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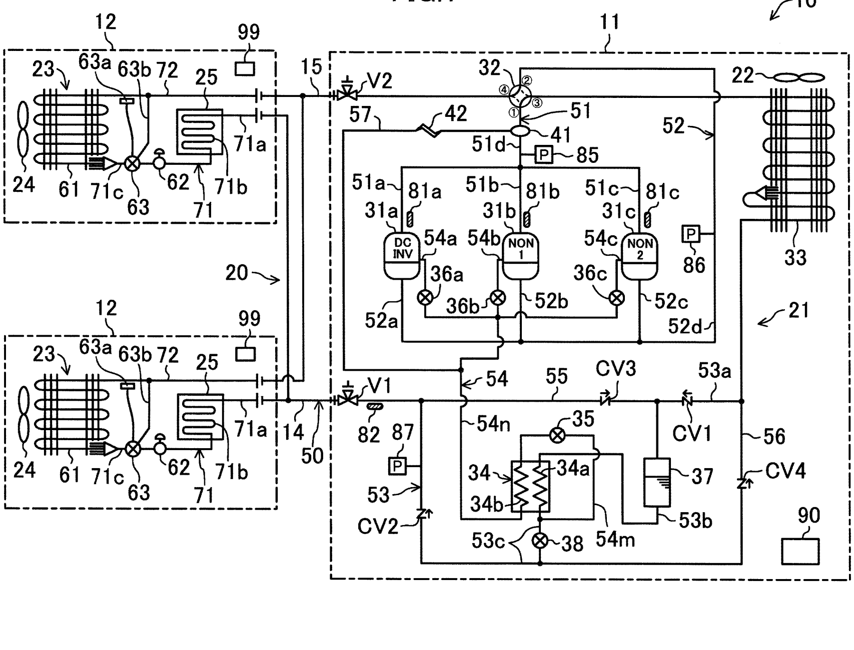
(54) **REFRIGERATION APPARATUS**

(57) A refrigeration apparatus (10) includes a plurality of utilization-side units (12) connected to a heat-source-side unit (11) through a pair of interconnecting pipes (14, 15). While one or some of the utilization-side units (12) are in a suspended state, a controller (90) performs a pressure control operation. The pressure control operation is performed to control the degree of opening of a heat-source-side expansion valve (38) to prevent the pressure of a refrigerant in the liquid interconnecting pipe (14) from exceeding a predetermined upper limit. The controller (90) performs the pressure control operation to prevent a liquid hammer phenomenon caused by opening solenoid valves of the utilization-side units (12).

FIG.1



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Description

TECHNICAL FIELD

[0001] The present invention relates to a refrigeration apparatus that circulates a refrigerant through a refrigerant circuit to perform a refrigeration cycle.

BACKGROUND ART

[0002] A refrigerant circuit of a refrigeration apparatus that performs a refrigeration cycle may be provided with a solenoid valve to control the flow of a refrigerant. A general solenoid valve interrupts the passage of electric current through a solenoid to switch between an open state and a closed state.

[0003] The refrigerant circuit of the refrigeration apparatus includes a pipe through which a high-pressure liquid refrigerant flows. The pipe may be provided with a solenoid valve. While this solenoid valve is closed, the flow of the high-pressure liquid refrigerant is interrupted by the solenoid valve. If the solenoid valve opens in a situation where the difference in pressure between both sides of the solenoid valve is large, a substantially incompressible liquid refrigerant having a relatively high density suddenly flows downstream of the solenoid valve, resulting in a liquid hammer phenomenon. This may cause a break in a pipe, an expansion valve, or any other component.

[0004] Patent Document 1 discloses that in order to prevent a liquid hammer phenomenon caused by opening a solenoid valve, a pipe through which a liquid refrigerant flows is heated with an electric heater. Specifically, heating the pipe with the electric heater allows part of the refrigerant in the pipe to evaporate, thereby producing a compressible gaseous refrigerant in the pipe. This reduces the degree to which the internal pressure of the pipe suddenly increases due to the opening of the solenoid valve.

