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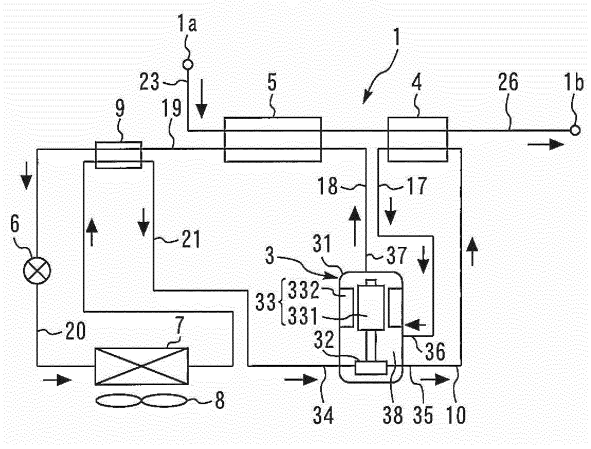
(54) **HEAT PUMP DEVICE**

(57) The present invention has an object to improve a COP of a heat pump device including a compressor having a first discharge passage and a second discharge passage, a mass flow rate of refrigerator oil discharged together with refrigerant from the first discharge passage being higher than a mass flow rate of the refrigerator oil discharged together with the refrigerant from the second discharge passage. The heat pump device according to the present invention includes: the compressor in which the mass flow rate of the refrigerator oil discharged from the first discharge passage is higher than the mass flow rate of the refrigerator oil discharged from the second discharge passage; a first heat exchanger including a first refrigerant heat transfer channel through which the refrigerant and the refrigerator oil discharged from the first discharge passage pass, and a first liquid heat transfer channel through which a liquid passes; and a second heat exchanger including a second refrigerant heat transfer channel through which the refrigerant and the refrigerator oil discharged from the second discharge passage pass, and a second liquid heat transfer channel through which the liquid passes, a total sectional area of the first refrigerant heat transfer channel being larger than a total sectional area of the second refrigerant heat transfer channel.

FIG. 1



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Description

Technical Field

[0001] The present invention relates to a heat pump device.

Background Art

[0002] Patent Literature 1 discloses a hot-water supply cycle device including: a gas cooler having high temperature side refrigerant piping, low temperature side refrigerant piping, and water piping; and a hot-water supply compressor having a sealed container, a compressing element, an electric actuating element, an intake pipe, a discharge pipe, a refrigerant reintroduction pipe, and a refrigerant redischarge pipe. In this device, the intake pipe guides low pressure refrigerant directly to the compressing element, the discharge pipe discharges high pressure refrigerant compressed by the compressing element directly to an outside of the sealed container without releasing the high pressure refrigerant into the sealed container, the refrigerant reintroduction pipe guides the refrigerant resulting from the high pressure refrigerant having passed through the high temperature side refrigerant piping and been subjected to heat exchange into the sealed container, and the refrigerant redischarge pipe redischarges the refrigerant having passed through the electric actuating element in the sealed container to the outside of the sealed container, and feeds the refrigerant to the low temperature side refrigerant piping.

Citation List

Patent Literature

[0003]

Patent Literature 1: Japanese Patent Laid-Open No. 2006-132427
Patent Literature 2: Japanese Patent Laid-Open No. 2004-108616
Patent Literature 3: Japanese Patent Laid-Open No. 2008-309361
Patent Literature 4: Japanese Patent Laid-Open No. 2009-168383

Summary of Invention

Technical Problem

[0004] In the conventional device described above, refrigerator oil is supplied into a compression chamber of the compressing element in order to lubricate and seal a slide portion and reduce friction and gap leakage. Thus, a large amount of refrigerator oil together with a compressed refrigerant gas is discharged from the discharge pipe of the compressor out of the compressor, and is circulated to the high temperature side refrigerant piping. On the other hand, the refrigerant discharged from the refrigerant redischarge pipe of the compressor contains a significantly smaller amount of refrigerator oil than that discharged from the discharge pipe.

[0005] The refrigerator oil has a much higher viscosity than the refrigerant. Thus, in the conventional device described above, the large amount of refrigerator oil together with the refrigerant is circulated to the high temperature side refrigerant piping, thereby increasing pressure loss of the refrigerant. This increases discharge pressure of the compressor and increases input power for the compressor, thereby reducing a coefficient of performance (COP).

[0006] The present invention is achieved to solve the above described problems, and has an object to improve a COP of a heat pump device including a compressor having a first discharge passage and a second discharge passage, a mass flow rate of refrigerator oil discharged together with a refrigerant from the first discharge passage being higher than a mass flow rate of refrigerator oil discharged together with a refrigerant from the second discharge passage.

Solution to Problem

[0007] A heat pump device of the invention includes: a compressor including a first discharge passage for discharging refrigerant and refrigerator oil, and a second discharge passage for discharging the refrigerant and the refrigerator oil, a mass flow rate of the refrigerator oil discharged from the first discharge passage being higher than a mass flow rate of the refrigerator oil discharged from the second discharge passage; a first heat exchanger including one or a plurality of first refrigerant heat transfer channels through which the refrigerant and the refrigerator oil discharged from the first discharge passage pass, and one or a plurality of first liquid heat transfer channels through which a liquid passes, heat exchange being performed between the first refrigerant heat transfer channel and the first liquid heat transfer channel; and a second heat exchanger including one or a plurality of second refrigerant heat transfer channels through which the refrigerant and the refrigerator oil discharged from the second discharge passage pass, and one or a plurality of second liquid heat transfer channels through which the liquid passes, heat exchange being performed between the second refrigerant heat transfer channel and the second liquid heat transfer channel. A total sectional area of the first refrigerant heat transfer channel(s) is larger than a total sectional area of the second refrigerant heat transfer channel(s).

Advantageous Effects of Invention

[0008] The heat pump device according to the present invention can reliably reduce pressure loss of the refrigerant in the first heat exchanger to which the refrigerant and the refrigerator oil are circulated, the refrigerant and the refrigerator oil being discharged from the first discharge passage with a large discharge amount of refrigerator oil. This can reduce input power for the compressor, and improve a COP.

Brief Description of Drawings

[0009]

[Figure 1] Figure 1 is a configuration diagram of a heat pump device according to Embodiment 1 of the present invention.
[Figure 2] Figure 2 is a configuration diagram of a storage type hot-water supply system including the heat pump device in Figure 1.
[Figure 3] Figure 3 is a perspective view of essential portions of a first gas cooler included in the heat pump device in Embodiment 1 of the present invention.
[Figure 4] Figure 4 is a sectional view of essential portions of the first gas cooler included in the heat pump device in Embodiment 1 of the present invention.
[Figure 5] Figure 5 is an enlarged sectional view of essential portions of the first gas cooler and a second gas cooler included in the heat pump device in Embodiment 1 of the present invention.
[Figure 6] Figure 6 shows temperature changes of refrigerant and water in the first gas cooler and the second gas cooler as a whole, and a split position between the first gas cooler and the second gas cooler.
[Figure 7] Figure 7 shows density change of the refrigerant in the first gas cooler and the second gas cooler as a whole.
[Figure 8] Figure 8 shows ratios of refrigerant pressure losses of the first gas cooler and the second gas cooler in a case where the first gas cooler and the second gas cooler have the same shape other than their channel lengths.
[Figure 9] Figure 9 is a configuration diagram of a conventional heat pump device.
[Figure 10] Figure 10 shows a relationship between a ratio of a twist pitch p to an inner diameter SRi of a first twist pipe and a heat transfer coefficient on water side.
[Figure 11] Figure 11 shows a relationship between the ratio of the twist pitch p to the inner diameter SRi of the first twist pipe and a required length of the first twist pipe.
[Figure 12] Figure 12 shows a relationship between the ratio of the twist pitch p to the inner diameter SRi of the first twist pipe and a required length of a first refrigerant heat transfer pipe.
[Figure 13] Figure 13 shows a relationship among a refrigerant pressure loss of the first gas cooler, the ratio of the twist pitch p to the inner diameter SRi of the first twist pipe, and an inner diameter di1 of the first refrigerant heat transfer pipe.
[Figure 14] Figure 14 shows a relationship between the ratio of the twist pitch p to the inner diameter SRi of the first twist pipe in the first gas cooler and a length of the first twist pipe in each of cases in Figure 13.
[Figure 15] Figure 15 shows change in the refrigerant pressure loss of the first gas cooler when an inner diameter ratio di1/di2 of the first refrigerant heat transfer pipe and a second refrigerant heat transfer pipe is changed at p/SRi of 1.8 of the first twist pipe.
[Figure 16] Figure 16 shows change in heat transfer coefficient on water side in a case where the twist pitch p of the first twist pipe is equal to a twist pitch p2 of the second twist pipe and inner diameters SRi of the first twist pipe and the second twist pipe are equal.

Description of Embodiment

[0010] Now, with reference to the drawings, embodiment of the present invention will be described. Throughout the drawings, common components are denoted by the same reference numerals and overlapping descriptions will be omitted. In the description below, a channel length is sometimes simply referred to as "length" for simplicity.

Embodiment 1

[0011] Figure 1 is a configuration diagram of a heat pump device according to Embodiment 1 of the present invention. Figure 2 is a configuration diagram of a storage type hot-water supply system including the heat pump device in Figure 1. As shown in Figure 1, the heat pump device 1 according to Embodiment 1 includes a refrigerant circuit including a compressor 3, a first gas cooler 4 as a first heat exchanger, a second gas cooler 5 as a second heat exchanger, an expansion valve 6 as expansion means, and an evaporator 7, connected by refrigerant piping. The first gas cooler 4 includes a first refrigerant heat transfer channel and a first liquid heat transfer channel, and performs heat exchange between the first refrigerant heat transfer channel and the first liquid heat transfer channel. The second gas cooler 5 includes a second refrigerant heat transfer channel and a second liquid heat transfer channel, and performs heat exchange between the second refrigerant heat transfer channel and the second liquid heat transfer channel. The heat pump device 1 causes a liquid to be a heat medium or an object to be heated to flow through the first liquid heat transfer channel in the first gas cooler 4 and the second liquid heat transfer channel in the second gas cooler 5, and heats the liquid. In the heat pump device according to Embodiment 1, the liquid to be heated is water. The evaporator 7 in Embodiment 1 is constituted by an air-refrigerant heat exchanger for performing heat exchange between air and refrigerant. The heat pump device 1 according to Embodiment 1 further includes a fan 8 for blowing air to the evaporator 7, and a high and low pressure heat exchanger 9 for performing heat exchange between high pressure refrigerant and low pressure refrigerant. During heating operation for heating water, the heat pump device 1 actuates the compressor 3 to operate a heat pump cycle (refrigeration cycle).

[0012] As shown in Figure 2, the heat pump device 1 according to Embodiment 1 may be combined with the tank unit 2 and used as a storage type hot-water supply system. In the tank unit 2, a hot water storage tank 2a for storing hot water and water, and a water pump 2b are provided. The heat pump device 1 and the tank unit 2 are connected by a pipe 11 and a pipe 12 through which water flows, and electric wires (not shown). One end of the pipe 11 is connected to a water inlet 1a of the heat pump device 1. The other end of the pipe 11 is connected to a lower portion of the hot water storage tank 2a in the tank unit 2. The water pump 2b is provided in a middle of the pipe 11 in the tank unit 2. One end of the pipe 12 is connected to a water outlet 1b of the heat pump device 1. The other end of the pipe 12 is connected to an upper portion of the hot water storage tank 2a in the tank unit 2. Instead of the shown configuration, the water pump 2b may be placed in the heat pump device 1.

[0013] As shown in Figure 1, the compressor 3 in the heat pump device 1 includes a sealed container 31, a compressing element 32 and an electric actuating element 33 provided in the sealed container 31, a first intake passage 34, a first discharge passage 35, a second intake passage 36, and a second discharge passage 37. Low pressure refrigerant sucked from the first intake passage 34 directly flows into the compressing element 32 without being released into an internal space 38 of the sealed container 31. The compressing element 32 is driven by the electric actuating element 33, and compresses the low pressure refrigerant into high pressure refrigerant. The high pressure refrigerant compressed by the compressing element 32 is discharged through the first discharge passage 35 directly out of the sealed container 31 without being released into the internal space 38 of the sealed container 31. The high pressure refrigerant discharged from the first discharge passage 35 flows through a pipe 10 into the first gas cooler 4. The high pressure refrigerant having passed through the first gas cooler 4 flows through a pipe 17 to the second intake passage 36 of the compressor 3. The high pressure refrigerant sucked from the second intake passage 36 into the compressor 3 is released into the internal space 38 of the sealed container 31. In Embodiment 1, the compressing element 32 is placed below the electric actuating element 33. An outlet of the second intake passage 36 opens into the internal space 38 of the sealed container 31 at a height between the electric actuating element 33 and the compressing element 32. An inlet of the second discharge passage 37 opens into the internal space 38 of the sealed container 31 at a height above the electric actuating element 33. The high pressure refrigerant released from the outlet of the second intake passage 36 into the internal space 38 of the sealed container 31 passes through a gap or the like between a rotor 331 and a stator 332 of the electric actuating element 33 to a top of the electric actuating element 33, and is discharged through the second discharge passage 37 out of the sealed container 31. The high pressure refrigerant discharged from the second discharge passage 37 flows through a pipe 18 into the second gas cooler 5. The high pressure refrigerant having passed through the second gas cooler 5 passes through a pipe 19 to the expansion valve 6. The high pressure refrigerant passes through the expansion valve 6 to turn into low pressure refrigerant. The low pressure refrigerant flows through a pipe 20 into the evaporator 7. The low pressure refrigerant having passed through the evaporator 7 flows through a pipe 21 to the first intake passage 34 of the compressor 3, and is sucked into the compressor 3. The high and low pressure heat exchanger 9 performs heat exchange between the high pressure refrigerant passing through the pipe 19 and the low pressure refrigerant passing through the pipe 21. The high pressure refrigerant discharged from the first discharge passage 35 is reduced in pressure due to pressure loss while returning through the first gas cooler 4 to the second intake passage 36. Thus, pressure PH2 of the high pressure refrigerant in the internal space 38 of the sealed container 31 is lower than pressure PH1 of the high pressure refrigerant discharged from the first discharge passage 35. Specifically, the discharge pressure PH1 of the first discharge passage 35 is higher than the discharge pressure PH2 of the second discharge passage 37.

