of a change trend of a heating quality of a refrigerant R134a with respect to different superheat degrees;
 Fig. 3 is a diagram of a change trend of a heating coefficient of performance of a refrigerant R134a with respect to different superheat degrees;
 20 Fig. 4 is a diagram of a change trend of a heating quality of a refrigerant R32 with respect to different superheat degrees;
 Fig. 5 is a diagram of a change trend of a heating coefficient of performance of a refrigerant R32 with respect to different superheat degrees;
 Fig. 6 is a diagram of a change trend of a heating quality of a refrigerant R410A containing R32 with respect to different superheat degrees;
 25 Fig. 7 is a diagram of a change trend of a heating coefficient of performance of a refrigerant R410A containing R32 with respect to different superheat degrees;
 Fig. 8 is a schematic view of a low back pressure rotary compressor according to an embodiment of the present invention;
 30 Fig. 9 is a schematic view of a refrigerating system according to an embodiment of the present invention;
 Fig. 10 is a schematic view of a refrigerating system according to another embodiment of the present invention;
 Fig. 11 is a schematic view of a heating system according to an embodiment of the present invention.

Reference numerals:

35

[0032]

- | | |
|-------|-------------------------------------|
| 100 | refrigerating system |
| 200 | heating system |
| 40 1 | low back pressure rotary compressor |
| S1 | upper gas suction pipe |
| S2 | middle gas suction pipe |
| D | exhaust pipe |
| 11 | upper shell |
| 45 12 | main shell |
| 13 | lower shell |
| 2 | four-way valve |
| 20 | exhaust valve port |
| 21 | gas suction valve port |
| 50 22 | outdoor heat exchanger valve port |
| 23 | indoor heat exchanger valve port |
| 3 | outdoor heat exchanger |
| 5 | indoor heat exchanger |
| 4 | throttling element |
| 55 6 | control valve assembly |
| G | first pipeline |
| F | second pipeline |
| E | third pipeline |

DETAILED DESCRIPTION

[0033] Embodiments of the present invention will be described in detail and examples of the embodiments will be illustrated in the drawings, in which same or similar reference numerals are used to indicate same or similar members or members with same or similar functions throughout the specification. The embodiments described herein with reference to drawings are explanatory, which are used to illustrate the present invention, but shall not be construed to limit the present invention.

[0034] In the description of the present invention, it is to be understood that terms such as "central", "upper", "lower", "front", "rear", "left", "right", "vertical", "horizontal", "top", "bottom", "inner", "outer" should be construed to refer to the orientation or position as shown in the drawings under discussion, only for convenience of description and do not indicate or imply that the apparatus or members must have a particular orientation or be constructed and operated in a particular orientation. Therefore, these terms shall not be construed to limit the present invention.

[0035] It shall be noted that terms such as "first" and "second" are used herein for purposes of description and are not intended to indicate or imply relative importance or to imply the number of indicated technical features. Thus, the feature defined with "first" and "second" may explicitly or implicitly include one or more of this feature. Furthermore, in the description of the present invention, "a plurality of" means two or more than two, unless specified otherwise.

[0036] First, the relationship between the performance of the refrigerating system and both the heating of a sucked gas by the motor and the cooling requirement of the motor of the low back pressure rotary compressor will be described below with reference to Fig. 1 to Fig. 7.

[0037] Generally, it is possible to determine the performance change trend of a refrigerant in practical applications based on the theoretical thermodynamic cycle of the refrigerant.

[0038] For example, in the following reference operation conditions, when the evaporation temperature, condensation temperature and supercooling degree are constant, the refrigeration cycle has a performance change trend as the superheat degree rises from 5°C to 35°C. For instance, the performance change trend of the refrigeration cycle is calculated when the superheat degree is 5°C, 15°C, 25°C and 35°C respectively. Supposing the superheat degree is 5°C when the refrigerating system is returning gas, while 15°C, 25°C and 35°C represent that heating of the sucked gas by the motor results in an increase of the temperature of the sucked gas by 10°C, 20°C and 30°C respectively. When the superheat degree is 5°C, the refrigeration cycle is taken as a benchmark, called a benchmark cycle.

[0039] To simplify the illustration, in the following, the total superheat degrees are used to represent different cycle conditions.

[0040] Calculation conditions are presented in Table 1.

Table 1. Calculation conditions in reference operation conditions

Operation Condition	Condensation Temperature (°C)	Evaporation Temperature (°C)	Supercooling Degree (°C)	Superheat Degree (°C)
Reference Operation Condition	60	10	10	5/15/25/35

[0041] In the pressure-enthalpy diagram of the refrigerant, i.e. lg p-h diagram, the compressed vapor refrigeration cycle is shown in Fig. 1, in which the horizontal coordinate denotes enthalpy values and the vertical coordinate denotes pressure values. In Fig. 1, the cycle is 1a-2a-3-4-5-1a at the superheat degree of 5°C, 1b-2b-3-4-5-1b at the superheat degree of 15°C, 1c-2c-3-4-5-1c at the superheat degree of 25°C, and 1d-2d-3-4-5-1d at the superheat degree of 35°C, in which the difference between the temperature at 1a, 1b, 1c, 1d and that at point 1 is the superheat degree.

[0042] In Fig. 1, H represents enthalpy values, and a point code is used as a subscript to represent the enthalpy value at a specific point. For example, H1 represents the enthalpy value at point 1, and H2a represents the enthalpy value at point 2a. In addition, K represents the circulated mass, in which K5 represents the circulated mass of the refrigerant when the superheat degree is 5°C, K15 represents the circulated mass of the refrigerant when the superheat degree is 15°C, and so on.

[0043] The refrigeration cycle at the superheat degree of 25°C and the refrigerating and heating performance trend of the benchmark cycle are taken as examples for comparative analysis:

When the superheat degree is 5°C, the performance of the refrigeration cycle is calculated as follows:

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$$Q_c = (H_{1a} - H_5) * K_5$$

$$Q_h = (H_{2a} - H_4) * K_5$$

$$P = (H_{2a} - H_{1a}) * K_5$$

$$COP_c = Q_c / P$$

$$COP_h = Q_h / P$$

where:

Q_c : cooling quality

Q_h : heating quality

P : compression power

COP_c : cooling coefficient of performance

COP_h : heating coefficient of performance

[0044] When the sucked gas is heated by the motor, the refrigeration cycle at the superheat degree of 25°C is taken as an example, and the performance of the refrigeration cycle is calculated as follows:

$$Q_c = (H_{1a} - H_5) * K_{25}$$

$$Q_h = (H_{2c} - H_4) * K_{25}$$

$$P = (H_{2c} - H_{1c}) * K_{25}$$

$$COP_c = Q_c / P$$

$$COP_h = Q_h / P$$

[0045] First, the refrigeration cycle is analyzed. When the sucked gas is heated by the motor, the specific volume of the gas is increased, but the suction volume of the compressor is constant, so the circulated mass of the refrigerant is decreased, i.e. $K_{25} < K_5$. Because the heated motor generates invalid overheating, the enthalpy difference of the refrigerating system stays the same and is still calculated according to the formula of $(H_{1a} - H_5)$, thereby reducing Q_c . In other words, the heating of the sucked gas will result in a decrease of the cooling quality. Regarding the compression power, with respect to the benchmark cycle, $(H_{2c} - H_{1c})$ is increased, while K_{25} is decreased. Therefore, P cannot be determined, and thus COP_c cannot be determined, either. As a matter of fact, COP_c also has a downward trend for most refrigerants.