CITATION LIST

PATENT DOCUMENT

[0005] [Patent Document 1] Japanese Unexamined Patent Publication No. H11-325654

SUMMARY OF THE INVENTION

TECHNICAL PROBLEM

[0006] The refrigeration apparatus of Patent Document 1 described above needs to include the electric heater for heating the pipe, in order to prevent the liquid hammer phenomenon caused by opening the solenoid valve. This increases the number of parts of the refrigeration apparatus, resulting in an increase in manufacturing cost. While the solenoid valve is closed, the electric heater needs to keep heating the pipe. This may increase power consumed by the refrigeration apparatus, and may increase the running cost of the refrigeration apparatus.

[0007] In view of the foregoing background, it is therefore an object of the present invention to reduce the degree to which the manufacturing or running cost of a refrigeration apparatus increases, and prevent a liquid hammer phenomenon caused by opening a solenoid valve.

SOLUTION TO THE PROBLEM

[0008] A first aspect of the present disclosure is directed to a refrigeration apparatus including: a refrigerant circuit (20) including a heat-source-side unit (11) and a plurality of utilization-side units (12), the heat-source-side unit (11) and the utilization-side units (12) being connected together through a liquid interconnecting pipe (14) and a gas interconnecting pipe (15), the utilization-side units (12) being arranged in parallel. The refrigeration apparatus circulates a refrigerant through the refrigerant circuit (20) to perform a refrigeration cycle. The heat-source-side unit (11) includes a compressor (31a-31c), a heat-source-side heat exchanger (33), and a heat-source-side expansion valve (38) provided on a pipe (53c) configured to deliver the refrigerant condensed in the heat-source-side heat exchanger (33) to the liquid interconnecting pipe (14). Each of the utilization-side units (12) includes a utilization-side heat exchanger (61), a utilization-side expansion valve (63), and a utilization-side solenoid valve (62), which are arranged in series, and is switchable between a cooling state where the utilization-side solenoid valve (62) opens to allow the utilization-side heat exchanger (61) to function as an evaporator, and a suspended state where the utilization-side solenoid valve (62) closes to interrupt flow of the refrigerant through the utilization-side heat exchanger (61). The refrigeration apparatus further includes: a controller (90) configured to perform a pressure control operation to control a degree of opening of the heat-source-side expansion valve (38) to prevent a pressure of the refrigerant in the liquid interconnecting pipe (14) from exceeding a predetermined upper-limit value while one or some of the utilization-side units (12) are in the suspended state.

[0009] According to the first aspect, the heat-source-side unit (11) and the utilization-side units (12) are provided for the refrigerant circuit (20). The refrigerant condensed in the heat-source-side heat exchanger (33) of the heat-source-side unit (11) flows through the liquid interconnecting pipe (14) into the utilization-side units (12). The refrigerant supplied from the liquid interconnecting pipe (14) to each utilization-side unit (12) expands when passing through the associated utilization-side expansion valve (63), and then flows into the associated utilization-side heat exchanger (61) to evaporate. In the utilization-side heat exchanger (61), a target to be cooled, such as air, is cooled by the refrigerant. The refrigerant evaporated in the utilization-side heat exchanger (61) of each utilization-side unit (12) flows through the gas interconnecting pipe (15) into the heat-source-side unit (11), and is then sucked into the compressor (31a-31c) to be compressed.

[0010] In the refrigeration apparatus (10) of the first aspect, one or some of the utilization-side units (12) may turn to the suspended state. In this case, the other utilization-side unit(s) (12) is in the cooling state. Thus, the compressor (31a-31c) of the heat-source-side unit (11) keeps operating. In the utilization-side unit(s) (12) in the suspended state, the liquid refrigerant delivered from the liquid interconnecting pipe (14) to the utilization-side unit(s) (12) resides near one end of the closed utilization-side solenoid valve(s) (62). In the utilization-side unit(s) (12) in the suspended state, the pressure of the refrigerant near the one end of the closed utilization-side solenoid valve(s) (62) is substantially equal to the pressure of the refrigerant in the liquid interconnecting pipe (14), and the pressure of the refrigerant near the other end thereof is substantially equal to the pressure of the refrigerant in the gas interconnecting pipe (15) communicating with the utilization-side heat exchangers (61).