[0014] The heat pump device 1 further includes a water channel 23 for guiding water having flowed in from the water inlet 1a to a water inlet of the second gas cooler 5, and a water channel 26 for guiding water (hot water) having flowed out of a water outlet of the first gas cooler 4 to the water outlet 1b. A water outlet of the second gas cooler 5 is connected to a water inlet of the first gas cooler 4. During heating operation, water having flowed in from the water inlet 1a flows through the water channel 23 into the second gas cooler 5, and is heated by heat from the refrigerant in the second gas cooler 5. Hot water generated by heating in the second gas cooler 5 flows into the first gas cooler 4, and is further heated by heat from the refrigerant in the first gas cooler 4. The hot water further increased in temperature by being further heated in the first gas cooler 4 passes through the water channel 26 to the hot water outlet 1b, and is fed through the pipe 12 to the tank unit 2.

[0015] The refrigerant may be refrigerant making it possible to supply high temperature hot-water such as, for example, carbon dioxide, R410A, propane, or propylene, but not limited to them.

[0016] The high temperature and high pressure refrigerant gas discharged from the first discharge passage 35 of the compressor 3 releases heat and is reduced in temperature while passing through the first gas cooler 4. In Embodiment 1, the refrigerant reduced in temperature while passing through the first gas cooler 4 is sucked from the second intake passage 36 into the internal space 38 of the sealed container 31 to cool the electric actuating element 33. Thus, a temperature of the electric actuating element 33 and a surface temperature of the sealed container 31 can be reduced. This can increase motor efficiency of the electric actuating element 33, and reduce heat dissipation loss from a surface of the sealed container 31. The refrigerant gas sucked into the internal space 38 of the sealed container 31 draws heat from the electric actuating element 33 and is increased in temperature. The refrigerant gas is then discharged from the second discharge passage 37 and flows into the second gas cooler 5, and releases heat and is reduced in temperature while passing through the second gas cooler 5. The high pressure refrigerant reduced in temperature heats the low pressure refrigerant while passing through the high and low pressure heat exchanger 9, and then passes through the expansion valve 6. The refrigerant passes through the expansion valve 6, and is thus reduced in pressure into a low pressure gas-liquid two-phase state. The refrigerant having passed through the expansion valve 6 absorbs heat from outside air while passing through the evaporator 7, and is evaporated and gasified. The low pressure refrigerant coming out of the evaporator 7 is heated by the high and low pressure heat exchanger 9, and then sucked from the first intake passage 34 into the compressor 3.

[0017] If the high pressure refrigerant pressure is not less than critical pressure, the refrigerant in the first gas cooler 4 and the second gas cooler 5 is reduced in temperature and releases heat still in a supercritical state without gas-liquid phase transition. If the high pressure refrigerant pressure is not more than the critical pressure, the refrigerant is liquefied and releases heat. In Embodiment 1, carbon dioxide or the like is preferably used as the refrigerant to bring the high pressure refrigerant pressure to the critical pressure or more. If the high pressure refrigerant pressure is not less than the critical pressure, the liquefied refrigerant can be reliably prevented from flowing from the second intake passage 36 into the internal space 38 of the sealed container 31. This can reliably prevent the liquefied refrigerant from adhering to the electric actuating element 33, and reduce rotational resistance of the electric actuating element 33. In addition, the liquefied refrigerant does not flow from the second intake passage 36 into the internal space 38 of the sealed container 31, thereby preventing the refrigerator oil from being diluted by the refrigerant.

[0018] As shown in Figure 2, a water supply pipe 13 is further connected to a lower portion of the hot water storage tank 2a of the tank unit 2. Water supplied from an external water source such as a water supply flows through the water supply pipe 13 into the hot water storage tank 2a and is stored. The hot water storage tank 2a is always filled with water flowing from the water supply pipe 13. A hot-water supplying mixing valve 2c is further provided in the tank unit 2. The hot-water supplying mixing valve 2c is connected via a hot water delivery pipe 14 to the upper portion of the hot water storage tank 2a. A water supply branch pipe 15 branching off from the water supply pipe 13 is connected to the hot-water supplying mixing valve 2c. One end of the hot-water supply pipe 16 is further connected to the hot-water supplying mixing valve 2c. The other end of the hot-water supply pipe 16 is connected to a hot-water supply terminal such as a tap, a shower, or a bathtub (not shown).

[0019] During heating operation for heating water stored in the hot water storage tank 2a, the water stored in the hot water storage tank 2a is fed by the water pump 2b through the pipe 11 to the heat pump device 1, and heated in the heat pump device 1 to be high temperature hot water. The high temperature hot water generated in the heat pump device 1 returns through the pipe 12 to the tank unit 2, and flows into the hot water storage tank 2a from above. By such heating operation, in the hot water storage tank 2a, hot water and water are stored so as to form temperature stratification with a hot upper side and a cold lower side.

[0020] When hot water is supplied from the hot-water supply pipe 16 to the hot-water supply terminal, the high temperature hot water in the hot water storage tank 2a is supplied through the hot water delivery pipe 14 to the hot-water supplying mixing valve 2c, and low temperature water is supplied through the water supply branch pipe 15 to the hot-water supplying mixing valve 2c. The high temperature hot water and the low temperature water are mixed by the hot-water supplying mixing valve 2c, and then supplied through the hot-water supply pipe 16 to the hot-water supply terminal.

The hot-water supplying mixing valve 2c has a function of adjusting a mixture ratio between the high temperature hot water and the low temperature water so as to reach a hot-water supply temperature set by a user.

[0021] The heat pump device 1 includes a control unit 50. The control unit 50 is electrically connected to actuators and sensors (not shown) included in the heat pump device 1 and the tank unit 2, and user interface devices (not shown), and functions as control means for controlling operation of the storage type hot-water supply system. In Figure 2, the control unit 50 is provided in the heat pump device 1, but the control unit 50 may be provided other than in the heat pump device 1. The control unit 50 may be provided in the tank unit 2. The control unit 50 may be provided in the heat pump device 1 and the tank unit 2 in a divided manner so as to be able to mutually communicate.

[0022] During heating operation, the control unit 50 performs control so that a temperature of the hot water supplied from the heat pump device 1 to the tank unit 2 (hereinafter referred to as "hot water delivery temperature") reaches a target hot water delivery temperature. The target hot water delivery temperature is set to, for example, 65°C to 90°C. In Embodiment 1, the control unit 50 adjusts a rotation speed of the water pump 2b to control the hot water delivery temperature. The control unit 50 detects the hot water delivery temperature using a temperature sensor (not shown) provided in the water channel 26. If the detected hot water delivery temperature is higher than the target hot water delivery temperature, the rotation speed of the water pump 2b is corrected to be higher, and if the hot water delivery temperature is lower than the target hot water delivery temperature, the rotation speed of the water pump 2b is corrected to be lower. As such, the control unit 50 can perform control so that the hot water delivery temperature matches the target hot water delivery temperature. The hot water delivery temperature may be controlled by controlling a temperature of the refrigerant discharged from the first discharge passage 35 of the compressor 3, a rotation speed of the compressor 3, or the like.

[0023] An oil reservoir (not shown) that stores refrigerator oil is located in a lower portion of the internal space 38 of the sealed container 31 of the compressor 3 in Figure 1. In order to lubricate and seal a slide portion to reduce friction and gap leakage, the refrigerator oil is supplied from the oil reservoir into the compressing element 32. The refrigerator oil supplied into the compressing element 32 together with the compressed high temperature and high pressure refrigerant gas is discharged from the first discharge passage 35. Thus, a relatively large amount of refrigerator oil is discharged from the first discharge passage 35. The refrigerant gas and the refrigerator oil discharged from the first discharge passage 35 form a gas-liquid two-phase flow, which flows through the first gas cooler 4 to the second intake passage 36, and is released from the second intake passage 36 into the internal space 38 of the sealed container 31.

[0024] The refrigerator oil has a higher density than the refrigerant gas. Thus, the refrigerator oil having flowed from the second intake passage 36 into the internal space 38 of the sealed container 31 falls by gravity, and is stored in the oil reservoir in the lower portion of the internal space 38 of the sealed container 31. As such, the refrigerant is separated from the refrigerator oil. However, a part of the refrigerator oil is atomized and mixed in the refrigerant gas. A part of the refrigerator oil as a liquid film may be also raised and spattered by a flow of the refrigerant gas when the refrigerant and the refrigerator oil are released from an outlet of the second intake passage 36 into the internal space 38 of the sealed container 31. Thus, a small amount of refrigerator oil is mixed in the refrigerant gas passing through the gap between the rotor 331 and the stator 332 of the electric actuating element 33 to a top of the electric actuating element 33. A part of the mixed refrigerator oil is separated from the refrigerant gas by a centrifugal force caused by rotation of the rotor 331. The remaining refrigerator oil together with the refrigerant gas is discharged through the second discharge passage 37 out of the sealed container 31. From the above, a mass flow rate of the refrigerator oil discharged from the first discharge passage 35 is higher than a mass flow rate of the refrigerator oil discharged from the second discharge passage 37. On the other hand, a mass flow rate of the refrigerant discharged from the first discharge passage 35 is equal to a mass flow rate of the refrigerant discharged from the second discharge passage 37.

[0025] A large amount of refrigerator oil together with the refrigerant gas is circulated to the first refrigerant heat transfer channel in the first gas cooler 4. On the other hand, a smaller amount of refrigerator oil is circulated to the second refrigerant heat transfer channel in the second gas cooler 5 as compared to the first gas cooler 4. The refrigerator oil has a much higher viscosity than the refrigerant. Thus, the large amount of refrigerator oil being circulated to the first gas cooler 4 easily increases refrigerant pressure loss. The increase in refrigerant pressure loss of the first gas cooler 4 increases discharge pressure of the compressor 3, and increases input power for the compressor 3, thereby reducing a COP (coefficient of performance). In order to solve this problem, in Embodiment 1, a total sectional area of the first refrigerant heat transfer channel(s) in the first gas cooler 4 through which the refrigerant and the refrigerator oil discharged from the first discharge passage 35 pass is larger than a total sectional area of the second refrigerant heat transfer channel(s) in the second gas cooler 5 through which the refrigerant and the refrigerator oil discharged from the second discharge passage 37 pass.

[0026] A sectional area of the channel herein refers to an area of a range of a flowing fluid in a section perpendicular to a flow direction of the fluid. If there are a plurality of first refrigerant heat transfer channels in the first gas cooler 4, that is, if the refrigerant and the refrigerator oil having flowed into the first gas cooler 4 are split into the plurality of first refrigerant heat transfer channels and flow in parallel, a total sectional area of the first refrigerant heat transfer channels refers to a sum of sectional area of each of the first refrigerant heat transfer channels. Similarly, if there are a plurality of second refrigerant heat transfer channels in the second gas cooler 5, that is, if the refrigerant and the refrigerator oil having flowed into the second gas cooler 5 are split into the plurality of second refrigerant heat transfer channels and flow in parallel, a total sectional area of the second refrigerant heat transfer channels refer to a sum of sectional area of each of the first refrigerant heat transfer channels.

[0027] As described below, in Embodiment 1, the total sectional area of the first refrigerant heat transfer channel(s) in the first gas cooler 4 is larger than the total sectional area of the second refrigerant heat transfer channel(s) in the second gas cooler 5, thereby reliably preventing an increase in refrigerant pressure loss of the first gas cooler 4. This reduces discharge pressure of the compressor 3, reduces input power for the compressor 3, and improves a COP.

[0028] Figure 3 is a perspective view of essential portions of the first gas cooler 4 in Embodiment 1. Figure 4 is a sectional view of essential portions of the first gas cooler 4 in Embodiment 1. As shown in Figures 3 and 4, the first gas cooler 4 includes one first twist pipe 41 and three first refrigerant heat transfer pipes 42. Figure 4 shows a section in a longitudinal direction of the first twist pipe 41. In Figure 3, the three first refrigerant heat transfer pipes 42 are denoted by reference numerals 42a, 42b, 42c for convenience. In addition, in Figure 3, for easy distinction between the first refrigerant heat transfer pipes 42a, 42b, 42c, the first refrigerant heat transfer pipes 42a, 42c are hatched for convenience. Specifically, the hatching in Figure 3 does not refer to sections.

[0029] In the first gas cooler 4 in Embodiment 1, the refrigerant and the refrigerator oil flow in the first refrigerant heat transfer pipe 42. Specifically, the first refrigerant heat transfer pipe 42 forms the first refrigerant heat transfer channel. The first gas cooler 4 in Embodiment 1 includes the three first refrigerant heat transfer pipes 42a, 42b, 42c, that is, the three first refrigerant heat transfer channels. The refrigerant and the refrigerator oil having flowed into the first gas cooler 4 are split into the three first refrigerant heat transfer pipes 42a, 42b, 42c, that is, the three first refrigerant heat transfer channels, and flow in parallel. However, in the present invention, the number of the first refrigerant heat transfer channel(s) in the first gas cooler 4, that is, the first heat exchanger is not limited to three, but may be one, two, four or more.

[0030] The first twist pipe 41 has a helical groove 411 in an outer periphery thereof. The number of the groove(s) 411 is equal to the number of the first refrigerant heat transfer pipe(s) 42. Specifically, in Embodiment 1, the first twist pipe 41 has three grooves 411 in parallel. In Figure 3, the three grooves 411 are denoted by reference numerals 411a, 411b, 411c. Each of the grooves 411a, 411b, 411c continuously forms a helix. The first refrigerant heat transfer pipes 42a, 42b, 42c are, respectively, fitted in the grooves 411a, 411b, 411c and wound helically along shapes of the grooves 411a, 411b, 411c. Such a configuration can increase a contact heat transfer area between the first twist pipe 41 and the first refrigerant heat transfer pipe 42.

[0031] In the first gas cooler 4 in Embodiment 1, the first twist pipe 41 forms a first liquid heat transfer channel through which water passes. In Embodiment 1, one first twist pipe 41, that is, one first liquid heat transfer channel is provided in the first gas cooler 4. However, in the present invention, a plurality of first liquid heat transfer channels may be provided in the first gas cooler 4, that is, the first heat exchanger so that a liquid such as water is split into the first liquid heat transfer channels and flows in parallel.

[0032] Water flows through the first twist pipe 41 from right to left in Figures 3 and 4. The refrigerant and the refrigerator oil flow helically through the first refrigerant heat transfer pipe 42 from left to right in Figures 3 and 4. Specifically, a flow direction of water is opposite to a traveling direction of the refrigerant flowing helically to form counter flows.

[0033] An inner diameter SRi of the first twist pipe 41 is herein defined as a length of a portion in Figure 4. Specifically, the inner diameter SRi of the first twist pipe 41 refers to an inner diameter of a portion with a smallest inner diameter in the first twist pipe 41.