[0046] Therefore, gas suction via the middle gas suction pipe S2 is needed to reduce the heating degree of the sucked gas by the motor. However, due to the cooling requirement of the motor, sometimes the upper gas suction pipe S1 is allowed to suck an appropriate amount of low-temperature gas to cool the motor, especially when the gas sucked via the middle gas suction pipe S2 is directly communicated with the gas suction chamber of the compressor without going

through the motor.

[0047] In other words, although the performance may be lost in refrigerating conditions, it is possible that a small amount of gas still needs being sucked via the upper gas suction pipe to cool the motor in some cases like the temperature of the motor being too high or in some structural designs like the middle gas suction pipe S2 being directly communicated with the gas suction chamber of the compressor. However, in refrigerating conditions, the refrigerants should be sucked via the middle gas suction pipe S2 as many as possible to reduce the heating degree of the sucked gas so as to improve the performance of the compressor and the refrigerating system.

[0048] Next, the heating cycle is analyzed. When the sucked gas is heated, the circulated mass of the refrigerant is decreased, i.e. $K_{25} < K_5$, but the enthalpy difference ($H_{2c} - H_4$) which is used to calculate Q_h is increased by ($H_{2c} - H_2$) compared with the benchmark cycle, i.e. ($H_{2c} - H_4$) is increased. Therefore, it is uncertain whether Q_h improves or declines, which is determined based on different refrigerants and practical situations.

[0049] For example, for R134a refrigerant, the change trends of Q_h and COP_h are shown in Fig. 2 and Fig. 3 respectively, in which the horizontal coordinate represents the superheat degree and the vertical ordinate represents the percent of the quality or coefficient of performance. It can be seen that both the heating quality and the heating coefficient of performance of R134a improve as the superheat degree rises.

[0050] For R32 refrigerant, the change trends of Q_h and COP_h are shown in Fig. 4 and Fig. 5 respectively. However, it can be seen that both the heating quality and the heating coefficient of performance of R134a deteriorate as the superheat degree rises.

[0051] In the refrigerating system provided with the low back pressure rotary compressor, due to the inevitable process that the sucked gas is heated by the motor, a suitable refrigerant should be chosen to guarantee the performance of the system; especially when the system operates in a heating condition, a suitable refrigerant can be chosen to take advantage of the feature that the sucked gas is heated in the low back pressure rotary compressor to improve the heating performance of the system. Based on the theoretical calculation method described above, a suitable refrigerant can be chosen so that the heating performance will not deteriorate as the superheat degree increases, such as one of R134a, R290, R410A, R161, HFO-1234yf, HFO-1234ze and so on.

[0052] Therefore, based on the parameters of refrigeration cycles of different refrigerants in reference operation conditions in respective pressure-enthalpy diagrams, the change trends of the heating quality and coefficient of performance can be calculated at different superheat degrees, which can be used to determine whether a refrigerant is suitable to apply in the low back pressure rotary compressor in heating conditions. Moreover, the proportion of the gas sucked via the upper gas suction pipe S1 can be designed to adjust the superheat degree in heating conditions, which can optimize the heating quality or coefficient of performance of the refrigerating system provided with the low back pressure rotary compressor.

[0053] In addition, considering the sensibility of R32 refrigerant to the superheat degree, the selected refrigerant should not contain too much R32 in the refrigerating system provided with the low back pressure rotary compressor. Based on the theoretical calculation results, for instance, when R410A refrigerant contains 50% of R32, the calculation results are shown in Fig. 6 and Fig. 7. It can be seen that the performance nearly reaches a critical state, so if a mixed refrigerant containing R32 is selected to apply in the refrigerating system provided with the low back pressure rotary compressor, the mass percent of R32 should be less than or equal to 50%.

[0054] In the refrigerating system and the heating system, it is possible to adjust the gas suction flow distribution between the upper gas suction pipe S1 and the middle gas suction pipe S2 of the compressor to optimize the heating performance of the refrigerating system and the heating system with different types of refrigerants. For example, when the refrigerating system adopts a refrigerant whose heating performance obviously improves as the superheat degree rises, it is possible to increase the gas inflow of the upper gas suction pipe S1 to improve the heating performance, or even make gas entirely sucked via the upper shell. However, for the refrigerating system that adopts a refrigerant whose heating performance reaches a critical state as the superheat degree rises, it is necessary to take practical situations into consideration. For example, the system is designed in accordance with different requirements, such as giving priority to the heating quality or coefficient of performance (COP), so as to achieve the desired effect of the system performance by adjusting the proportion of the gas sucked via the upper gas suction pipe S1.

[0055] The refrigerating system 100 provided with the low back pressure rotary compressor 1 will be described in detail herein according to two embodiments of the present invention with reference to Fig. 8 to Fig. 10. With the refrigerating system 100, it is possible to achieve the double objectives of optimizing the performance of the refrigerating system 100 and meeting the cooling requirement of the motor based on the above principle.

Embodiment 1:

[0056] As shown in Fig. 8 and Fig. 9, the refrigerating system 100 according to embodiments of the present invention includes: a low back pressure rotary compressor 1, a four-way valve 2, an outdoor heat exchanger 3, an indoor heat exchanger 5, a throttling element 4 and a control valve assembly 6. The refrigerating system 100 has a refrigerating

mode and a heating mode.

[0057] The low back pressure rotary compressor 1 includes a shell, a motor and a compressing mechanism. The shell includes an upper shell 11, a main shell 12 and a lower shell 13. The upper shell 11 is disposed at an upper part of the main shell 12, and the lower shell 13 is disposed at a lower part of the main shell 12. The upper shell 11, the main shell 12 and the lower shell 12 define an inner space of the shell, in which the motor and compressing mechanism are disposed respectively. An upper gas suction pipe S1 is disposed at the upper part of the shell, i.e. the upper shell 11; a middle gas suction pipe S2 is disposed at the middle part of the shell, i.e. the main shell 12; an exhaust pipe D is disposed at the shell, and in the example of Fig. 8, the exhaust pipe D is disposed at the lower part of the main shell 12. The high pressure gas in the low back pressure rotary compressor 1 is exhausted from the exhaust pipe D, and the refrigerant that undergoes the refrigeration cycle or the heating cycle is sucked into the inner space of the shell via the upper gas suction pipe S1 and the middle gas suction pipe S2. It should be understood that the specific structure and operation mechanism of the low back pressure rotary compressor 1 are well-known to those skilled in the art, which will not be described in detail herein.