[0011] If one or some of the utilization-side units (12) are in the suspended state, the controller (90) of the first aspect performs the pressure control operation. In the pressure control operation, the controller (90) controls the degree of opening of the heat-source-side expansion valve (38) to prevent the pressure of the refrigerant in the liquid interconnecting pipe (14) from exceeding a predetermined upper limit. Thus, while the utilization-side unit(s) (12) is in the suspended state, the pressure of the refrigerant in the liquid interconnecting pipe (14) is kept at values substantially equal to or lower than the upper limit.

[0012] According to a second aspect of the present disclosure which is an embodiment of the first aspect, while one or some of the utilization-side units (12) are in the suspended state, and a difference in pressure between the refrigerant in the liquid interconnecting pipe (14) and the refrigerant in the gas interconnecting pipe (15) is larger than or equal to a predetermined upper-limit pressure difference, the controller (90) may perform the pressure control operation.

[0013] If one or some of the utilization-side units (12) turn to the suspended state during the pressure control operation, and the difference in pressure between the refrigerant in the liquid interconnecting pipe (14) and the refrigerant in the gas interconnecting pipe (15) is larger than or equal to a predetermined upper-limit pressure difference, the controller (90) of the second aspect performs the pressure control operation.

[0014] According to a third aspect of the present disclosure which is an embodiment of the second aspect, the controller (90) that is performing the pressure control operation may adjust the degree of opening of the heat-source-side expansion valve (38) such that the difference in pressure between the refrigerant in the liquid interconnecting pipe (14) and the refrigerant in the gas interconnecting pipe (15) is larger than or equal to a lower-limit pressure difference that is smaller than the upper-limit pressure difference.

[0015] According to the third aspect, the difference in pressure between the refrigerant in the liquid interconnecting pipe (14) and the refrigerant in the gas interconnecting pipe (15) during the pressure control operation of the controller (90) is kept at values greater than or equal to the lower-limit pressure difference.

[0016] According to a fourth aspect of the present disclosure which is an embodiment of any one of the first through third aspects, the heat-source-side unit (11) may include a subcooling heat exchanger (34) configured to exchange heat between a liquid refrigerant condensed in the heat-source-side heat exchanger (33) and delivered to the liquid interconnecting pipe (14) and a cooling fluid to cool the liquid refrigerant.

[0017] In the heat-source-side unit (11) of the fourth aspect, the refrigerant condensed in the heat-source-side heat exchanger (33) is cooled by exchanging heat with the cooling fluid in the subcooling heat exchanger (34), and is then supplied to the liquid interconnecting pipe (14).

[0018] According to a fifth aspect of the present disclosure which is an embodiment of the fourth aspect, the heat-source-side unit (11) may include a subcooling pipe (54m) configured to supply part of the refrigerant condensed in the heat-source-side heat exchanger (33) as the cooling fluid to the subcooling heat exchanger (34), and a subcooling expansion valve (35) provided on the subcooling pipe (54m). The controller (90) may be configured to control a degree of opening of the subcooling expansion valve (35) to reduce a temperature of a liquid refrigerant delivered from the subcooling heat exchanger (34) to the liquid interconnecting pipe (14) if a difference in pressure between the refrigerant in the liquid interconnecting pipe (14) and the refrigerant in the gas interconnecting pipe (15) is smaller than a predetermined reference pressure difference.

[0019] According to the fifth aspect, part of the refrigerant condensed in the heat-source-side heat exchanger (33) flows into the subcooling pipe (54m). The refrigerant flowing through the subcooling pipe (54m) expands when passing through the subcooling expansion valve (35), and is then supplied, as the cooling fluid, to the heat-source-side heat exchanger (33). Changing the degree of opening of the subcooling expansion valve (35) triggers a change in the temperature of the refrigerant supplied, as the cooling fluid, through the subcooling pipe (54m) to the heat-source-side heat exchanger (33). This causes the temperature of the liquid refrigerant delivered from the subcooling heat exchanger (34) to the liquid interconnecting pipe (14) to vary.