[0034] Figure 5 is an enlarged sectional view of essential portions of the first gas cooler 4 and the second gas cooler 5 in Embodiment 1. In Figure 5, (1) shows the first gas cooler 4. In Figure 5, (2) shows the second gas cooler 5. As shown in Figure 5, the first twist pipe 41 and the first refrigerant heat transfer pipe 42 are joined with a heat transfer material 60 such as solder. The second gas cooler 5 includes a second twist pipe 51 and a second refrigerant heat transfer pipe 52. The second twist pipe 51 has a helical groove 511 in an outer periphery thereof. In the second gas cooler 5 in Embodiment 1, the second refrigerant heat transfer pipe 52 forms a second refrigerant heat transfer channel, and the second twist pipe 51 forms a second liquid heat transfer channel. Since the second gas cooler 5 has similar structure as the first gas cooler 4, drawings corresponding to Figures 3 and 4 are omitted. The description above on the first gas cooler 4 also applies to the second gas cooler 5. Figure 5 shows a section in a longitudinal direction of the first twist pipe 41 or the second twist pipe 51.

[0035] As shown in Figure 5, in a case where the first refrigerant heat transfer pipe 42 or the second refrigerant heat transfer pipe 52 originally having a circular tubular shape is wound helically around the first twist pipe 41 or the second twist pipe 51, a sectional shape of the first refrigerant heat transfer pipe 42 or the second refrigerant heat transfer pipe 52 after being wound is not a circle, but is a flat or elliptical shape with a long side in an axial direction of the first twist pipe 41 or the second twist pipe 51. An inner diameter di1 of the first refrigerant heat transfer pipe 42 or an inner diameter di2 of the second refrigerant heat transfer pipe 52 herein refer to an inner diameter of a circular state before the refrigerant heat transfer pipe is wound around the first twist pipe 41 or the second twist pipe 51.

[0036] Generally, in the first gas cooler 4 or the second gas cooler 5, an end of the first refrigerant heat transfer pipe

42 or the second refrigerant heat transfer pipe 52 is not wound around the first twist pipe 41 or the second twist pipe 51. Thus, in such a portion, the inner diameter di1 of the first refrigerant heat transfer pipe 42 or the inner diameter di2 of the second refrigerant heat transfer pipe 52 before being wound around the first twist pipe 41 or the second twist pipe 51 may be measured.

[0037] Instead of the above definition, the first refrigerant heat transfer pipe 42 or the second refrigerant heat transfer pipe 52 wound around the first twist pipe 41 or the second twist pipe 51 may be regarded to have an elliptical shape, and an average value of a long diameter and a short diameter of the ellipse may be used as the inner diameter di1 of the first refrigerant heat transfer pipe 42 or the inner diameter di2 of the second refrigerant heat transfer pipe 52.

[0038] As shown in Figure 5, in Embodiment 1, the inner diameter di1 of the first refrigerant heat transfer pipe 42 in the first gas cooler 4 is desirably larger than the inner diameter di2 of the second refrigerant heat transfer pipe 52 in the second gas cooler 5. In addition, a twist pitch p of the first twist pipe 41 in the first gas cooler 4 is desirably larger than a twist pitch p2 of the second twist pipe 51 in the second gas cooler 5. The twist pitch p of the first twist pipe 41 in the first gas cooler 4 and the twist pitch p2 of the second twist pipe 51 in the second gas cooler 5 are herein defined as lengths of portions in Figure 5. Specifically, the twist pitch p of the first twist pipe 41 is a distance between centers of two peaks with the groove 411 therebetween in a section in a longitudinal direction of the first twist pipe 41. Similarly, the twist pitch p2 of the second twist pipe 51 is a distance between centers of two peaks with the groove 511 therebetween in a section in a longitudinal direction of the second twist pipe 51.

[0039] In an example described below, a case of using carbon dioxide as the refrigerant will be described. In the example described below, the number of the first refrigerant heat transfer channel(s) in the first gas cooler 4 is equal to the number of the second refrigerant heat transfer channel(s) in the second gas cooler 5. Figure 6 shows temperature changes of the refrigerant and water in the first gas cooler 4 and the second gas cooler 5 as a whole, and a split position between the first gas cooler 4 and the second gas cooler 5. The axis of abscissa in Figure 6 represents a ratio to a total length of the first twist pipe 41 and the second twist pipe 51 (that is, a sum of lengths of the first liquid heat transfer channel and the second liquid heat transfer channel). An origin (0) on the axis of abscissa in Figure 6 represents a water outlet and a refrigerant inlet of the first gas cooler 4, and a right end (1) on the axis of abscissa represents a water inlet and a refrigerant outlet of the second gas cooler 5.

[0040] As described above, a large amount of refrigerator oil together with the refrigerant gas is circulated in the first refrigerant heat transfer pipe 42 in the first gas cooler 4. In the first gas cooler 4, hot refrigerator oil is also subjected to heat exchange with water. Specific heat of the refrigerator oil being lower than specific heat of the refrigerant gas may cause a reduction in heating capability and a resulting reduction in hot-water supply efficiency. In a relationship between the temperature and the specific heat of the refrigerant gas and the refrigerator oil, the specific heat of the refrigerant gas significantly increases at a temperature of 20°C to 60°C, while the specific heat of the refrigerator oil is substantially constant irrespective of the temperature. In order to prevent a reduction in heating capability due to the refrigerant gas containing a large amount of refrigerator oil, the refrigerant gas needs to contain little refrigerator oil in a temperature zone with a significant increase in specific heat of the refrigerant gas. As shown in Figure 6, a temperature of a pinch point at which temperatures of the refrigerant gas and water are closest is about 50°C. Thus, an upper limit temperature in a range with a rapid increase in specific heat of the refrigerant gas is about the temperature at the pinch point plus 10°C. Thus, if an outlet temperature (≈ a temperature of the second intake passage 36) of the first refrigerant heat transfer pipe 42 in the first gas cooler 4 is 10°C or more higher than the temperature at the pinch point, a reduction in heating capability can be prevented. If the outlet temperature of the first refrigerant heat transfer pipe 42 in the first gas cooler 4 is at least higher than the temperature at the pinch point, a significant reduction in heating capability can be prevented. From the above, the split position between the first gas cooler 4 and the second gas cooler 5 is desirably on a high temperature side of the pinch point at which the temperatures of the refrigerant gas and water are closest. In particular, in Embodiment 1, as shown in Figure 6, the length of the first twist pipe 41 in the first gas cooler 4 is desirably about 10% on the high temperature side of the total length of the first twist pipe 41 and the second twist pipe 51.

[0041] Figure 7 shows density change of the refrigerant in the first gas cooler 4 and the second gas cooler 5 as a whole. The axis of abscissa in Figure 7 refers to the same as the axis of abscissa in Figure 6. As shown in Figure 7, the refrigerant at higher temperature has a lower density.

[0042] Pressure loss ΔP of the refrigerant in the refrigerant heat transfer pipe is expressed by the following expression 1. Here, the sectional shape of the refrigerant heat transfer pipe is a circle for simplicity of description.

$$\Delta P = \lambda / di \cdot \rho / 2 \cdot u^2 \cdot L \qquad \text{(Expression 1)}$$

where λ is a pipe friction coefficient, di [m] is an inner diameter of the refrigerant heat transfer pipe, p [kg/m^3] is a refrigerant density, u [m/s] is a refrigerant flow speed, and L [m] is a channel length.

[0043] When the mass flow rate of the refrigerant is Gr [kg/s] and the channel sectional area of the refrigerant heat transfer pipe is A [m^2], the refrigerant flow speed u is expressed by the following expressions 2 and 3.

$$u = Gr/(\rho \cdot A) \qquad \text{(Expression 2)}$$

$$A = \pi/4 \cdot di^2 \qquad \text{(Expression 3)}$$

[0044] Here, for simplicity of description, it is assumed that the shapes and the refrigerant flow rates of the first refrigerant heat transfer pipe 42 and the second refrigerant heat transfer pipe 52 are constant, and the pipe friction coefficient λ does not change. From the above expression, the refrigerant pressure loss ΔP per unit channel length is proportional to $1/\rho$.

[0045] In Embodiment 1, the refrigerant gas containing the large amount of refrigerator oil is circulated to the first gas cooler 4, and the refrigerant gas containing only the small amount of refrigerator oil is circulated to the second gas cooler 5. If a viscosity of a CO_2 gas refrigerant in the first gas cooler 4 is 1, an average viscosity ratio of the refrigerator oil is 311. As such, the refrigerator oil has a significantly higher viscosity than the CO_2 gas refrigerant. This increases pressure loss of the refrigerant gas containing the large amount of refrigerator oil.

[0046] The mass flow rate of the refrigerator oil is Goil [kg/s]. An oil circulation rate OC [%] of the first gas cooler 4 or the second gas cooler 5 is expressed by the following expression 4.

$$OC = Goil / (Gr + Goil) \times 100 \qquad \text{(Expression 4)}$$

[0047] The oil circulation rate OC is a ratio of the mass flow rate of the refrigerator oil with respect to a sum of the mass flow rate of the refrigerant and the mass flow rate of the refrigerator oil. In a rated operation state of the heat pump device 1, the oil circulation rate OC of the first gas cooler 4 is preferably not less than 2%, and more preferably not less than 5%. In addition, in the rated operation state of the heat pump device 1, the oil circulation rate OC of the first gas cooler 4 is preferably not more than 20%, and more preferably not more than 10%. Setting the oil circulation rate OC of the first gas cooler 4 to the above-described lower limit value or more allows heat from the hot refrigerator oil in the compressor 3 to be effectively used for heating water in the first gas cooler 4, improving heating capability. Setting the oil circulation rate OC of the first gas cooler 4 to the above-described upper limit value or less can reliably reduce the refrigerant pressure loss of the first gas cooler 4, and also reliably prevent an excessive reduction in the amount of the refrigerator oil in the compressor 3.

[0048] In the rated operation state of the heat pump device 1, the oil circulation rate OC of the second gas cooler 5 is preferably not less than 0.01%, and more preferably not less than 0.1%. In the rated operation state of the heat pump device 1, the oil circulation rate OC of the second gas cooler 5 is preferably not more than 1%, and more preferably not more than 0.5%. Setting the oil circulation rate OC of the second gas cooler 5 to the above-described upper limit value or less can reliably reduce the refrigerant pressure loss of the second gas cooler 5. If the oil circulation rate OC of the second gas cooler 5 is low and close to the above-described lower limit value, the refrigerator oil has little influence, and there is no need to further reduce the oil circulation rate OC of the second gas cooler 5 to be lower than the above-described lower limit value. Depending on operation conditions of the heat pump device 1, the oil circulation rate OC of the second gas cooler 5 may be lower than the above-described lower limit value.

[0049] If the oil circulation rate OC is about 5% to 10%, the refrigerant pressure loss is about 1.6 to 2.0 times larger than that when the oil circulation rate OC is 0.5% or less under the same other conditions.

[0050] Figure 8 shows ratios of refrigerant pressure losses of the first gas cooler 4 and the second gas cooler 5 in a case where the first gas cooler 4 and the second gas cooler 5 have the same shape other than their channel lengths. Figure 9 is a configuration diagram of a conventional heat pump device. First, a conventional heat pump device 70 in Figure 9 will be described. Components common with those of the heat pump device 1 according to Embodiment 1 are denoted by the same reference numerals and overlapping descriptions will be omitted. The heat pump device 70 in Figure 9 includes a compressor 71 having one intake passage and one discharge passage instead of the compressor 3 in the heat pump device 1 according to Embodiment 1. The heat pump device 70 includes a single gas cooler 72 instead of the first gas cooler 4 and the second gas cooler 5. In the heat pump device 70, low pressure refrigerant sucked from the pipe 21 into the compressor 71 is compressed by the compressor 71 into high pressure refrigerant. The high pressure refrigerant is discharged from the compressor 71 and passes through the pipe 10 and the gas cooler 72 to the pipe 19.

[0051] The case of "0.5% OR LESS IN OVERALL GAS COOLER(S)" in Figure 8 refers to a case where, as in the conventional heat pump device 70 in Figure 9, the gas cooler 72 is not split into the first gas cooler 4 and the second gas cooler 5, and the refrigerant from which the refrigerator oil is separated in the sealed container of the compressor 71 is caused to flow into the gas cooler 72. Specifically, this refers to a case of a conventional refrigeration cycle where the refrigerant is not returned into the sealed container 31 of the compressor 3 between the first gas cooler 4 and the second gas cooler 5. In this case, if the refrigerant pressure loss of the overall gas cooler 72 is 1, a ratio of the refrigerant pressure loss of a portion corresponding to a channel length of 10% on a refrigerant high temperature side of a total channel length of the gas cooler 72 is 0.17. The ratio of the refrigerant pressure loss of a remaining portion corresponding to a channel length of 90% on a refrigerant low temperature side is 0.83. As shown in Figure 7, on the high temperature side of the refrigerant gas, the refrigerant density is low, and thus the ratio of the refrigerant pressure loss of the portion corresponding to the channel length of 10% of the total channel length is 17% of the total refrigerant pressure loss, and higher than the ratio of the channel length.

[0052] The case of "OIL CIRCULATION RATE IS HIGH IN FIRST GAS COOLER AND 0.5% OR LESS IN SECOND GAS COOLER" in Figure 8 refers to a case where the refrigerant pressure loss is twice larger than that when the oil circulation rate is 0.5% or less because of the oil circulation rate of about 5% to 10% of the first gas cooler 4. Here, the channel length of 10% on the refrigerant high temperature side of the total channel length of the first gas cooler 4 and the second gas cooler 5 corresponds to the first gas cooler 4. In this case, if the refrigerant pressure loss of the overall gas cooler 72 is 1, the ratio of the refrigerant pressure loss of the first gas cooler 4 is 0.17 x 2 = 0.34. Thus, the ratio of the refrigerant pressure loss of the first gas cooler 4 and the second gas cooler 5 as a whole is 0.34 + 0.83 = 1.17. As such, if the refrigerant pressure loss is twice larger on the refrigerant high temperature side with high refrigerant pressure loss per unit channel length, the refrigerant pressure loss of the overall gas coolers is significantly influenced. Thus, the refrigerant pressure loss of the overall gas coolers is 1.17 times as compared to a case with a low oil circulation rate as a whole. The ratio of the refrigerant pressure loss of the first gas cooler 4 with respect to the overall gas coolers is 29% and high.

[0053] Although the first gas cooler 4 has a higher oil circulation rate than the second gas cooler 5, mainly flowing medium is the refrigerant. Thus, the heat exchanger constituting the first gas cooler 4 preferably has a configuration of a general heat exchanger for a refrigerant rather than of an oil cooler type heat exchanger. For example, the first gas cooler 4 preferably uses a twist pipe like the second gas cooler 5.