[0058] The four-way valve 2 is formed with an exhaust valve port 20, a gas suction valve port 21, an outdoor heat exchanger valve port 22 and an indoor heat exchanger valve port 23, in which the exhaust valve port 20 is connected with the exhaust pipe D, the indoor heat exchanger valve port 23 is connected with an end of the indoor heat exchanger 5, and the outdoor heat exchanger valve port 22 is connected with an end of the outdoor heat exchanger 3. When the exhaust valve port 20 of the four-way valve 2 is communicated with the outdoor heat exchanger valve port 22 and the gas suction valve port 21 of the four-way valve 2 is communicated with the indoor heat exchanger valve port 23, the refrigerating system 100 is in the refrigerating mode; when the exhaust valve port 20 of the four-way valve 2 is communicated with the indoor heat exchanger valve port 23 and the gas suction valve port 21 of the four-way valve 2 is communicated with the outdoor heat exchanger valve port 22, the refrigerating system 100 is in the heating mode. Further, an oil separator is disposed between the exhaust valve port 20 and the exhaust pipe D to conduct oil-gas separation of the refrigerant exhausted from the exhaust pipe D. The structure and operation mechanism of the oil separator are well-known to those skilled in the art, which will not be described in detail herein.

[0059] The other end of the indoor heat exchanger 5 is connected with the other end of the outdoor heat exchanger 3. The throttling element 4 is connected between the outdoor heat exchanger 3 and the indoor heat exchanger 5 in series. Optionally, the throttling element 4 is a capillary or an electromagnetic valve.

[0060] The control valve assembly 6 is connected with the gas suction valve port 21, the upper gas suction pipe S1 and the middle gas suction pipe S2 via a first pipeline G, a second pipeline F and a third pipeline E respectively. In other words, the control valve assembly 6 is connected with the gas suction valve port 21 via the first pipeline G, connected with the upper gas suction pipe S1 via the second pipeline F and connected with the middle gas suction pipe S2 via the third pipeline E. The control valve assembly 6 has a function of determining the operation mode (the refrigerating mode or the heating mode) of the refrigerating system 100, and the control valve assembly 6 controls the gas suction flows of the second pipeline F and the third pipeline E based on the operation mode of the refrigerating system 100. In some examples of the present invention, the control valve assembly 6 includes a first control valve and a second control valve disposed at the second pipeline F and the third pipeline E respectively, in which case, the gas suction flows of the second pipeline F and the third pipeline E can be controlled by controlling the size of the openings of the first control valve and the second control valve or the position of the valve core. However, the present invention is not limited thereby. The control valve assembly 6 can achieve the objective of controlling the gas suction flows of the second pipeline F and the third pipeline E by controlling the diameter ratio or the flow area ratio of the second pipeline F and the third pipeline E.

[0061] When the refrigerating system 100 is in the refrigerating mode, the indoor heat exchanger 5 is a low pressure side heat exchanger and the outdoor heat exchanger 3 is a high pressure side heat exchanger. The high temperature and high pressure gas exhausted from the exhaust pipe D flows into the outdoor heat exchanger 3 for condensation and heat exchange by means of the flow direction control of the four-way valve 2, and then flows into the indoor heat exchanger 5 for cooling the indoor environment by means of the throttling function of the throttling element 4. The low temperature and low pressure return gas of the system exhausted from the outlet of the indoor heat exchanger 5 flows to the control valve assembly 6 via the first pipeline G. The control valve assembly 6 controls the gas suction flow of the third pipeline E to be larger than that of the second pipeline F. In other words, the control valve assembly 6 makes the return gas of the system mainly flow to the direction of the middle gas suction pipe S2. Specifically, the total flow of the return gas of the system is the gas suction flow of the first pipeline G, i.e. v , and the flow into the middle gas suction pipe S2 is the gas suction flow of the third pipeline E, i.e. v_2 . The ratio between v_2 and the total flow v is v_3 , and v_3 is greater than or equal to 0.6. In some examples of the present invention, v_2 can be equal to v , i.e. the gas suction flow of the third pipeline E is equal to that of the first pipeline G, in which case the gas suction is entirely conducted via the middle gas suction pipe S2.

[0062] In such a case, the gas is mainly sucked via the middle gas suction pipe S2 without going through the motor, thereby reducing the heating degree of the sucked gas by the motor and the deterioration of the performance due to heating. Meanwhile, a small amount of gas is sucked via the upper gas suction pipe S1, and cools the motor after going

through the motor so as to guarantee the reliability of the motor. In certain use conditions, even though no gas suction is conducted via the upper gas suction pipe S1, the temperature of the motor will not rise to a higher level due to the low-temperature environment. In this case, it is allowable that the gas is entirely sucked via the middle gas suction pipe S2, so as to furthest avoid the deterioration of the performance due to the heating of the sucked gas and to improve the performance of the low back pressure rotary compressor 1 and the refrigerating system 100.

[0063] When the refrigerating system 100 is in the heating mode, the indoor heat exchanger 5 is a high pressure side heat exchanger and the outdoor heat exchanger 3 is a low pressure side heat exchanger. The high temperature and high pressure gas exhausted from the exhaust pipe D flows into the indoor heat exchanger 5 for heating the indoor environment by means of the flow direction control of the four-way valve 2, then flows into the outdoor heat exchanger 3 for heat exchange with the outside air by means of the throttling function of the throttling element 4, and finally the low temperature and low pressure return gas of the system exhausted from the outlet of the outdoor heat exchanger 3 flows to the control valve assembly 6 via the first pipeline G. The control valve assembly 6 controls the gas suction flow of the second pipeline F to be larger than that of the third pipeline E. In other words, the control valve assembly 6 makes the return gas of the system mainly flow to the direction of the upper gas suction pipe S1. Specifically, the total flow of the return gas of the system is the gas suction flow of the first pipeline G, i.e. v , and the flow into the upper gas suction pipe S1 is the gas suction flow of the second pipeline F, i.e. v_1 . The ratio between v_1 and the total flow v is v_4 , and v_4 is greater than or equal to 0.6. In some examples of the present invention, v_1 can be equal to v , i.e. the gas suction flow of the second pipeline F is equal to that of the first pipeline G, in which case the gas suction is entirely conducted via the upper gas suction pipe S1.

[0064] In the refrigerating system 100 according to embodiments of the present invention, when the refrigerating system 100 is in the refrigerating mode, the control valve assembly 6 controls the gas suction flow of the third pipeline E to be larger than that of the second pipeline F; when the refrigerating system is in the heating mode, the control valve assembly 6 controls the gas suction flow of the second pipeline F to be larger than that of the third pipeline E. Therefore, when the refrigerating system 100 is in the refrigerating mode, the gas is mainly sucked via the middle gas suction pipe S2 without going through the motor, thereby reducing the heating degree of the sucked gas by the motor and the deterioration of the performance due to heating. Meanwhile, a small amount of gas is sucked via the upper gas suction pipe S1 and cools the motor after going through the motor so as to guarantee the reliability of the motor. When the refrigerating system 100 is in the heating mode, gas suction is mainly conducted via the upper gas suction pipe S1 and the sucked gas cools the motor after going through the motor so as to maximize the reliability of the motor.

[0065] In the specific embodiment of the present invention, the control valve assembly 6 is connected with the four-way valve 2 to determine whether the refrigerating system 100 is in the refrigerating mode or the heating mode in accordance with the flow direction of the refrigerant in the four-way valve 2. Specifically, as shown in Fig. 9, the control valve assembly 6 monitors the flow direction of the refrigerant in the four-way valve 2 via a channel L. However, the present invention is not limited thereby. The control valve assembly 6 can determine the operation mode of the refrigerating system 100 by other means, such as by means of remote signals from the refrigerating system 100.