[0020] The controller (90) of the fifth aspect controls the degree of opening of the subcooling expansion valve (35) to reduce the temperature of the liquid refrigerant delivered from the subcooling heat exchanger (34) to the liquid interconnecting pipe (14) if the difference in pressure between the refrigerant in the liquid interconnecting pipe (14) and the refrigerant in the gas interconnecting pipe (15) is smaller than the predetermined reference pressure difference. A reduction in the temperature of the liquid refrigerant delivered from the subcooling heat exchanger (34) to the liquid interconnecting pipe (14) increases the density of the liquid refrigerant.

ADVANTAGES OF THE INVENTION

[0021] According to the aspect of the present disclosure, while one or some of the utilization-side units (12) are in a suspended state, the controller (90) performs a pressure control operation. This reduces the pressure and density of a liquid refrigerant present near one end of the closed utilization-side solenoid valve (62) of a suspended one of the utilization-side units (12) to a low level. Thus, adjusting the degree of opening of the heat-source-side expansion valve (38) of the refrigeration apparatus (10) can prevent the liquid hammer phenomenon caused by the associated utilization-side solenoid valve (62) that has just switched from the closed state to the open state. Thus, this aspect can prevent the liquid hammer phenomenon without adding a new member to the refrigeration apparatus (10).

[0022] In the refrigeration apparatus (10) of the present disclosure described above, the degree of opening of the heat-source-side expansion valve (38) is controlled to prevent the liquid hammer phenomenon. For this reason, unlike a known situation where the use of an electric heater prevents the liquid hammer phenomenon, preventing the liquid hammer phenomenon hardly leads to an increase in the consumed power. Thus, this aspect can prevent the liquid hammer phenomenon while avoiding an increase in the running cost of the refrigeration apparatus (10).

[0023] The larger the difference in pressure between the refrigerant in the liquid interconnecting pipe (14) and the refrigerant in the gas interconnecting pipe (15) is, the more likely the liquid hammer phenomenon is to be caused by the associated utilization-side solenoid valve (62) that has just switched from the closed state to the open state.

[0024] To address this problem, according to the second aspect, if the difference in pressure between the refrigerant in the liquid interconnecting pipe (14) and the refrigerant in the gas interconnecting pipe (15) is larger than or equal to a predetermined upper-limit pressure difference, the controller (90) performs the pressure control operation. This allows the controller (90) to perform the pressure control operation in a state where the liquid hammer phenomenon is highly likely to occur.

[0025] While the controller (90) is performing the pressure control operation, one or some of the utilization-side units (12) are in the suspended state. In this state, the other utilization-side unit(s) (12) is in the cooling state. If the difference in pressure between the refrigerant in the liquid interconnecting pipe (14) and the refrigerant in the gas interconnecting pipe (15) is too small in this state, the mass flow rate of the refrigerant passing through the utilization-side expansion valve (63) of (each of) the utilization-side unit(s) (12) in the cooling state may become too low. This may prevent the utilization-side unit(s) (12) in the cooling state from having adequate cooling capability.

[0026] To address this problem, according to the third aspect, the difference in pressure between the refrigerant in the liquid interconnecting pipe (14) and the refrigerant in the gas interconnecting pipe (15) during the pressure control operation of the controller (90) can be kept at values greater than or equal to the lower-limit pressure difference. Thus, this aspect allows the mass flow rate of refrigerant passing through the utilization-side expansion valve (63) of (each of) the utilization-side unit(s) (12) in the cooling state to be high enough even during the pressure control operation of the controller (90), and allows the utilization-side unit(s) (12) in the cooling state to have adequate cooling capability.

[0027] As described above, if the difference in pressure between the refrigerant in the liquid interconnecting pipe (14) and the refrigerant in the gas interconnecting pipe (15) is too small, the mass flow rate of the refrigerant passing through the utilization-side expansion valve (63) of (each of) the utilization-side unit(s) (12) in the cooling state may become too low.