[0054] From the above, the high oil circulation rate of the first gas cooler 4 easily increases the refrigerant pressure loss of the first gas cooler 4, and increases discharge pressure of the compressor 3. This easily increases input power for the compressor 3 and reduces the COP. Thus, in Embodiment 1, the refrigerant pressure loss of the first gas cooler 4 is reduced as described below.

[0055] A relationship among the inner diameter di1 of the first refrigerant heat transfer pipe 42, the channel length L of the first refrigerant heat transfer pipe 42, and the refrigerant pressure loss of the first gas cooler 4 will be described. The refrigerant pressure loss ΔP in the first refrigerant heat transfer pipe 42 has the following proportional relationship from the expressions 1 to 3 above, with a pipe friction coefficient, a refrigerant density, and a refrigerant flow rate being constant.

$$\Delta P \propto L / (di1)^5$$

[0056] Thus, in order to reduce the refrigerant pressure loss of the first gas cooler 4, it is advantageous to reduce the channel length L of the first refrigerant heat transfer pipe 42 and increase the inner diameter di1 of the first refrigerant heat transfer pipe 42.

[0057] Next, an advantage of an increase in the twist pitch p of the first twist pipe 41 will be described. Figure 10 shows a relationship between a ratio of the twist pitch p to the inner diameter SRi of the first twist pipe 41 and a heat transfer coefficient on water side. Figure 10 shows change in heat transfer coefficient on water side with a constant inner diameter SRi and an increased twist pitch p of the first twist pipe 41. In Figure 10, the heat transfer coefficient on water side is represented by a ratio to the heat transfer coefficient on water side when p/SRi is 1. As shown in Figure 10, with increasing p/SRi, that is, with increasing twist pitch p of the first twist pipe 41, the heat transfer coefficient on water side increases.

[0058] Figure 11 shows a relationship between the ratio of the twist pitch p to the inner diameter SRi of the first twist pipe 41 and a required length of the first twist pipe 41. In Figure 11, a length of the first twist pipe 41 required, when the twist pitch p is increased with a constant inner diameter SRi of the first twist pipe 41, for obtaining an equal amount of heat exchange is represented by a ratio to a reference length. The first gas cooler 4 as a twist pipe type heat exchanger is configured so that the first refrigerant heat transfer pipe 42 is wound along the helical groove 411 in the first twist pipe 41. Thus, increasing the twist pitch p of the first twist pipe 41 reduces the length of the first refrigerant heat transfer pipe 42 wound around the first twist pipe 41 per unit length, thereby reducing a contact area between the first refrigerant heat transfer pipe 42 and the first twist pipe 41. Thus, with increasing twist pitch p of the first twist pipe 41, the length of the first twist pipe 41 required for equalizing the amounts of heat exchange of the refrigerant and water is increased. However, as shown in Figure 10, with increasing twist pitch p, the heat transfer coefficient on water side increases, thereby increasing heat exchange efficiency per unit length of the first twist pipe 41. These relationships determines the relationship in Figure 11.

[0059] Figure 12 shows a relationship between the ratio of the twist pitch p to the inner diameter SRi of the first twist pipe 41 and a required length of the first refrigerant heat transfer pipe 42. In Figure 12, the length of the first refrigerant heat transfer pipe 42 required, when the twist pitch p is increased with a constant inner diameter SRi of the first twist pipe 41, for obtaining an equal amount of heat exchange is represented by a ratio to the length of the first refrigerant heat transfer pipe 42 required at p/SRi of 1. As described with reference to Figure 11, with increasing twist pitch p of the first twist pipe 41, a required length of the first twist pipe 41 is increased. However, with increasing twist pitch p of the first twist pipe 41, the length of the first refrigerant heat transfer pipe 42 wound around the first twist pipe 41 per unit length is reduced. Thus, as shown in Figure 12, with increasing twist pitch p of the first twist pipe 41, the required length of the first refrigerant heat transfer pipe 42 is reduced. However, in a region at p/SRi of about 1.8 or more, the required length of the first refrigerant heat transfer pipe 42 is less likely to be reduced.

[0060] Summarizing the above characteristics, if the twist pitch p of the first twist pipe 41 is increased, the length of the first twist pipe 41 required for obtaining an equal amount of heat exchange is increased, while the ratio of the heat transfer coefficient on water side is increased, thereby relatively gently increasing the required length of the first twist pipe 41. As shown in Figure 12, with increasing twist pitch p of the first twist pipe 41, the length of the first refrigerant heat transfer pipe 42 can be effectively reduced, which is advantageous for reducing the refrigerant pressure loss of the first gas cooler 4.

[0061] Figure 13 shows a relationship among the refrigerant pressure loss of the first gas cooler 4, the ratio of the twist pitch p to the inner diameter SRi of the first twist pipe 41, and the inner diameter di1 of the first refrigerant heat transfer pipe 42. In Figure 13 and thereafter, a ratio di1/di2 of the inner diameter di1 of the first refrigerant heat transfer pipe 42 in the first gas cooler 4 to the inner diameter di2 of the second refrigerant heat transfer pipe 52 in the second gas cooler is referred to as "an inner diameter ratio". Figure 13 shows changes in refrigerant pressure loss of the first gas cooler 4 when the twist pitch p of the first twist pipe 41 is changed for each of cases where the inner diameter ratio di1/di2 is set to a plurality of values in Figure 13 with a constant amount of heat exchange in the first gas cooler 4. In Figure 13, the refrigerant pressure loss of the first gas cooler 4 is represented by a ratio to the refrigerant pressure loss of the first gas cooler 4 when values of the inner diameter ratio di1/di2 and p/SRi are both 1.

[0062] Figure 14 shows a relationship between the ratio of the twist pitch p to the inner diameter SRi of the first twist pipe 41 in the first gas cooler 4 and the length of the first twist pipe 41 in each of the cases in Figure 13. In Figure 14, the length of the first twist pipe 41 is represented by a ratio to the length of the first twist pipe 41 when the values of the inner diameter ratio di1/di2 and p/SRi are both 1. In Figures 13 and 14, the ratio of the twist pitch p2 to the inner diameter SRi of the second twist pipe 51 in the second gas cooler 5 is about 1. The inner diameter SRi of the first twist pipe 41 in the first gas cooler 4 is equal to the inner diameter SRi of the second twist pipe 51 in the second gas cooler 5.

[0063] As shown in Figure 13, in a case where the inner diameter ratio di1/di2 is equal, with increasing p/SRi, that is, with increasing twist pitch p of the first twist pipe 41, the refrigerant pressure loss of the first gas cooler 4 is reduced. In a case where the twist pitch p of the first twist pipe 41 is equal, with increasing inner diameter ratio di1/di2, that is, with increasing inner diameter di1 of the first refrigerant heat transfer pipe 42, the refrigerant pressure loss of the first gas cooler 4 is reduced.

[0064] As described above, with increasing inner diameter ratio di1/di2, that is, with increasing inner diameter di1 of the first refrigerant heat transfer pipe 42, the refrigerant pressure loss of the first gas cooler 4 is more effectively reduced. However, with increasing inner diameter di1 of the first refrigerant heat transfer pipe 42, the flow speed of the refrigerant in the first refrigerant heat transfer pipe 42 is reduced, thereby reducing a heat transfer coefficient in the first refrigerant heat transfer pipe 42. Thus, as shown in Figure 14, with increasing inner diameter ratio di1/di2, that is, with increasing inner diameter di1 of the first refrigerant heat transfer pipe 42, the length of the first twist pipe 41 required for obtaining an equal amount of heat exchange is increased. In addition, as shown in Figure 14, with increasing p/SRi, that is, with increasing twist pitch p of the first twist pipe 41, the required length of the first twist pipe 41 is increased.

[0065] If the length of the first twist pipe 41 in the first gas cooler 4 is increased, a size of the overall gas coolers including the first gas cooler 4 and the second gas cooler 5 may be increased to increase a size of a casing of the heat pump device 1. In addition, if the length of the first twist pipe 41 in the first gas cooler 4 is increased, an amount of material required for the first twist pipe 41 is increased to increase weight and cost. In addition, if the length of the first twist pipe 41 forming the water channel is excessively increased, an amount of heat dissipation from the first gas cooler 4 out of the heat pump device 1 may be increased or the pressure loss on water side may be increased.

[0066] As described above, with increasing twist pitch p of the first twist pipe 41 in the first gas cooler 4, the refrigerant pressure loss of the first gas cooler 4 is reduced, while the length of the first twist pipe 41 is increased. Thus, excessively increasing the twist pitch p of the first twist pipe 41 may excessively increase the length of the first twist pipe 41, thereby causing the negative effects as described above. In this view, p/SRi as the ratio of the twist pitch p to the inner diameter SRi of the first twist pipe 41 is desirably not more than 1.8. As described above, in the region at p/SRi of more than 1.8, the required length of the first refrigerant heat transfer pipe 42 is less likely to be reduced by increasing the twist pitch p of the first twist pipe 41. Thus, in the region at p/SRi of more than 1.8, further increasing the twist pitch p of the first twist pipe 41 is less effective for further reducing the refrigerant pressure loss, and also easily causes the negative effects of the increased length of the first twist pipe 41. On the other hand, p/SRi of 1.8 or less can reliably prevent the negative effects of the increased length of the first twist pipe 41.

[0067] In addition, p/SRi of the first twist pipe 41 in the first gas cooler 4 is preferably not less than 1.1, more preferably not less than 1.2, and further preferably not less than 1.4. Setting p/SRi to preferably 1.1 or more, more preferably 1.2 or more, and further preferably 1.4 or more can effectively reduce the length of the first refrigerant heat transfer pipe 42 (see Figure 12). This can more reliably reduce the refrigerant pressure loss of the first gas cooler 4. In short, p/SRi of the first twist pipe 41 in the first gas cooler 4 is preferably not less than 1.1 and not more than 1.8, more preferably not less than 1.2 and not more than 1.8, and further preferably not less than 1.4 and not more than 1.8. By setting p/SRi to such a range, markedly advantageously, increasing the twist pitch p of the first twist pipe 41 can sufficiently increase the effect of reducing the refrigerant pressure loss of the first gas cooler 4 and can reliably prevent the negative effects due to the increased length of the first twist pipe 41.

[0068] Next, a preferable maximum value of the inner diameter ratio di1/di2 of the first refrigerant heat transfer pipe 42 and the second refrigerant heat transfer pipe 52 will be described. Figure 15 shows change in the refrigerant pressure loss of the first gas cooler 4 when the inner diameter ratio di1/di2 of the first refrigerant heat transfer pipe 42 and the second refrigerant heat transfer pipe 52 is changed at p/SRi of 1.8 of the first twist pipe 41. In Figure 15, the refrigerant pressure loss of the first gas cooler 4 is represented by a ratio to a sum of the refrigerant pressure loss of the first gas cooler 4 and the refrigerant pressure loss of the second gas cooler 5 (that is, the refrigerant pressure loss of the overall gas coolers). As shown in Figure 15, with increasing inner diameter ratio di1/di2, that is, with increasing inner diameter di1 of the first refrigerant heat transfer pipe 42, the refrigerant pressure loss of the first gas cooler 4 is reduced, and the ratio of the refrigerant pressure loss of the first gas cooler 4 to the refrigerant pressure loss of the overall gas coolers is reduced. However, as shown in Figure 14, with increasing inner diameter ratio di1/di2, that is, with increasing inner diameter di1 of the first refrigerant heat transfer pipe 42, the length of the first twist pipe 41 is increased. Also, in the first gas cooler 4 to which the large amount of refrigerator oil is circulated, too large an inner diameter di1 of the first refrigerant heat transfer pipe 42 may reduce the refrigerant flow speed, thereby reducing flowage of the refrigerator oil. This may significantly increase retention of refrigerator oil in the first gas cooler 4. For these reasons, it is desirable to set the inner diameter ratio di1 of the first refrigerant heat transfer pipe 42 in the first gas cooler to a value that is not too large.

[0069] As shown in Figure 6, the channel length of the first gas cooler 4 is about 10% of the channel length of the overall gas coolers. Thus, if the ratio of the refrigerant pressure loss of the first gas cooler 4 with respect to the refrigerant pressure loss of the overall gas coolers can be reduced to about 10%, it can be said that the refrigerant pressure loss of the first gas cooler 4 is sufficiently reduced. It can be also said that further reducing the refrigerant pressure loss of the first gas cooler 4, that is, reducing the refrigerant pressure loss per unit channel length in the first gas cooler 4 to be smaller than the refrigerant pressure loss per unit channel length in the second gas cooler 5 is an excess. As shown in Figure 15, if the inner diameter ratio di1/di2 is about 1.4, the ratio of the refrigerant pressure loss of the first gas cooler 4 with respect to the refrigerant pressure loss of the overall gas coolers is about 10%. Thus, it can be said that setting the value of the inner diameter ratio di1/di2 to 1.4 sufficiently reduces the refrigerant pressure loss of the first gas cooler 4 in the relationship with the ratio of the channel length. However, too large a value of the inner diameter ratio di1/di2, that is, too large an inner diameter di1 of the first refrigerant heat transfer pipe 42 may cause the negative effects as described above such as an excessive length of the first twist pipe 41 or an increase in the retention of refrigerator oil in the first gas cooler 4. On the other hand, with the value of the inner diameter ratio di1/di2 of 1.4 or less, the inner diameter di1 of the first refrigerant heat transfer pipe 42 is not too large, thereby reliably preventing the negative effects.

[0070] In addition, the value of the inner diameter ratio di1/di2 of the first refrigerant heat transfer pipe 42 and the second refrigerant heat transfer pipe 52 is preferably not less than 1.1, and more preferably not less than 1.2. Setting the value of the inner diameter ratio di1/di2 to preferably 1.1 or more, and more preferably 1.2 or more can more reliably reduce the refrigerant pressure loss of the first gas cooler 4 (see Figure 13). In short, the value of the inner diameter ratio di1/di2 is preferably not less than 1.1 and not more than 1.4, and more preferably not less than 1.2 and not more than 1.4. By setting the value of the inner diameter ratio di1/di2 to such a range, markedly advantageously, the negative effects described above due to the excessive increase in the inner diameter di1 of the first refrigerant heat transfer pipe 42 can be reliably prevented, and the refrigerant pressure loss of the first gas cooler 4 can be sufficiently reduced.

[0071] As described above, according to Embodiment 1, the refrigerant pressure loss of the first gas cooler 4 can be reliably prevented to reduce input power for the compressor 3 and improve a COP.