[0066] In this embodiment, the refrigerant adopted in the refrigerating system 100 has the following property: when the evaporation temperature, condensation temperature and supercooling degree are constant, the heating quality or coefficient of performance of the refrigerant in the low back pressure rotary compressor 1 improves as the superheat degree rises. For example, when the refrigeration cycle is calculated at the constant evaporation temperature, condensation temperature and supercooling degree, the heating quality or coefficient of performance has an upward trend as the superheat degree rises from 5°C to 35°C. Specifically, the refrigerant is one of R290, R134a and R410A.

Embodiment 2:

[0067] The refrigerating system 100 according to embodiments of the present invention, as shown in Fig. 8 and Fig. 10, includes: a low back pressure rotary compressor 1, an outdoor heat exchanger 3, an indoor heat exchanger 5, a throttling element 4 and a control valve assembly 6. The refrigerating system 100 can only operate in a refrigerating mode. In other words, the refrigerating system 100 is only a cooler.

[0068] The low back pressure rotary compressor 1 includes a shell, and a motor and a compressing mechanism disposed in the shell, an upper gas suction pipe S1 is disposed at an upper part of the shell, a middle gas suction pipe S2 is disposed at a middle part of the shell, and an exhaust pipe D is disposed at the shell. It should be noted that the specific structure and operation mechanism of the low back pressure rotary compressor 1 are well-known to those skilled in the art, which will not be described in detail herein.

[0069] An end of the outdoor heat exchanger 3 is connected with the exhaust pipe D. An end of the indoor heat exchanger 5 is connected with the other end of the outdoor heat exchanger 3. The throttling element 4 is connected between the outdoor heat exchanger 3 and the indoor heat exchanger 5 in series. Optionally, the throttling element 4 is a capillary or an electromagnetic valve.

[0070] The control valve assembly 6 is connected with the other end of the indoor heat exchanger 5, the upper gas

suction pipe S 1 and the middle gas suction pipe S2 via a first pipeline G, a second pipeline F and a third pipeline E respectively. In other words, the control valve assembly 6 is connected with the other end of the indoor heat exchanger 5 via the first pipeline G, connected with the upper gas suction pipe S1 via the second pipeline F and connected with the middle gas suction pipe S2 via the third pipeline E. The control valve assembly 6 controls the gas suction flow of the third pipeline E to be larger than that of the second pipeline F. Specifically, the control valve assembly 6 may be a three-way valve or include valve bodies disposed at the second pipeline F and the third pipeline E respectively, in which case, the gas suction flows of the second pipeline F and the third pipeline E can be controlled by controlling the cross-sectional areas of the second pipeline F and the third pipeline E.

[0071] When the refrigerating system 100 is in operation, the indoor heat exchanger 5 always operates as a low pressure side heat exchanger and the outdoor heat exchanger 3 always operates as a high pressure side heat exchanger. The high temperature and high pressure gas exhausted from the compressor flows into the outdoor heat exchanger 3 for condensation, and then flows into the indoor heat exchanger 5 after passing through the throttling element 4 for evaporation to achieve the purpose of refrigerating. The flow of the return gas of the system flowing from the indoor heat exchanger 5 is v and the gas flows into the control valve assembly 6.

[0072] Because the refrigerating system always works in the refrigerating mode, the return gas of the system is mainly sucked via the middle gas suction pipe S2. Specifically, the ratio between the gas suction flow v_2 of the third pipeline E and the gas suction flow v of the first pipeline G is v_3 , and v_3 is greater than or equal to 0.6. In some examples of the present invention, v_2 can be equal to v , i.e. the gas suction flow of the third pipeline E is equal to that of the first pipeline G, in which case the gas suction is entirely conducted via the middle gas suction pipe S2.

[0073] Therefore, in the refrigerating system 100 according to embodiments of the present invention, the control valve assembly 6 controls the gas suction flow of the third pipeline E to be larger than that of the second pipeline F, and the gas is mainly sucked via the middle gas suction pipe S2 without going through the motor, thereby reducing the heating degree of the sucked gas by the motor and the deterioration of the performance due to heating. Meanwhile, a small amount of gas is sucked via the upper gas suction pipe S1, and cools the motor after going through the motor so as to guarantee the reliability of the motor. In certain use conditions, even though no gas suction is conducted via the upper gas suction pipe S1, the temperature of the motor will not rise to a higher level due to the low-temperature environment. In this case, it is allowable that the gas is entirely sucked via the middle gas suction pipe S2, so as to furthest avoid the deterioration of the performance due to the heating of the sucked gas and to improve the performance of the compressor and the system.

[0074] In this embodiment, the refrigerant adopted in the refrigerating system 100 has the following property: when the evaporation temperature, condensation temperature and supercooling degree are constant, the heating quality or coefficient of performance of the refrigerant in the low back pressure rotary compressor 1 improves as the superheat degree rises. For example, when the refrigeration cycle is calculated at the constant evaporation temperature, condensation temperature and supercooling degree, the heating quality or coefficient of performance has an upward trend as the superheat degree rises from 5°C to 35°C. Specifically, the refrigerant is one of R290, R134a and R410A.

[0075] The heating system 200 provided with the low back pressure rotary compressor 1 according to embodiments of the present invention will be described in detail herein with reference to Fig. 8 and Fig. 11. With the heating system 200, it is possible to optimize the heating quality or coefficient of performance of the heating system 200 in the heating mode based on the above principle.

[0076] As shown in Fig. 8 and Fig. 11, the heating system 200 according to embodiments of the present invention includes a low back pressure rotary compressor 1, an indoor heat exchanger 5, an outdoor heat exchanger 3, a throttling element 4 and a control valve assembly 6. The heating system 200 can only operate in a heating mode. In some specific embodiments of the present invention, the heating system 200 is a heat pump water heater and the indoor heat exchanger 5 is disposed in a water tank 9 to heat water in the water tank 9.

[0077] An end of the indoor heat exchanger 5 is connected with the exhaust pipe D. An end of the outdoor heat exchanger 3 is connected with the other end of the indoor heat exchanger 5. The throttling element 4 is connected between the indoor heat exchanger 5 and the outdoor heat exchanger 3 in series. Optionally, the throttling element 4 is a capillary or an electromagnetic valve.

[0078] The control valve assembly 6 is connected with the outdoor heat exchanger 3, the upper gas suction pipe S 1 and the middle gas suction pipe S2 via a first pipeline G, a second pipeline F and a third pipeline E respectively. In other words, the control valve assembly 6 is connected with the outdoor heat exchanger 3 via the first pipeline G, connected with the upper gas suction pipe S1 via the second pipeline F and connected with the middle gas suction pipe S2 via the third pipeline E. The control valve assembly 6 can control the gas suction flows of the second pipeline F and the third pipeline E, and the control valve assembly 6 controls the gas suction flow of the second pipeline F to be larger than that of the third pipeline E. Specifically, the control valve assembly 6 may be a three-way valve or include valve bodies disposed at the second pipeline F and the third pipeline E respectively, in which case, the gas suction flows of the second pipeline F and the third pipeline E can be controlled by controlling the cross-sectional areas of the second pipeline F and the third pipeline E.