[0028] To address this problem, according to the fifth aspect, if the difference in pressure between the refrigerant in the liquid interconnecting pipe (14) and the refrigerant in the gas interconnecting pipe (15) is smaller than a predetermined reference pressure difference during the pressure control operation, the controller (90) controls the degree of opening of the subcooling expansion valve (35) to reduce the temperature of the liquid refrigerant delivered from the subcooling heat exchanger (34) to the liquid interconnecting pipe (14). This can increase the density of the liquid refrigerant delivered from the subcooling heat exchanger (34) to the liquid interconnecting pipe (14), thus reducing the degree to which the mass flow rate of the refrigerant passing through each utilization-side expansion valve (63) decreases.

BRIEF DESCRIPTION OF THE DRAWINGS

[0029]

[FIG. 1] FIG. 1 is a refrigerant circuit diagram showing a schematic configuration for a refrigeration apparatus according to a first embodiment.

[FIG. 2] FIG. 2 is a refrigerant circuit diagram showing a refrigeration apparatus in a normal mode.

[FIG. 3] FIG. 3 is a refrigerant circuit diagram showing a refrigeration apparatus in a defrosting mode.

[FIG. 4] FIG. 4 is a block diagram showing a configuration for a main controller.

[FIG. 5] FIG. 5 is a flowchart showing an operation performed by a heat-source-side expansion valve control section of a main controller.

[FIG. 6] FIG. 6 is a flowchart showing an operation performed by a subcooling expansion valve control section of a main controller.

DETAILED DESCRIPTION

[0030] Embodiments of the present invention will be described in detail with reference to the drawings. Note that the following embodiments and variations are merely beneficial examples in nature, and are not intended to limit the scope, applications, or use of the invention.

<<First Embodiment>>

[0031] A first embodiment will be described. A refrigeration apparatus (10) according to this embodiment is used to cool a space in a refrigerator.

[0032] As shown in FIG. 1, the refrigeration apparatus (10) includes a single heat-source-side unit (11) and a plurality of (in this embodiment, two) utilization-side units (12). The heat-source-side unit (11) is a so-called outdoor unit, and is installed outdoors. The utilization-side unit (12) is a so-called unit cooler, and is installed in the refrigerator. Note that the number of the utilization-side units (12) is merely an example.

[0033] The heat-source-side unit (11) is provided with a heat-source-side circuit (21), a heat-source-side fan (22), and a main controller (90). Meanwhile, each utilization-side unit (12) is provided with a utilization-side circuit (23), a utilization-side fan (24), a drain pan (25), and a utilization-side controller (99).

[0034] In the refrigeration apparatus (10), the heat-source-side circuit (21) of the heat-source-side unit (11) and the utilization-side circuits (23) of the utilization-side units (12) are connected together through a liquid interconnecting pipe (14) and a gas interconnecting pipe (15) to form a refrigerant circuit (20). The refrigerant circuit (20) circulates a refrigerant therethrough to perform a vapor compression refrigeration cycle.

[0035] The heat-source-side circuit (21) has liquid and gas ends respectively provided with a liquid stop valve (VI) and a gas stop valve (V2). The liquid interconnecting pipe (14) provides connection between the liquid stop valve (VI) of the heat-source-side circuit (21) and the liquid ends of the utilization-side circuits (23). The gas interconnecting pipe (15) provides connection between the gas stop valve (V2) of the heat-source-side circuit (21) and the gas ends of the utilization-side circuits (23). In the refrigerant circuit (20), the utilization-side circuits (23) of the utilization-side units (12) are connected together in parallel.

-Heat-Source-Side Circuit-

[0036] The heat-source-side circuit (21) includes first through third compressors (31a, 31b, 31c), a four-way valve (32), a heat-source-side heat exchanger (33), a subcooling heat exchanger (34), a subcooling expansion valve (35), first through third intermediate expansion valves (36a, 36b, 36c), a receiver (37), a heat-source-side expansion valve (38), first through third check valves (CV1-CV3), and an oil separator (41). The heat-source-side circuit (21) is provided with a discharge refrigerant pipe (51), a suction refrigerant pipe (52), a heat-source-side liquid refrigerant pipe (53), an injection pipe (54), a first connection pipe (55), a second connection pipe (56), and an oil return pipe (57). Note that the number of the compressors (31a-31c) of the heat-source-side unit (11) is merely an example.