[0072] As shown in Figure 7, the refrigerant density in the second gas cooler 5 is higher than the refrigerant density in the first gas cooler 4. As described above, with increasing refrigerant density, the refrigerant pressure loss per unit channel length is reduced. Thus, assuming other conditions are equal, the refrigerant pressure loss per unit length of the second refrigerant heat transfer pipe 52 in the second gas cooler 5 is smaller than the refrigerant pressure loss per unit length of the first refrigerant heat transfer pipe 42 in the first gas cooler 4. Thus, even if the inner diameter di2 of the second refrigerant heat transfer pipe 52 or the sectional area of each second refrigerant heat transfer channel in the second gas cooler 5 is smaller than the inner diameter di1 of the first refrigerant heat transfer pipe 42 or the sectional area of each first refrigerant heat transfer channel in the first gas cooler 4, the refrigerant pressure loss of the second gas cooler 5 can be sufficiently reduced. In addition, the inner diameter di2 of the second refrigerant heat transfer pipe 52 or the sectional area of each second refrigerant heat transfer channel in the second gas cooler 5 being relatively small increases the refrigerant flow speed in the second refrigerant heat transfer pipe 52, that is, in each second refrigerant heat transfer channel, thereby increasing a heat transfer coefficient of the refrigerant. This can reduce the length of the second twist pipe 51, that is, the second liquid heat transfer channel in the second gas cooler. From the above, the inner diameter di1 of the first refrigerant heat transfer pipe 42 or the sectional area of each first refrigerant heat transfer channel in the first gas cooler 4 is preferably larger than the inner diameter di2 of the second refrigerant heat transfer pipe 52 or the sectional area of each second refrigerant heat transfer channel in the second gas cooler 5.

[0073] Figure 16 shows change in heat transfer coefficient on water side in a case where the twist pitch p of the first twist pipe 41 is equal to the twist pitch p2 of the second twist pipe 51 and the inner diameters SRi of the first twist pipe 41 and the second twist pipe 51 are equal. The axis of abscissa in Figure 16 refers to the same as the axis of abscissa in Figure 6. In Figure 16, the heat transfer coefficient on water side is represented by a ratio to the heat transfer coefficient on water side at the water outlet of the first gas cooler 4. As shown in Figure 16, with increasing distance from the refrigerant inlet and the water outlet of the first gas cooler 4, that is, with decreasing temperature of water, the heat transfer coefficient on water side is reduced. Thus, if the twist pitch p of the first twist pipe 41 is equal to the twist pitch p2 of the second twist pipe 51, and the inner diameters SRi of the first twist pipe 41 and the second twist pipe 51 are equal, the heat transfer coefficient on water side in the second gas cooler 5 is lower than the heat transfer coefficient on water side in the first gas cooler 4. In this view, in the second gas cooler 5, it is desirable to set a relatively small twist pitch p2 of the second twist pipe 51 to increase a contact area between the second refrigerant heat transfer pipe 52 and the second twist pipe 51. This can reduce the length of the second twist pipe 51 in the second gas cooler 5. In contrast, as described above, the twist pitch p of the first twist pipe 41 in the first gas cooler 4 is desirably relatively large. From the above, the twist pitch p of the first twist pipe 41 in the first gas cooler 4 is preferably larger than the twist pitch p2 of the second twist pipe 51 in the second gas cooler 5.

[0074] In Embodiment 1, the inner diameter SRi of the first twist pipe 41 in the first gas cooler 4 is preferably equal to the inner diameter SRi of the second twist pipe 51 in the second gas cooler 5. If the second gas cooler 5 is placed near the first gas cooler 4, an upstream end of the first twist pipe 41 is connected to a downstream end of the second twist pipe 51. In this case, the inner diameter SRi of the first twist pipe 41 being equal to the inner diameter SRi of the second twist pipe 51 allows easy connection between the first twist pipe 41 and the second twist pipe 51. In addition, the inner diameter SRi of the first twist pipe 41 being equal to the inner diameter SRi of the second twist pipe 51 allows material and a manufacturing method used for the first twist pipe 41 and the second twist pipe 51 to be shared, thereby reducing cost.

[0075] In Embodiment 1, the number of the first refrigerant heat transfer pipe(s) 42, that is, the number of the first refrigerant heat transfer channel(s) in the first gas cooler 4 is preferably equal to the number of the second refrigerant heat transfer pipe(s) 52, that is, the number of the second refrigerant heat transfer channel(s) in the second gas cooler 5. The number of the first refrigerant heat transfer pipe(s) 42 being equal to the number of the second refrigerant heat transfer pipe(s) 52 allows the first twist pipe 41 and the second twist pipe 51 to be similarly designed, thereby reducing cost.

[0076] In Embodiment 1, the case where the first heat exchanger (first gas cooler 4) and the second heat exchanger (second gas cooler 5) are the twist pipe type heat exchangers has been described as an example. However, in the present invention, the first heat exchanger and the second heat exchanger are not limited to the twist pipe type heat exchanger, but various types of heat exchangers may be used.

[0077] As described above, the value of the inner diameter ratio di1/di2 of the first refrigerant heat transfer pipe 42 and the second refrigerant heat transfer pipe 52 is preferably not less than 1.1 and not more than 1.4, and more preferably not less than 1.2 and not more than 1.4. If the inner diameter ratio di1/di2 is 1.1, the ratio of the total sectional area of the first refrigerant heat transfer channels in the first heat exchanger to the total sectional area of the second refrigerant heat transfer channels in the second heat exchanger is $(11.1)^2 \approx 1.2$. If the inner diameter ratio di1/di2 is 1.2, the ratio of the total sectional area of the first refrigerant heat transfer channels in the first heat exchanger to the total sectional area of the second refrigerant heat transfer channels in the second heat exchanger is $(1.2)^2 \approx 1.4$. If the inner diameter ratio di1/di2 is 1.4, the ratio of the total sectional area of the first refrigerant heat transfer channels in the first heat exchanger to the total sectional area of the second refrigerant heat transfer channels in the second heat exchanger is $(1.4)^2 \approx 2$. Thus, if a numerical range of the inner diameter ratio di1/di2 is replaced by a numerical range of the ratio of the channel sectional area, it can be said that the ratio of the total sectional area of the first refrigerant heat transfer channels to the total sectional area of the second refrigerant heat transfer channels is preferably not less than 1.2 and not more than 2, and more preferably not less than 1.4 and not more than 2. The ratio of the channel sectional area within such a range provides advantages similar to those described above.

[0078] In Embodiment 1 described above, the case where the number of the first refrigerant heat transfer channels in the first heat exchanger (first gas cooler 4) is equal to the number of the second refrigerant heat transfer channels in the second heat exchanger (second gas cooler 5) has been mainly described, however, in the present invention, the number of the first refrigerant heat transfer channels may be larger than the number of the second refrigerant heat transfer channel(s). If the number of the first refrigerant heat transfer channels is larger than the number of the second refrigerant heat transfer channel(s), the total sectional area of the first refrigerant heat transfer channels can be larger than the total sectional area of the second refrigerant heat transfer channels with a simple configuration. If the number of the first refrigerant heat transfer channels is larger than the number of the second refrigerant heat transfer channel(s), for example, the sectional area of the first refrigerant heat transfer channel may be equal to the sectional area of the second refrigerant heat transfer channel. This allows the first refrigerant heat transfer pipe 42 in the first gas cooler 4 and the second refrigerant heat transfer pipe 52 in the second gas cooler 5 to be made of a common material, thereby reducing cost.

[0079] In Embodiment 1, the heat pump device for heating water using the first heat exchanger and the second heat exchanger has been described as an example, but in the present invention, the liquid heated by the first heat exchanger and the second heat exchanger is not limited to water, but for example, may be brine, antifreeze liquid, or the like.

Reference Signs List

[0080]

1 heat pump device
1a water inlet
1b water outlet
2 tank unit
2a hot water storage tank
2b water pump
2c hot-water supplying mixing valve
3 compressor
4 first gas cooler
5 second gas cooler
6 expansion valve
7 evaporator
8 fan
9 high and low pressure heat exchanger
10, 11, 12, 17, 18, 19, 20, 21 pipe
13 water supply pipe
14 hot water delivery pipe
15 water supply branch pipe
16 hot-water supply pipe
23, 26 water channel
31 sealed container
32 compressing element
33 electric actuating element
34 first intake passage
35 first discharge passage
36 second intake passage
37 second discharge passage
38 internal space
41 first twist pipe
42, 42a, 42b, 42c first refrigerant heat transfer pipe
50 control unit
51 second twist pipe
52 second refrigerant heat transfer pipe
60 heat transfer material
70 heat pump device
71 compressor
72 gas cooler
331 rotor
332 stator

411, 411a, 411b, 411c, 511 groove

Claims

1. A heat pump device comprising:

a compressor including a first discharge passage for discharging refrigerant and refrigerator oil, and a second discharge passage for discharging the refrigerant and the refrigerator oil, a mass flow rate of the refrigerator oil discharged from the first discharge passage being higher than a mass flow rate of the refrigerator oil discharged from the second discharge passage;
a first heat exchanger including one or a plurality of first refrigerant heat transfer channels through which the refrigerant and the refrigerator oil discharged from the first discharge passage pass, and one or a plurality of first liquid heat transfer channels through which a liquid passes, heat exchange being performed between the first refrigerant heat transfer channel and the first liquid heat transfer channel; and
a second heat exchanger including one or a plurality of second refrigerant heat transfer channels through which the refrigerant and the refrigerator oil discharged from the second discharge passage pass, and one or a plurality of second liquid heat transfer channels through which the liquid passes, heat exchange being performed between the second refrigerant heat transfer channel and the second liquid heat transfer channel,
a total sectional area of the first refrigerant heat transfer channel(s) being larger than a total sectional area of the second refrigerant heat transfer channel(s).

2. The heat pump device according to claim 1, wherein the number of the first refrigerant heat transfer channel(s) is equal to the number of the second refrigerant heat transfer channel(s), and
a sectional area of each of the first refrigerant heat transfer channel(s) is larger than a sectional area of each of the second refrigerant heat transfer channel(s).

3. The heat pump device according to claim 1 or 2, wherein the first refrigerant heat transfer channel is formed of a first refrigerant heat transfer pipe,
the second refrigerant heat transfer channel is formed of a second refrigerant heat transfer pipe,
the first liquid heat transfer channel is formed of a first twist pipe having a helical groove in an outer periphery,
the second liquid heat transfer channel is formed of a second twist pipe having a helical groove in an outer periphery,
the first refrigerant heat transfer pipe is located along the groove in the first twist pipe, and
the second refrigerant heat transfer pipe is located along the groove in the second twist pipe.

4. The heat pump device according to claim 3, wherein a ratio of an inner diameter of the first refrigerant heat transfer pipe to an inner diameter of the second refrigerant heat transfer pipe is not less than 1.1 and not more than 1.4.

5. The heat pump device according to claim 3 or 4, wherein a twist pitch of the first twist pipe is larger than a twist pitch of the second twist pipe.

6. The heat pump device according to any one of claims 3 to 5, wherein when p is a twist pitch of the first twist pipe and SRi is an inner diameter of the first twist pipe, p/SRi is not less than 1.1 and not more than 1.8.

7. The heat pump device according to any one of claims 3 to 6, wherein an inner diameter of the first twist pipe is equal to an inner diameter of the second twist pipe.

8. The heat pump device according to any one of claims 3 to 7, wherein the number of the first refrigerant heat transfer channel(s) is larger than the number of the second refrigerant heat transfer channel(s).

9. The heat pump device according to any one of claims 1 to 8, wherein a ratio of the total sectional area of the first refrigerant heat transfer channel(s) to the total sectional area of the second refrigerant heat transfer channel(s) is not less than 1.2 and not more than 2.

10. The heat pump device according to any one of claims 1 to 9, wherein a ratio of the mass flow rate of the refrigerator oil with respect to a sum of the mass flow rate of the refrigerant and the mass flow rate of the refrigerator oil in the first heat exchanger is not less than 2% and not more than 20%.

11. The heat pump device according to any one of claims 1 to 10, wherein a ratio of the mass flow rate of the refrigerator oil with respect to a sum of the mass flow rate of the refrigerant and the mass flow rate of the refrigerator oil in the second heat exchanger is not less than 0.01% and not more than 1%.

12. The heat pump device according to any one of claims 1 to 11, wherein the liquid is water, and the heat pump device has a function of supplying hot water obtained by heating the water.