[0079] When the heating system 200 is in operation, the outdoor heat exchanger 3 is a low pressure side heat exchanger and the indoor heat exchanger 5 is a high pressure side heat exchanger. The high temperature and high pressure gas exhausted from the exhaust pipe D flows into the indoor heat exchanger 5 to heat water in the water tank 9 or heat the indoor environment, then flows into the outdoor heat exchanger 3 for heat exchange with the outside air by means of the throttling function of the throttling element 4, and finally the low temperature and low pressure return gas of the system exhausted from the outlet of the outdoor heat exchanger 3 flows to the control valve assembly 6. The control valve assembly 6 controls the gas suction flow of the second pipeline F to be larger than that of the third pipeline E. In other words, the control valve assembly 6 makes the return gas of the system mainly flow to the direction of the upper gas suction pipe S1. Specifically, the total flow of the return gas of the system is v , and the flow into the upper gas suction pipe S1 is v_1 , i.e. the gas suction flow of the second pipeline F. The ratio between v_1 and the total flow v is v_4 , and v_4 is greater than or equal to 0.8. In some examples of the present invention, v_1 can be equal to v , i.e. the gas suction flow of the second pipeline F is equal to that of the first pipeline G, in which case the gas suction is entirely conducted via the upper gas suction pipe S1.

[0080] In the heating system 200 according to embodiments of the present invention, the control valve assembly 6 controls the gas suction flow of the second pipeline F to be larger than that of the third pipeline E, so that the sucked gas goes through the motor to cool the motor so as to maximize the reliability of the motor.

[0081] In this embodiment, the refrigerant in the low back pressure rotary compressor 1 is a mixed refrigerant containing R32 with a mass percent less than or equal to 50%.

[0082] Reference throughout this specification to "an embodiment," "some embodiments," "exemplary embodiments," "examples," "specific examples," or "some examples," means that a particular feature, structure, material, or characteristic described in connection with the embodiment or example is included in at least one embodiment or example of the present invention. Thus, these terms throughout this specification do not necessarily refer to the same embodiment or example of the present invention. Furthermore, the particular features, structures, materials, or characteristics may be combined in any suitable manner in one or more embodiments or examples.

[0083] Although embodiments of the present invention have been shown and described, it would be appreciated by those skilled in the art that the above embodiments cannot be construed to limit the present invention, and changes, alternatives, and modifications can be made in the embodiments without departing from spirit, principles and scope of the present invention. The scope of the present invention is defined by the claims and the equivalents thereof.

Claims

1. A refrigerating system comprising:

a low back pressure rotary compressor comprising a shell, and a motor and a compressing mechanism disposed in the shell respectively, an upper gas suction pipe being disposed at an upper part of the shell, a middle gas suction pipe being disposed at a middle part of the shell, and an exhaust pipe being disposed at the shell;
a four-way valve formed with an exhaust valve port, a gas suction valve port, an outdoor heat exchanger valve port and an indoor heat exchanger valve port, the exhaust valve port being connected with the exhaust pipe;
an outdoor heat exchanger defining an end connected with the outdoor heat exchanger valve port;
an indoor heat exchanger defining an end connected with the indoor heat exchanger valve port, and the other end connected with the other end of the outdoor heat exchanger;
a throttling element connected between the outdoor heat exchanger and the indoor heat exchanger in series;
a control valve assembly connected with the gas suction valve port, the upper gas suction pipe and the middle gas suction pipe via a first pipeline, a second pipeline and a third pipeline respectively, wherein when the refrigerating system is in a refrigerating mode, the control valve assembly controls a gas suction flow of the third pipeline to be larger than that of the second pipeline; when the refrigerating system is in a heating mode, the control valve assembly controls the gas suction flow of the second pipeline to be larger than that of the third pipeline.

2. The refrigerating system according to claim 1, wherein in the refrigerating mode, a ratio between the gas suction flow of the third pipeline and that of the first pipeline is greater than or equal to 0.6.

3. The refrigerating system according to claim 2, wherein in the refrigerating mode, the gas suction flow of the third pipeline is equal to that of the first pipeline.

4. The refrigerating system according to claim 1, wherein in the heating mode, a ratio between the gas suction flow of the second pipeline and that of the first pipeline is greater than or equal to 0.6.

5. The refrigerating system according to claim 4, wherein in the heating mode, the gas suction flow of the second pipeline is equal to that of the first pipeline.

6. The refrigerating system according to claim 1, wherein the control valve assembly comprises a first control valve and a second control valve disposed at the second pipeline and the third pipeline respectively.

7. The refrigerating system according to claim 1, wherein the control valve assembly is connected with the four-way valve to determine whether the refrigerating system is in the refrigerating mode or the heating mode in accordance with a flow direction of a refrigerant in the four-way valve.

8. The refrigerating system according to claim 1, wherein when an evaporation temperature, a condensation temperature and a supercooling degree are constant, a heating quality or coefficient of performance of a refrigerant in the low back pressure rotary compressor increases as a superheat degree rises.

9. A refrigerating system comprising:

a low back pressure rotary compressor comprising a shell, and a motor and a compressing mechanism disposed in the shell respectively, an upper gas suction pipe being disposed at an upper part of the shell, a middle gas suction pipe being disposed at a middle part of the shell, and an exhaust pipe being disposed at the shell;
an outdoor heat exchanger defining an end connected with the exhaust pipe;
an indoor heat exchanger defining an end connected with the other end of the outdoor heat exchanger;
a throttling element connected between the outdoor heat exchanger and the indoor heat exchanger in series;
a control valve assembly connected with the other end of the indoor heat exchanger, the upper gas suction pipe and the middle gas suction pipe via a first pipeline, a second pipeline and a third pipeline respectively, wherein the control valve assembly controls a gas suction flow of the third pipeline to be larger than that of the second pipeline.

10. The refrigerating system according to claim 9, wherein a ratio between the gas suction flow of the third pipeline and that of the first pipeline is greater than or equal to 0.6.

11. The refrigerating system according to claim 10, wherein the gas suction flow of the third pipeline is equal to that of the first pipeline.

12. The refrigerating system according to claim 9, wherein when an evaporation temperature, a condensation temperature and a supercooling degree are constant, a heating quality or coefficient of performance of a refrigerant in the low back pressure rotary compressor increases as a superheat degree rises.

13. The refrigerating system according to claim 12, wherein the refrigerant is one of R290, R134a and R410A.

14. The refrigerating system according to claim 9, wherein the control valve assembly is a three-way valve.

15. A heating system comprising:

a low back pressure rotary compressor comprising a shell, and a motor and a compressing mechanism disposed in the shell respectively, an upper gas suction pipe being disposed at an upper part of the shell, a middle gas suction pipe being disposed at a middle part of the shell, and an exhaust pipe being disposed at the shell;
an indoor heat exchanger defining an end connected with the exhaust pipe;
an outdoor heat exchanger defining an end connected with the other end of the indoor heat exchanger;
a throttling element connected between the indoor heat exchanger and the outdoor heat exchanger in series;
a control valve assembly connected with the outdoor heat exchanger, the upper gas suction pipe and the middle gas suction pipe via a first pipeline, a second pipeline and a third pipeline respectively, wherein the control valve assembly controls a gas suction flow of the second pipeline to be larger than that of the third pipeline.

16. The heating system according to claim 15, wherein a ratio between the gas suction flow of the second pipeline and that of the first pipeline is greater than or equal to 0.8.

17. The heating system according to claim 16, wherein the gas suction flow of the second pipeline is equal to that of the first pipeline.