<Compressors>

[0037] The first through third compressors (31a, 31b, 31c) are all hermetic scroll compressors. Each compressor (31a-31c) has a suction port, an intermediate port, and a discharge port. The compressor (31a-31c) compresses a refrigerant sucked thereinto through the suction port, and discharges the compressed refrigerant through the discharge port. The intermediate port of the compressor (31a-31c) is used to introduce a refrigerant into a compression chamber in the course of compression.

[0038] The first compressor (31a) has a variable capacity. An electric motor of the first compressor (31a) is supplied with power from an inverter outside the drawing. Changing the output frequency of the inverter triggers a change in the rotational speed of the first compressor (31a). This causes the operational capacity of the first compressor (31a) to vary. On the other hand, the second and third compressors (31b) and (31c) each have a fixed capacity. The second and third compressors (31b) and (31c) rotate at a constant rotational speed.

<Four-Way Valve>

[0039] The four-way valve (32) is switchable between a first state (indicated by the solid curves shown in FIG. 1) and a second state (indicated by the dashed curves shown in FIG. 1). In the first state, a first port communicates with a third port, and a second port communicates with a fourth port. In the second state, the first port communicates with the fourth port, and the second port communicates with the third port.

[0040] The first port of the four-way valve (32) is connected to the discharge ports of the compressors (31a-31c) through the discharge refrigerant pipe (51), and the second port thereof is connected to the suction ports of the compressors (31a-31c) through the suction refrigerant pipe (52). The third port of the four-way valve (32) is connected to the gas end of the heat-source-side heat exchanger (33), and the fourth port thereof is connected to the gas stop valve (V2).

<Discharge Refrigerant Pipe, Suction Refrigerant Pipe>

[0041] The discharge refrigerant pipe (51) includes (in this embodiment, three) discharge pipes (51a, 51b, 51c) equal in number to the compressors (31a-31c), and a single discharge collection pipe (51d). One end of the first discharge pipe (51a) is connected to the discharge port of the first compressor (31a), one end of the second discharge pipe (51b) is connected to the discharge port of the second compressor (31b), and one end of the third discharge pipe (51c) is connected to the discharge port of the third compressor (31c). The other end of each discharge pipe (51a, 51b, 51c) is connected to one end of the discharge collection pipe (51d). The other end of the discharge collection pipe (51d) is connected to the first port of the four-way valve (32).

[0042] The suction refrigerant pipe (52) includes (in this embodiment, three) suction pipes (52a, 52b, 52c) equal in number to the compressors (31a-31c), and a single main suction pipe (52d). One end of the first suction pipe (52a) is connected to the suction port of the first compressor (31a), one end of the second suction pipe (52b) is connected to the suction port of the second compressor (31b), and one end of the third suction pipe (52c) is connected to the suction port of the third compressor (31c). The other end of each suction pipe (52a, 52b, 52c) is connected to one end of the main suction pipe (52d). The other end of the main suction pipe (52d) is connected to the second port of the four-way valve (32).

<Heat-Source-Side Heat Exchanger>

[0043] The heat-source-side heat exchanger (33) is a cross-fin, fin-and-tube heat exchanger, and exchanges heat between a refrigerant and outdoor air. The heat-source-side heat exchanger (33) has its liquid end connected to the heat-source-side liquid refrigerant pipe (53), and has its gas end connected to the third port of the four-way valve (32). The heat-source-side fan (22) for supplying outdoor air to the heat-source-side heat exchanger (33) is disposed near the heat-source-side heat exchanger (33).

<Subcooling Heat Exchanger>

[0044] The subcooling heat exchanger (34) is a so-called plate-type heat exchanger. The subcooling heat exchanger (34) has a plurality of first channels (34a) and a plurality of second channels (34b). The subcooling heat exchanger (34) exchanges heat between a refrigerant flowing through the first channels (34a) and a refrigerant flowing through the second channels (34b).