FIG. 1

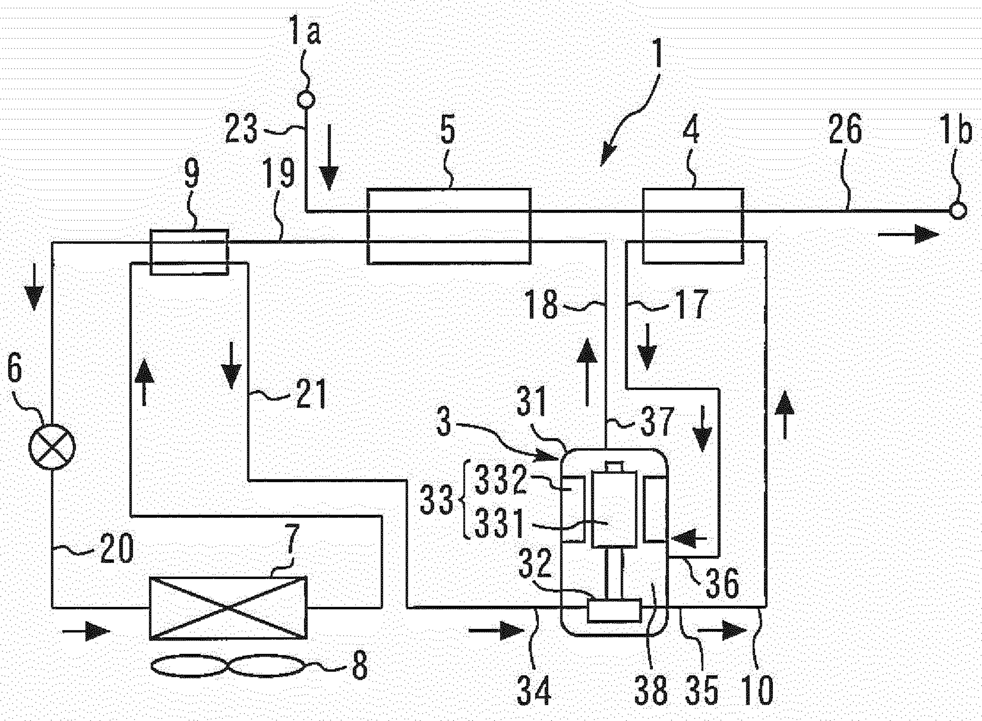

1a
1
23
5
4
26
1b
9
19
18
17
6
21
3
31
37
33
332
331
20
7
32
36
8
34
38
35
10

FIG. 2

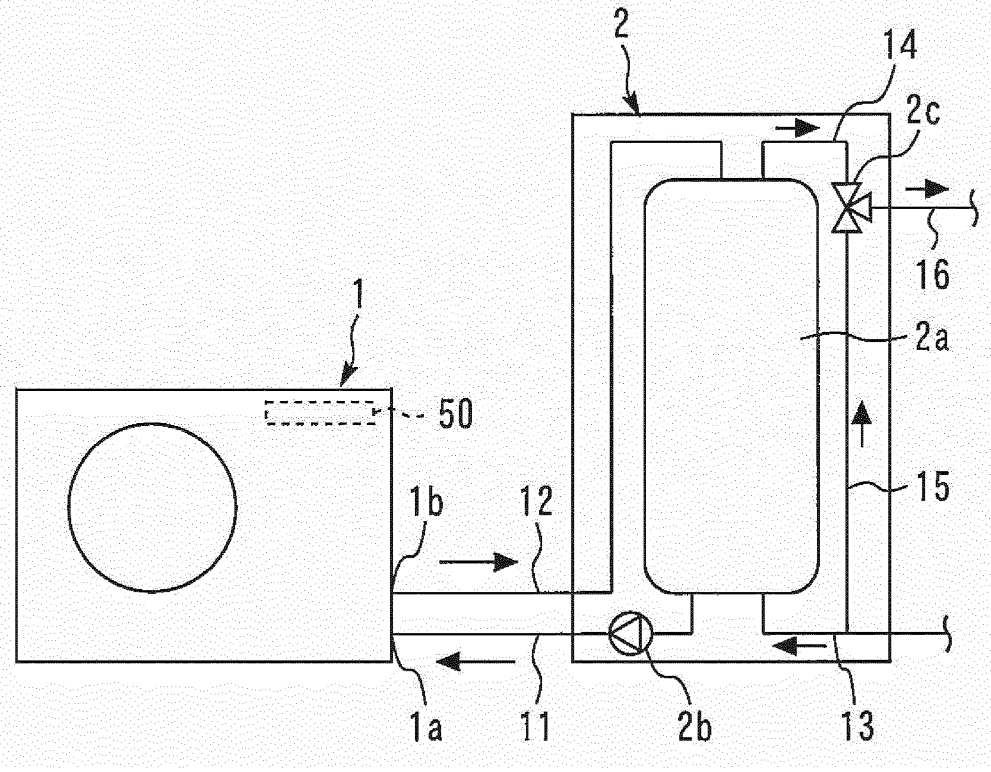

2
14
2c
16
1
2a
50
1b
12
15
1a
11
2b
13

FIG. 3

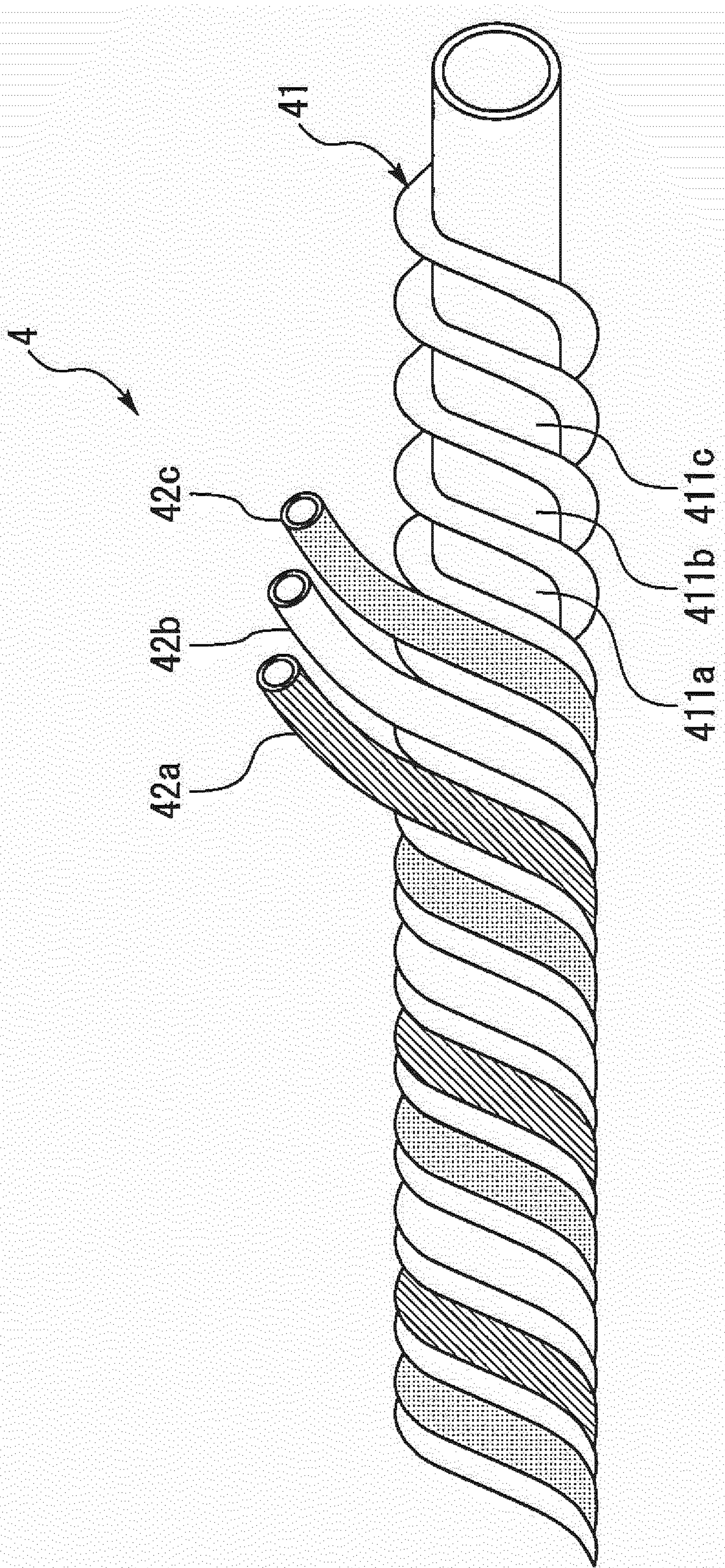


FIG. 4

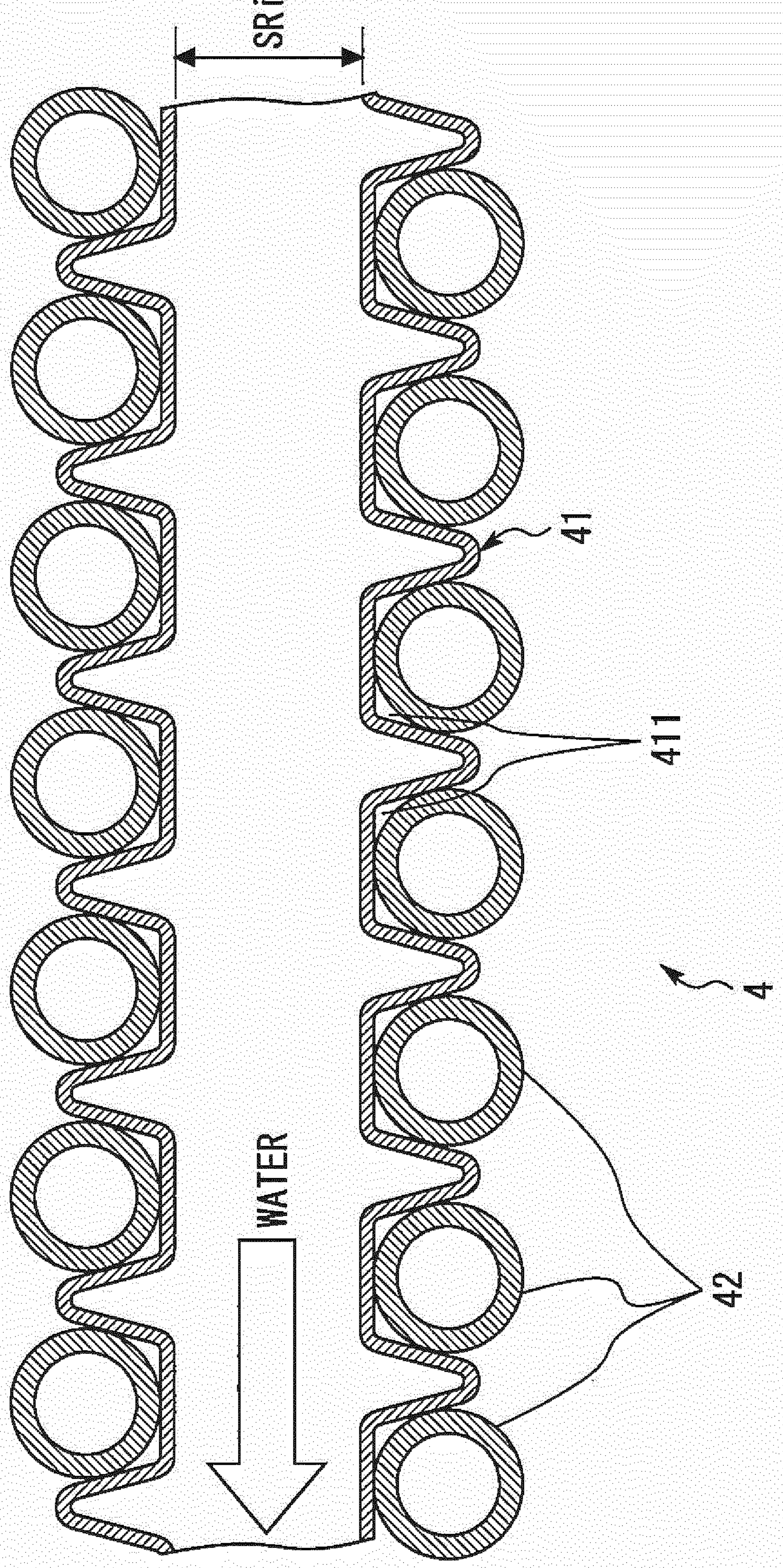


FIG. 5

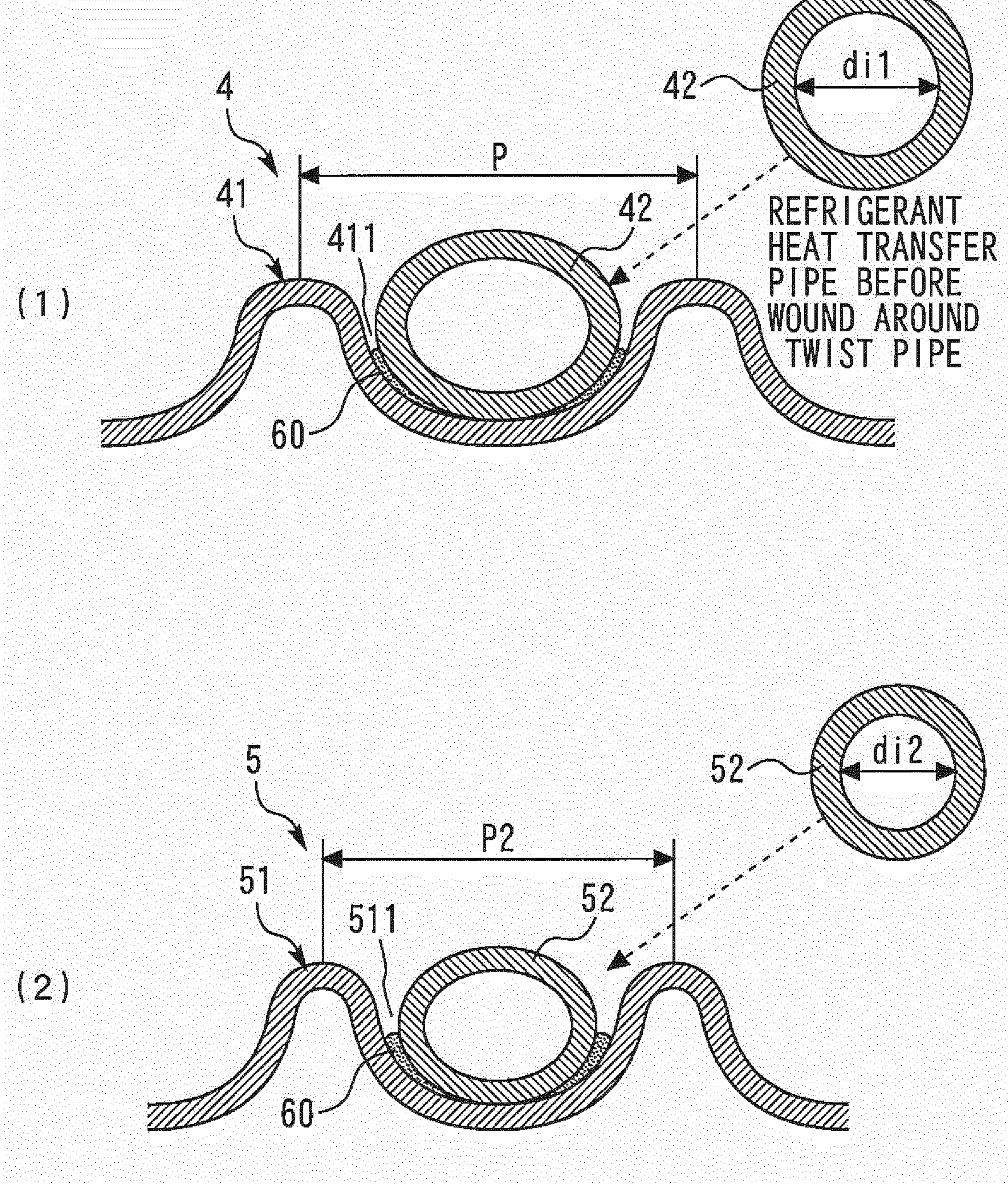

(1)
4
41
411
P
42
di1
42
60
REFRIGERANT HEAT TRANSFER PIPE BEFORE WOUND AROUND TWIST PIPE
(2)
5
51
511
P2
52
di2
52
60

FIG. 6

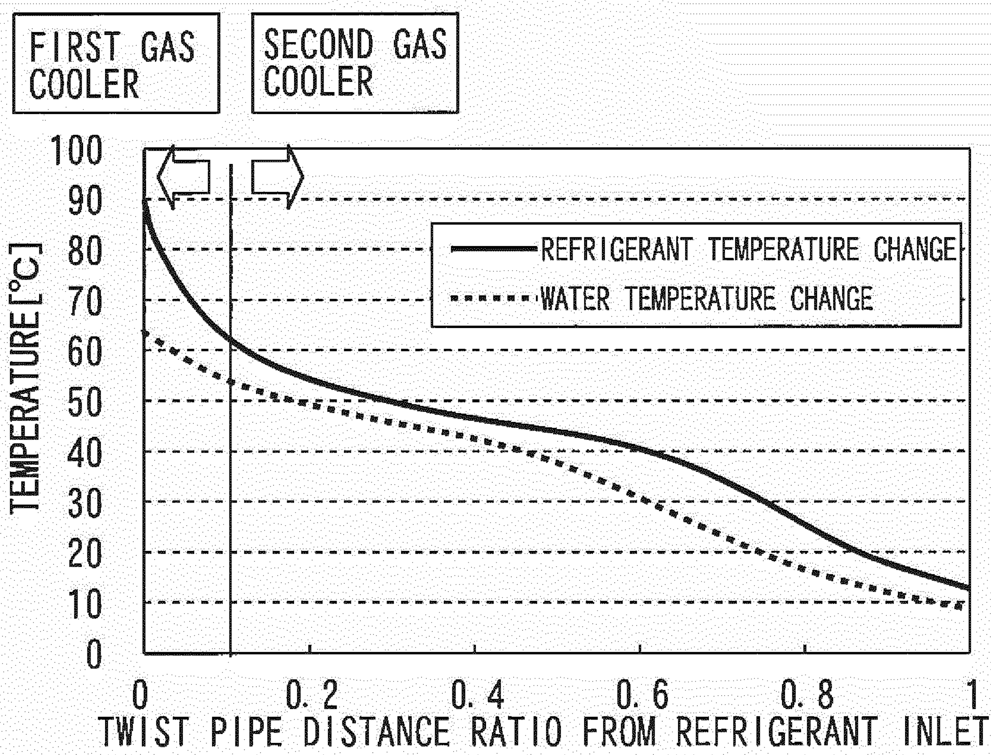

FIRST GAS COOLER
SECOND GAS COOLER
TEMPERATURE [°C]
100
90
80
70
60
50
40
30
20
10
0
REFRIGERANT TEMPERATURE CHANGE
WATER TEMPERATURE CHANGE
0
0. 2
0. 4
0. 6
0. 8
1
TWIST PIPE DISTANCE RATIO FROM REFRIGERANT INLET