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18. The heating system according to claim 15, wherein a refrigerant in the low back pressure rotary compressor is a mixed refrigerant containing R32 with a mass percent less than or equal to 50%.

19. The heating system according to claim 15, wherein the control valve assembly is a three-way valve.

20. The heating system according to claim 15, wherein the heating system is a heat pump water heater, and the indoor heat exchanger is disposed in a water tank to heat water in the water tank.

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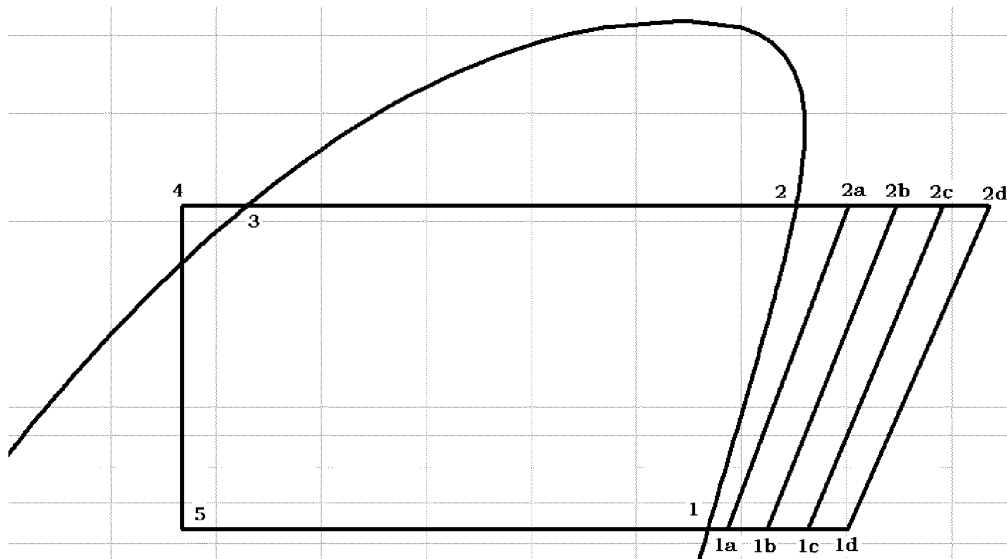