FIG. 7

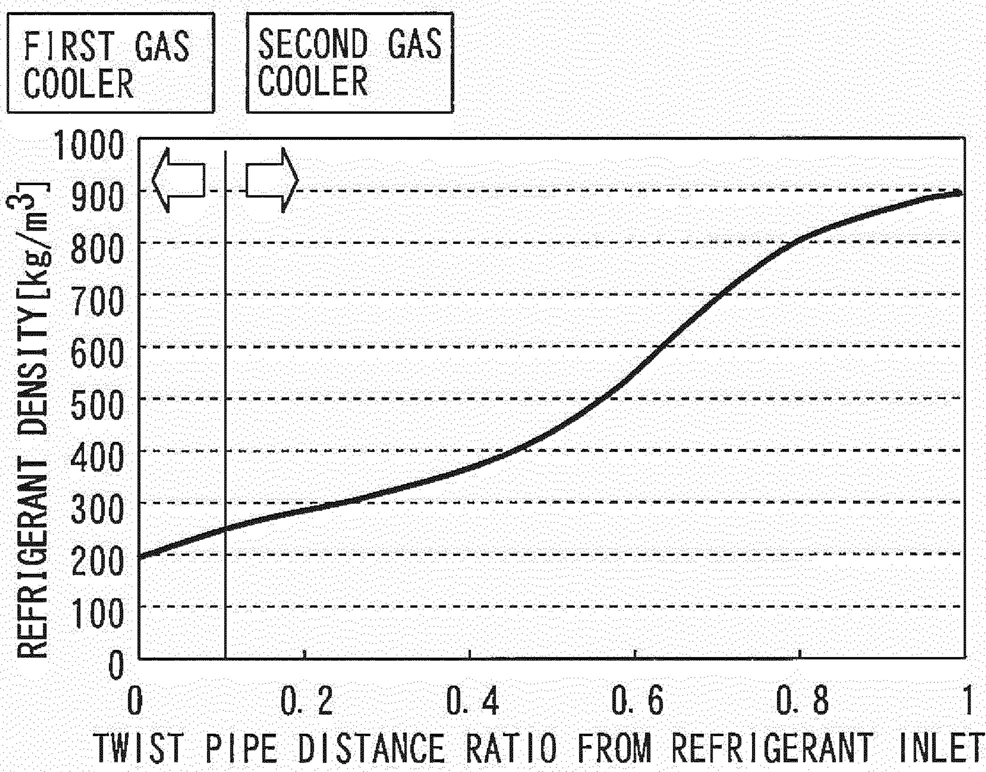

FIRST GAS COOLER
SECOND GAS COOLER
REFRIGERANT DENSITY [kg/m3]
1000
900
800
700
600
500
400
300
200
100
0
0
0. 2
0. 4
0. 6
0. 8
1
TWIST PIPE DISTANCE RATIO FROM REFRIGERANT INLET

FIG. 8

OIL CIRCULATION RATE	0.5% OR LESS IN OVERALL GAS COOLER(S)	OIL CIRCULATION RATE IS HIGH IN FIRST GAS COOLER AND 0.5% OR LESS IN SECOND GAS COOLER
RATIO OF REFRIGERANT PRESSURE LOSS OF OVERALL GAS COOLER(S)	1.00	1.17
RATIO OF REFRIGERANT PRESSURE LOSS OF FIRST GAS COOLER (LENGTH OF 10% ON HIGH TEMPERATURE SIDE)	0.17	0.34 (29% OF OVERALL GAS COOLERS)
RATIO OF REFRIGERANT PRESSURE LOSS OF FIRST GAS COOLER (LENGTH OF 90% ON LOW TEMPERATURE SIDE)	0.83	0.83

FIG. 9

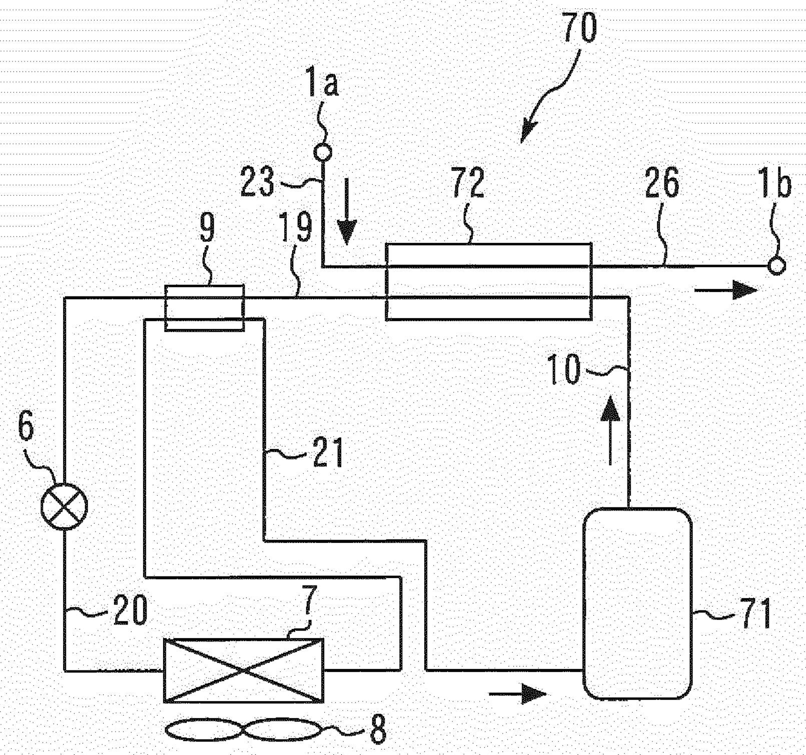

70
1a
23
72
26
1b
9
19
10
6
21
20
7
71
8

FIG. 10

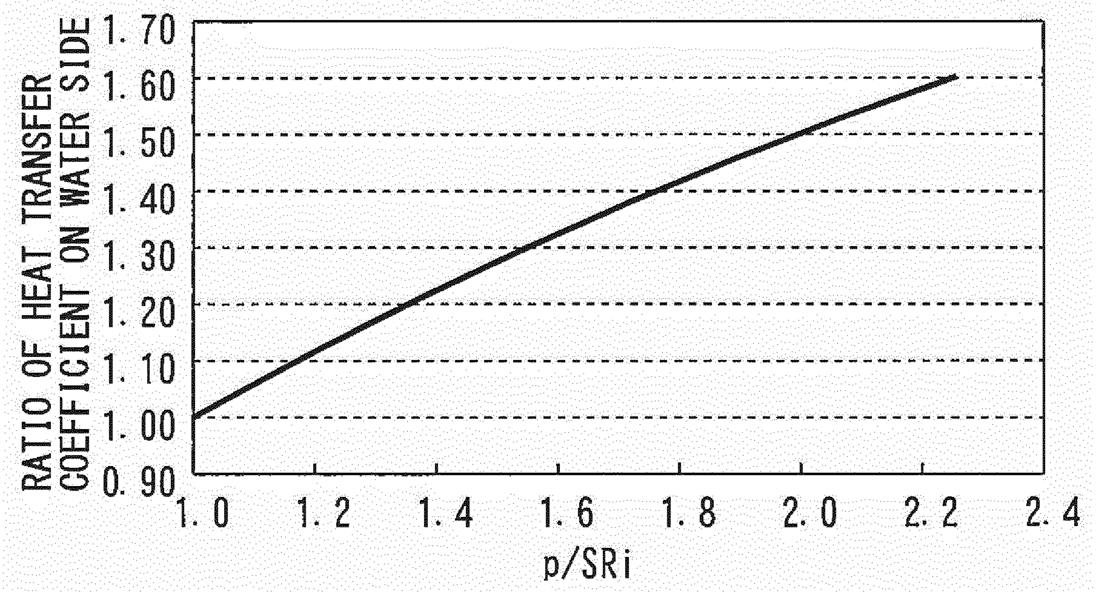

RATIO OF HEAT TRANSFER COEFFICIENT ON WATER SIDE
1.70
1.60
1.50
1.40
1.30
1.20
1.10
1.00
0.90
1.0
1.2
1.4
1.6
1.8
2.0
2.2
2.4
p/SRi

FIG. 11

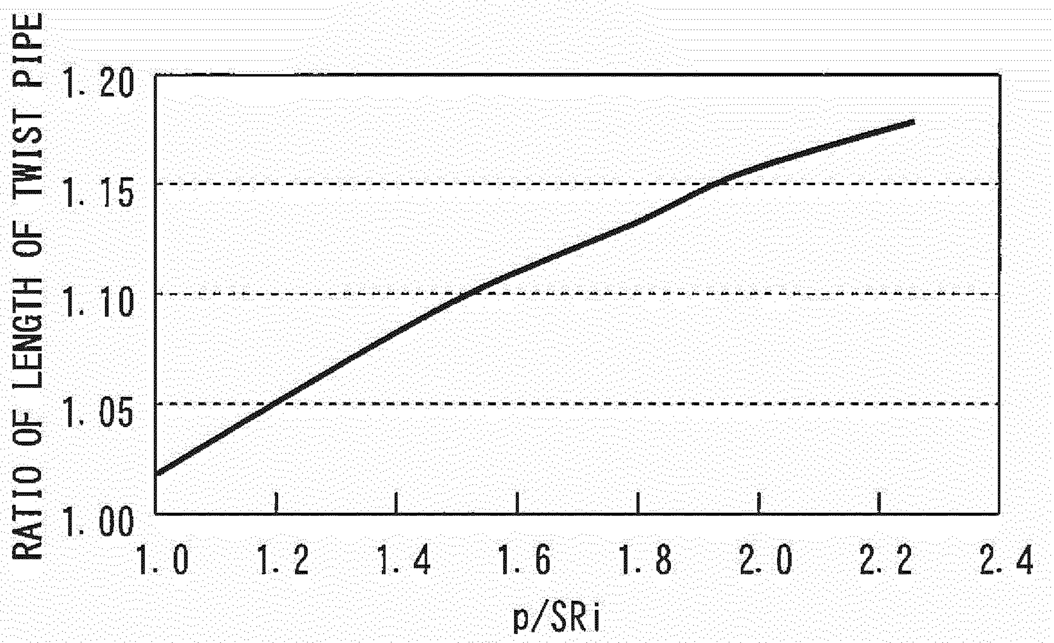

RATIO OF LENGTH OF TWIST PIPE
1. 20
1. 15
1. 10
1. 05
1. 00
1. 0
1. 2
1. 4
1. 6
1. 8
2. 0
2. 2
2. 4
p/SRi

FIG. 12

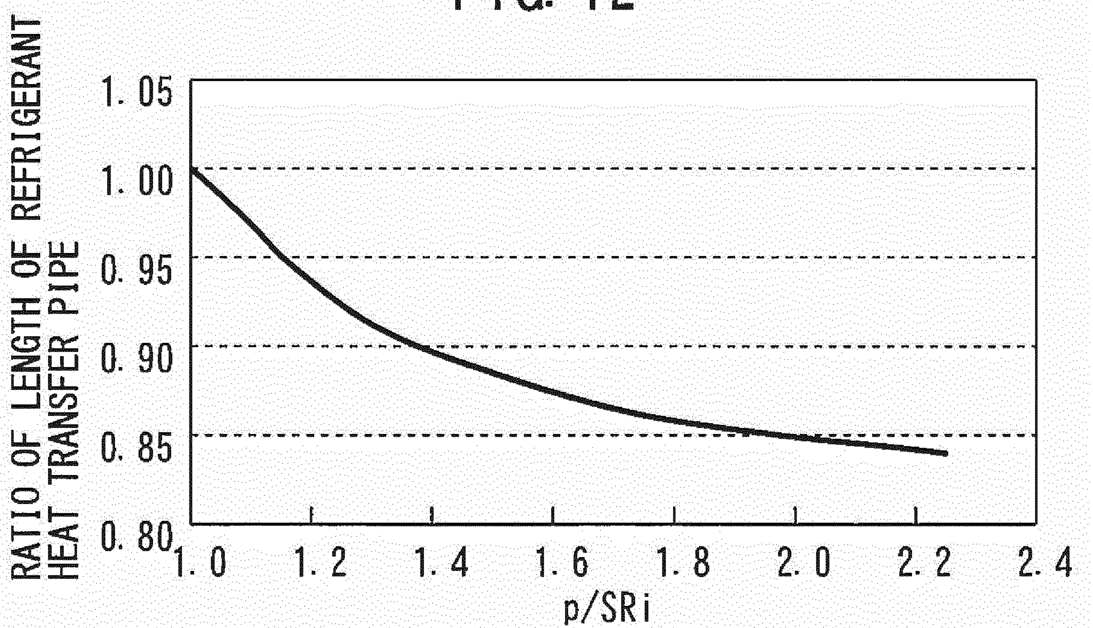

RATIO OF LENGTH OF REFRIGERANT HEAT TRANSFER PIPE
1. 05
1. 00
0. 95
0. 90
0. 85
0. 80
1. 0
1. 2
1. 4
1. 6
1. 8
2. 0
2. 2
2. 4
p/SRi