Fig. 1

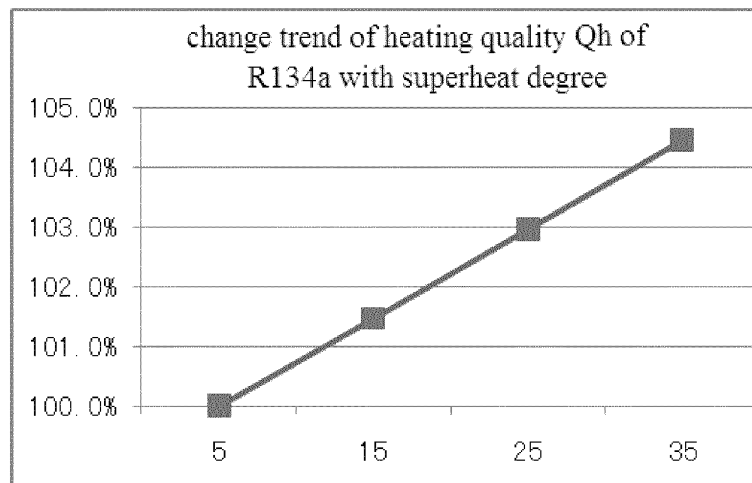


Fig. 2

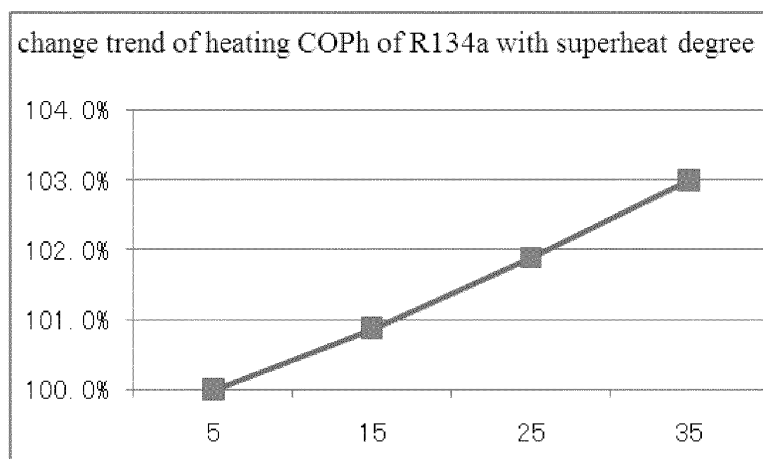


Fig. 3

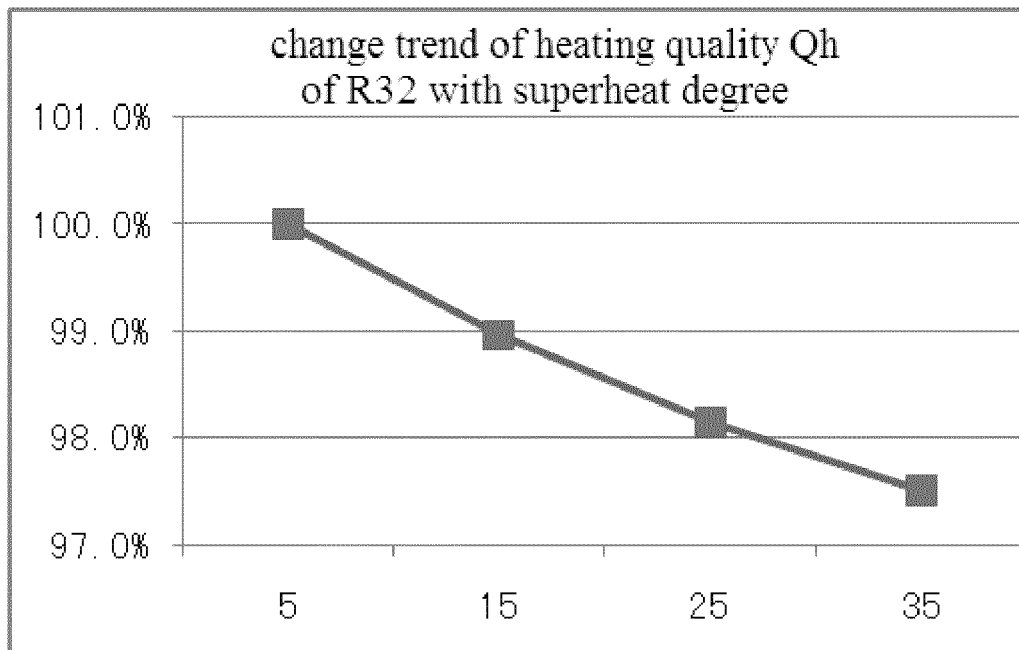


Fig. 4

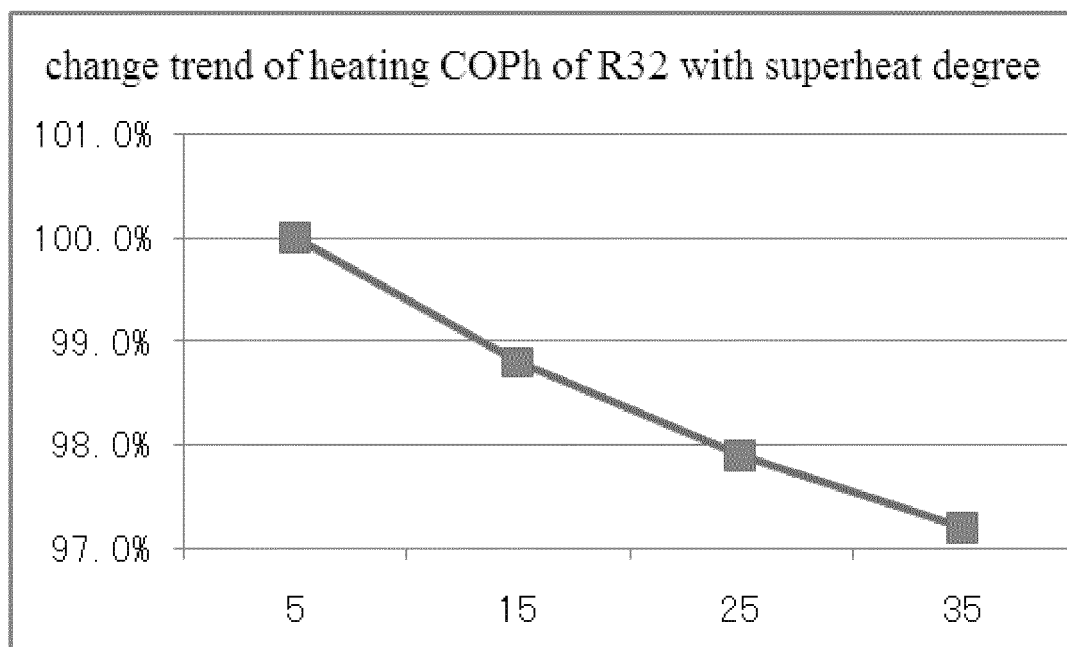


Fig. 5

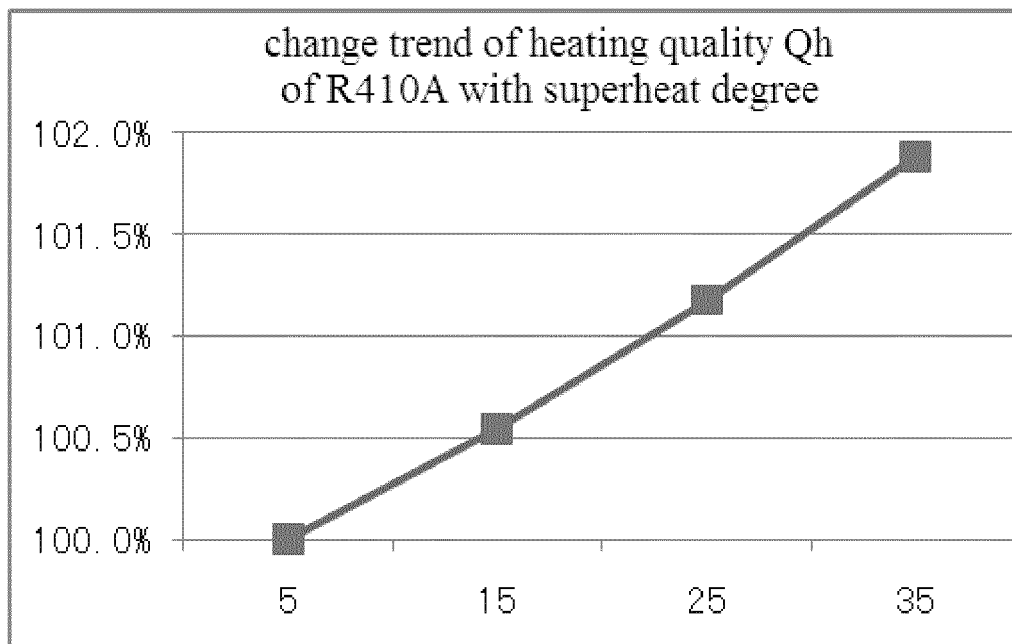


Fig. 6

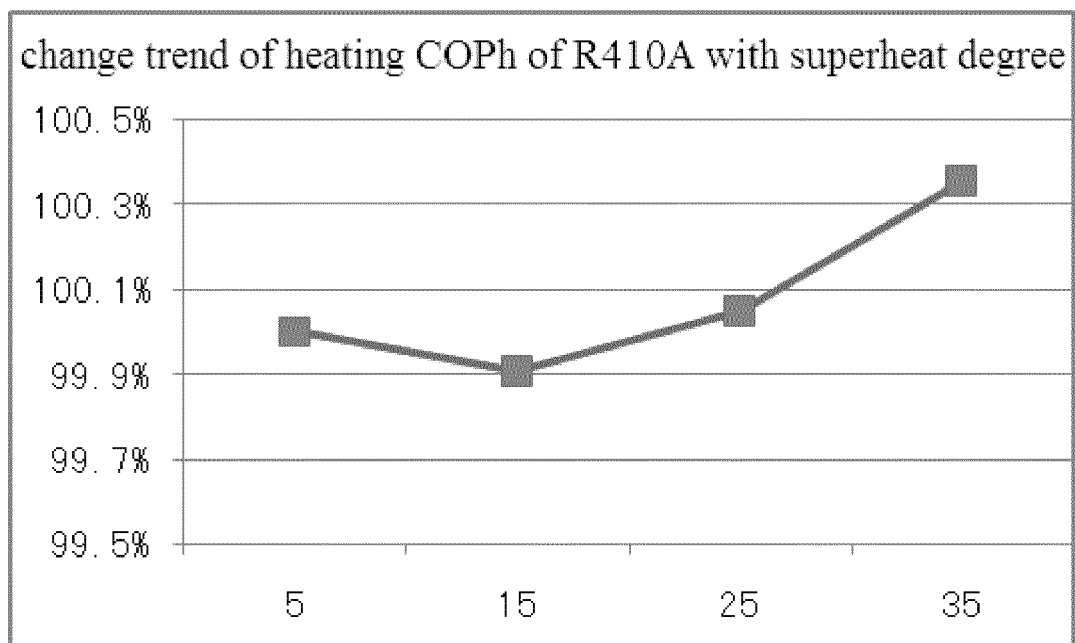


Fig. 7

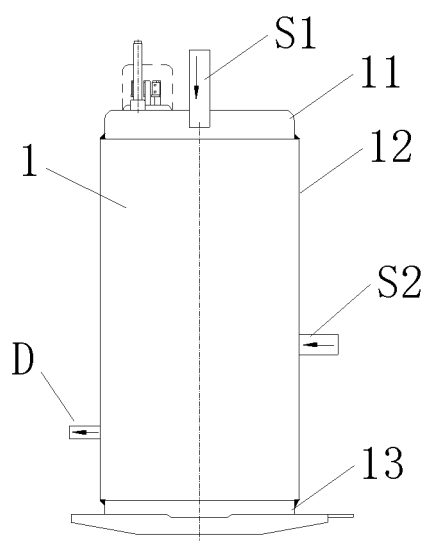


Fig. 8

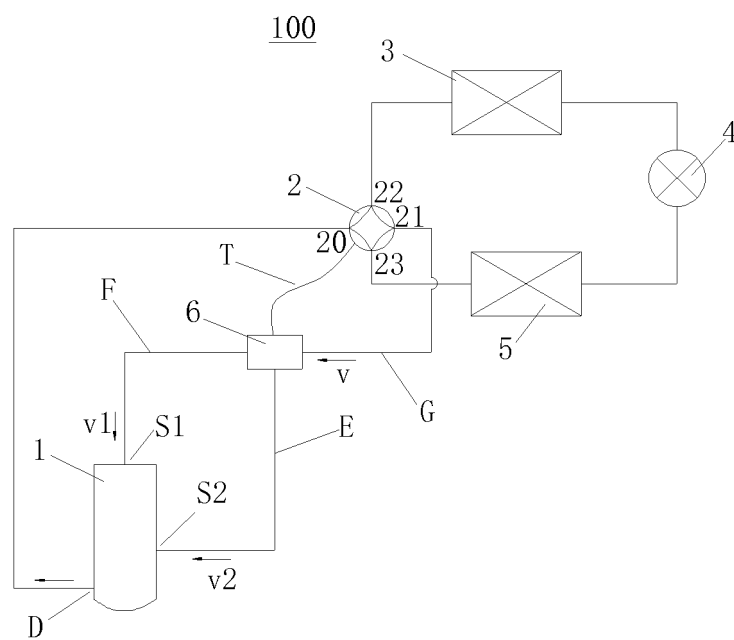


Fig. 9

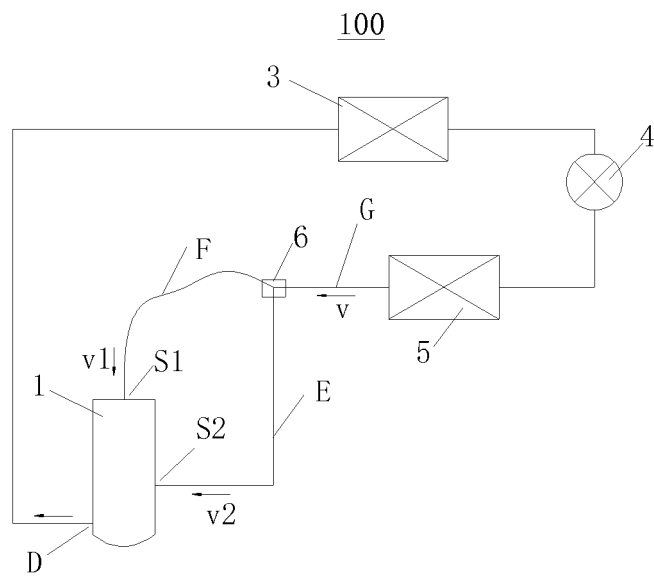


Fig. 10

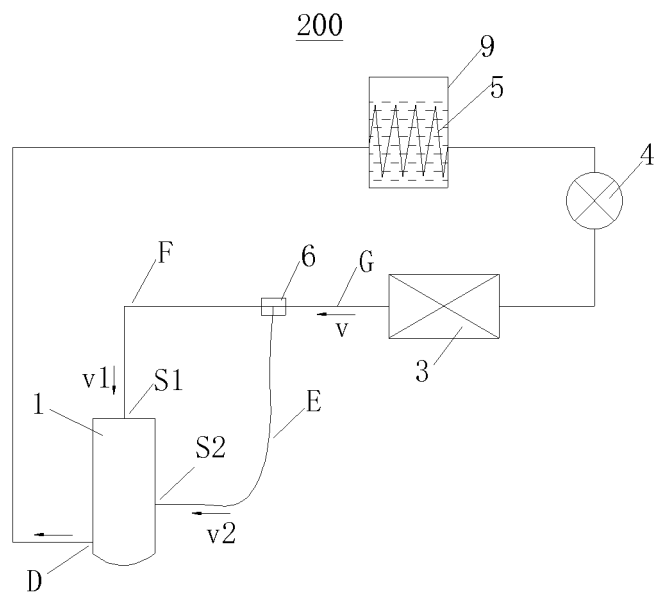


Fig. 11

INTERNATIONAL SEARCH REPORT

International application No.

PCT/CN2013/084749

A. CLASSIFICATION OF SUBJECT MATTER

F25B 41/04 (2006.01) i; F04C 29/04 (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)

F25B; F04C

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

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

CNKI; CNPAT; CNABS; CNTXT; EPODOC; WPI: refrigeration, low back pressure, four-way, valve, air suction, middle, precool, cool+, heat+, four way valve, compressor, suction, middle, flow, motor

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	CN 102678572 A (GREE ELECTRIC APPLIANCES, INC. OF ZHUHAI), 19 September 2012 (19.09.2012), description, paragraphs [0003]-[0027], and figures 1-2	9-11, 14
Y	CN 102678572 A (GREE ELECTRIC APPLIANCES, INC. OF ZHUHAI), 19 September 2012 (19.09.2012), description, paragraphs [0003]-[0027], and figures 1-2	12-13
Y	CN 101158352 A (MIDEA GROUP CO., LTD.), 09 April 2008 (09.04.2008), description, page 4, penultimate paragraph to page 8, paragraph 3, and figures 1-2	12-13
E	CN 103557646 A (GUANGDONG MEIZHI COMPRESSOR CO., LTD.), 05 February 2014 (05.02.2014), claims 1-20	1-20
A	CN 202393113 U (GUANGDONG MEIZHI COMPRESSOR CO., LTD.), 22 August 2012 (22.08.2012), the whole document	1-20
A	CN 202182584 U (DALIAN SANYO COMPRESSOR CO., LTD.), 04 April 2012 (04.04.2012), the whole document	1-20