FIG. 13

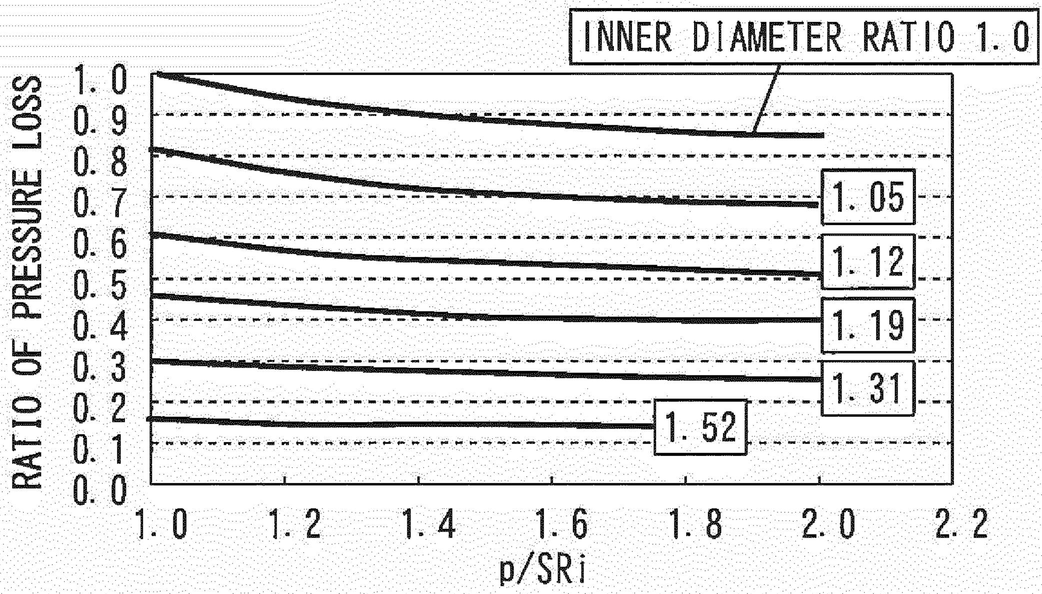

INNER DIAMETER RATIO 1. 0
1. 05
1. 12
1. 19
1. 31
1. 52
RATIO OF PRESSURE LOSS
1. 0
0. 9
0. 8
0. 7
0. 6
0. 5
0. 4
0. 3
0. 2
0. 1
0. 0
1. 0
1. 2
1. 4
1. 6
1. 8
2. 0
2. 2
p/SRi

FIG. 14

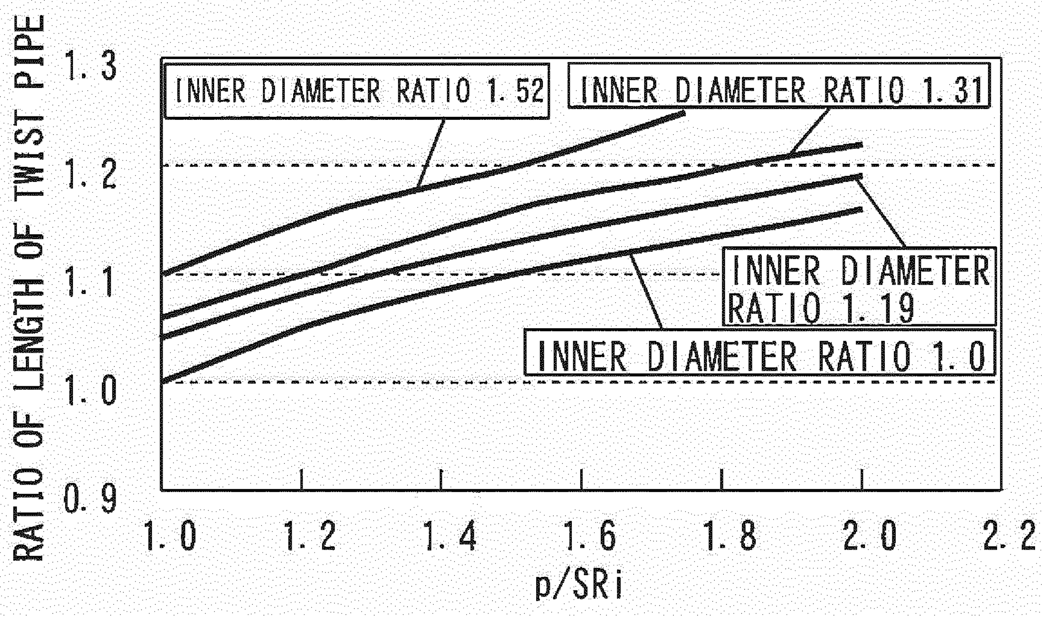

INNER DIAMETER RATIO 1. 52
INNER DIAMETER RATIO 1. 31
INNER DIAMETER RATIO 1. 19
INNER DIAMETER RATIO 1. 0
RATIO OF LENGTH OF TWIST PIPE
1. 3
1. 2
1. 1
1. 0
0. 9
1. 0
1. 2
1. 4
1. 6
1. 8
2. 0
2. 2
p/SRi

FIG. 15

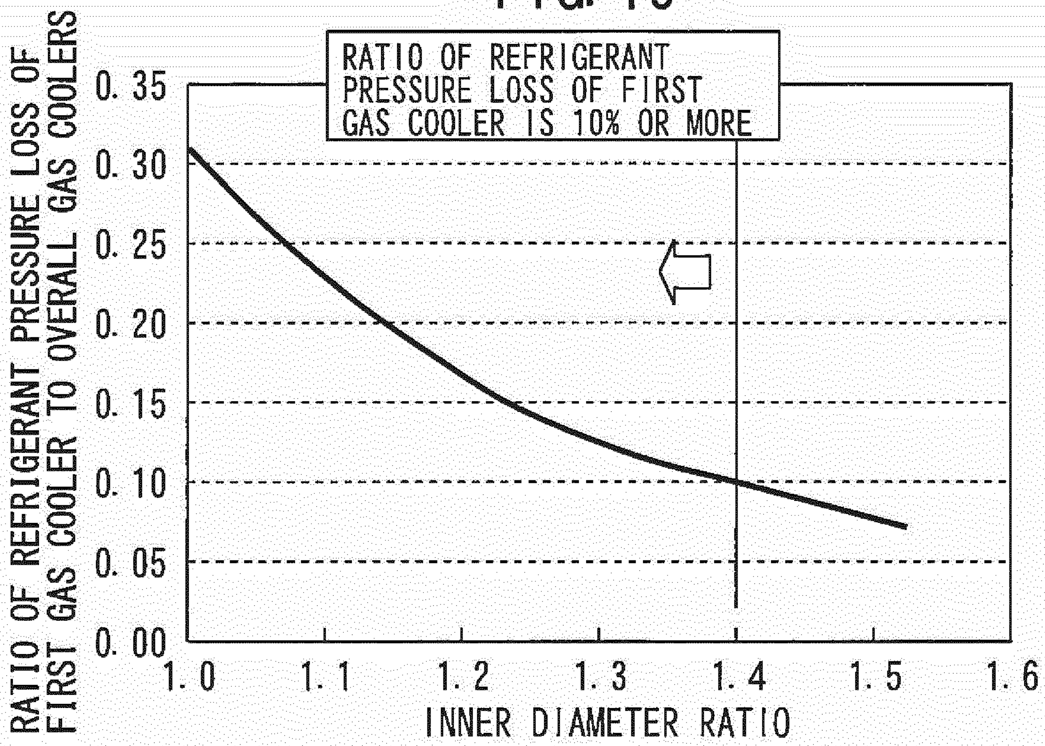
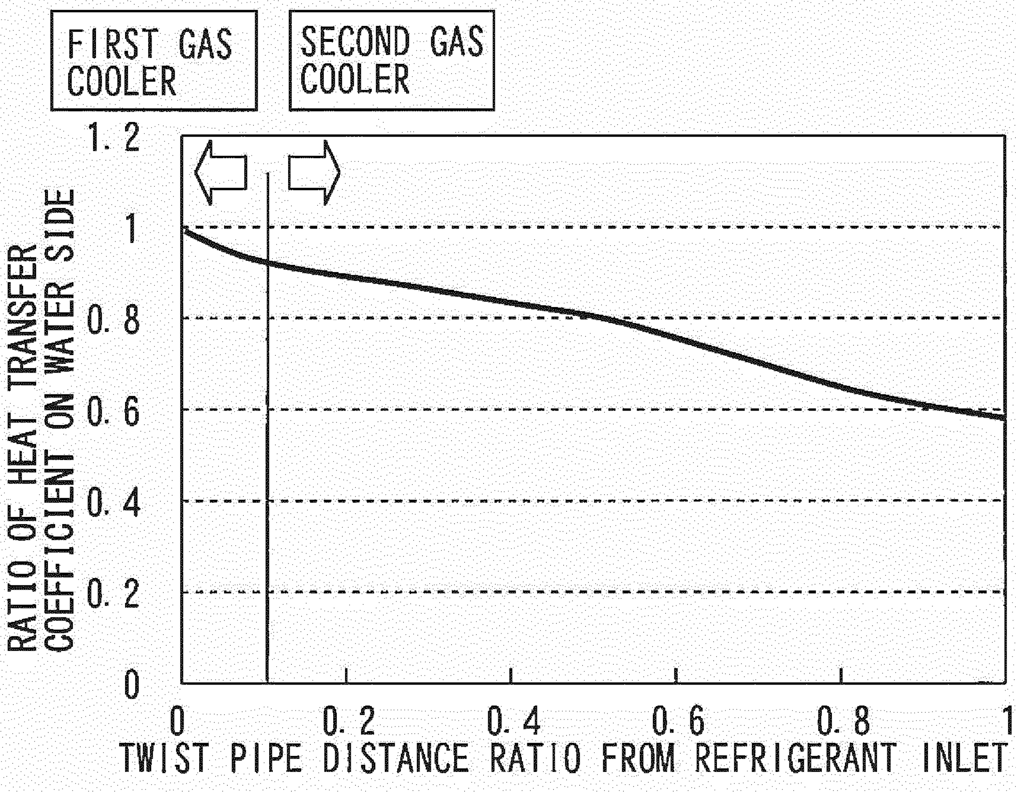

RATIO OF REFRIGERANT PRESSURE LOSS OF FIRST GAS COOLER IS 10% OR MORE
RATIO OF REFRIGERANT PRESSURE LOSS OF FIRST GAS COOLER TO OVERALL GAS COOLERS
0. 35
0. 30
0. 25
0. 20
0. 15
0. 10
0. 05
0. 00
1. 0
1. 1
1. 2
1. 3
1. 4
1. 5
1. 6
INNER DIAMETER RATIO

FIG. 16


FIRST GAS COOLER
SECOND GAS COOLER
RATIO OF HEAT TRANSFER COEFFICIENT ON WATER SIDE
1. 2
1
0. 8
0. 6
0. 4
0. 2
0
0
0. 2
0. 4
0. 6
0. 8
1
TWIST PIPE DISTANCE RATIO FROM REFRIGERANT INLET

INTERNATIONAL SEARCH REPORT

International application No. PCT/JP2013/066313

A. CLASSIFICATION OF SUBJECT MATTER
F25B6/04(2006.01)i, *F04B39/06*(2006.01)i, *F04C29/04*(2006.01)i, *F28D7/02*(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)
F25B6/04, F04B39/06, F04C29/04, F28D7/02

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

Jitsuyo Shinan Koho	1922-1996	Jitsuyo Shinan Toroku Koho	1996-2013
Kokai Jitsuyo Shinan Koho	1971-2013	Toroku Jitsuyo Shinan Koho	1994-2013

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	JP 2006-132427 A (Mitsubishi Electric Corp.), 25 May 2006 (25.05.2006), paragraphs [0013] to [0028]; fig. 1 to 5 (Family: none)	1-12
Y	JP 2009-168383 A (Hitachi Appliances, Inc.), 30 July 2009 (30.07.2009), paragraphs [0018] to [0026]; fig. 1 to 8 (Family: none)	1-12
Y	JP 2013-88045 A (Hitachi Appliances, Inc.), 13 May 2013 (13.05.2013), paragraphs [0017] to [0040]; fig. 1 to 9 (Family: none)	3-8

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

* Special categories of cited documents:
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"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
09 September, 2013 (09.09.13)	17 September, 2013 (17.09.13)

Name and mailing address of the ISA/	Authorized officer
Japanese Patent Office	
Facsimile No.	Telephone No.

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INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2013/066313

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	JP 2006-90697 A (Mitsubishi Electric Corp.), 06 April 2006 (06.04.2006), paragraphs [0013] to [0027]; fig. 1 to 12 (Family: none)	3-8
Y	JP 2009-47394 A (Mitsubishi Electric Corp.), 05 March 2009 (05.03.2009), paragraphs [0012] to [0017]; fig. 1 to 3 (Family: none)	3-8
Y	JP 2004-205077 A (Sapporo Holdings Ltd.), 22 July 2004 (22.07.2004), paragraphs [0011] to [0019] (Family: none)	3-8
Y	JP 2004-85166 A (Pacific Engineering Corp.), 18 March 2004 (18.03.2004), paragraph [0003] (Family: none)	3-8
A	JP 2004-108616 A (Mayekawa Mfg., Co., Ltd.), 08 April 2004 (08.04.2004), entire text; all drawings (Family: none)	1-12
A	JP 2008-309361 A (Panasonic Corp.), 25 December 2008 (25.12.2008), entire text; all drawings (Family: none)	1-12
A	Microfilm of the specification and drawings annexed to the request of Japanese Utility Model Application No. 091363/1971(Laid-open No. 47010/1973) (Tokyo Shibaura Electric Co., Ltd.), 20 June 1973 (20.06.1973), entire text; all drawings (Family: none)	1-12
A	Microfilm of the specification and drawings annexed to the request of Japanese Utility Model Application No. 110697/1970(Laid-open No. 2325/1976) (Sanyo Electric Co., Ltd.), 23 January 1976 (23.01.1976), entire text; all drawings (Family: none)	1-12

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Patent documents cited in the description

- JP 2006132427 A **[0003]**
- JP 2004108616 A **[0003]**
- JP 2008309361 A **[0003]**
- JP 2009168383 A **[0003]**