A	US 2012/0055185 A1 (LUO, R. et al.), 08 March 2012 (08.03.2012), the whole document	1-20

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

* Special categories of cited documents:	"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
"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	"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
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"O" document referring to an oral disclosure, use, exhibition or other means	"&" document member of the same patent family
"P" document published prior to the international filing date but later than the priority date claimed	

Date of the actual completion of the international search
27 May 2014 (27.05.2014)Date of mailing of the international search report
27 June 2014 (27.06.2014)Name and mailing address of the ISA/CN:
State Intellectual Property Office of the P. R. China
No. 6, Xitucheng Road, Jimenqiao
Haidian District, Beijing 100088, China
Facsimile No.: (86-10) 62019451

Authorized officer

WAN, ShanshanTelephone No.: (86-10) **62414199**

INTERNATIONAL SEARCH REPORT

International application No.

PCT/CN2013/084749

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	JP 8-247060 A (TOSHIBA CORP. et al.), 24 September 1996 (24.09.1996), the whole document	1-20

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

INTERNATIONAL SEARCH REPORT
Information on patent family members

International application No.

PCT/CN2013/084749

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CN 102678572 A	19 September 2012	None	
CN 101158352 A	09 April 2008	CN 100529406 C	19 August 2009
		WO 2009059488 A1	14 May 2009
CN 103557646 A	05 February 2014	None	
CN 202393113 U	22 August 2012	None	
CN 202182584 U	04 April 2012	None	
US 2012/0055185 A1	08 March 2012	None	
JP 8-247060 A	24 September 1996	None	

Form PCT/ISA/210 (patent family annex) (July 2009)