<Heat-Source-Side Liquid Refrigerant Pipe>

[0045] The heat-source-side liquid refrigerant pipe (53) has two ends respectively connected to the heat-source-side heat exchanger (33) and the liquid stop valve (VI). The heat-source-side liquid refrigerant pipe (53) includes three heat-source-side liquid pipes (53a, 53b, 53c). The first heat-source-side liquid pipe (53a) provides connection between the liquid end of the heat-source-side heat exchanger (33) and the inlet of the receiver (37). The second heat-source-side liquid pipe (53b) provides connection between the outlet of the receiver (37) and the inlets of the first channels (34a) of the subcooling heat exchanger (34). The third heat-source-side liquid pipe (53c) provides connection between the outlets of the first channels (34a) of the subcooling heat exchanger (34) and the liquid stop valve (VI).

[0046] The first heat-source-side liquid pipe (53a) is provided with the first check valve (CV1). The first check valve (CV1) allows a refrigerant to flow from the heat-source-side heat exchanger (33) toward the receiver (37), but disallows a refrigerant to flow in the reverse direction.

[0047] The third heat-source-side liquid pipe (53c) is provided with the heat-source-side expansion valve (38) and the second check valve (CV2) in this order from the subcooling heat exchanger (34) toward the liquid stop valve (VI). The heat-source-side expansion valve (38) is an electric expansion valve having a variable degree of opening. The second check valve (CV2) allows a refrigerant to flow from the subcooling heat exchanger (34) toward the liquid stop valve (VI), but disallows a refrigerant to flow in the reverse direction.

<Injection Pipe>

[0048] The injection pipe (54) includes two main injection pipes (54m, 54n), and three injection branch pipes (54a, 54b, 54c).

[0049] The first main injection pipe (54m) has two ends respectively connected to a portion of the third heat-source-side liquid pipe (53c) between the subcooling heat exchanger (34) and the heat-source-side expansion valve (38), and the inlets of the second channels (34b) of the subcooling heat exchanger (34). The first main injection pipe (54m) constitutes a subcooling pipe. The first main injection pipe (54m) is provided with the subcooling expansion valve (35). One end of the second main injection pipe (54n) is connected to the outlets of the second channels (34b) of the subcooling heat exchanger (34). The other end of the second main injection pipe (54n) is connected to one end of each injection branch pipe (54a, 54b, 54c).

[0050] The other ends of the first, second, and third injection branch pipes (54a), (54b), and (54c) are respectively connected to the intermediate ports of the first, second, and third compressors (31a), (31b), and (31c). The injection branch pipes (54a-54c) are respectively provided with the intermediate expansion valves (36a, 36b, 36c). Each intermediate expansion valve (36a-36c) is an electric expansion valve having a variable degree of opening.

<Connection Pipes>

[0051] One end of the first connection pipe (55) is connected to a portion of the third heat-source-side liquid pipe (53c) between the second check valve (CV2) and the liquid stop valve (VI), and the other end thereof is connected to a portion of the first heat-source-side liquid pipe (53a) between the first check valve (CV1) and the receiver (37). The first connection pipe (55) is provided with the third check valve (CV3). The third check valve (CV3) allows a refrigerant to flow from the one end toward the other end of the first connection pipe (55), but disallows a refrigerant to flow in the reverse direction.

[0052] One end of the second connection pipe (56) is connected to a portion of the third heat-source-side liquid pipe (53c) between the heat-source-side expansion valve (38) and the second check valve (CV2), and the other end thereof is connected to a portion of the first heat-source-side liquid pipe (53a) between the heat-source-side heat exchanger (33) and the first check valve (CV1). The second connection pipe (56) is provided with the fourth check valve (CV4). The fourth check valve (CV4) allows a refrigerant to flow from the one end toward the other end of the second connection pipe (56), but disallows a refrigerant to flow in the reverse direction.

<Oil Separator, Oil Return Pipe>

[0053] The oil separator (41) is provided on a discharge collection pipe (51d) of the discharge refrigerant pipe (51). A gaseous refrigerant containing refrigerating machine oil in the form of mist is discharged from the compressors (31a-31c). The oil separator (41) separates the refrigerating machine oil from the refrigerant discharged from the compressors (31a-31c).

[0054] The oil return pipe (57) is used to return the refrigerating machine oil from the oil separator (41) to the compressors (31a-31c). The oil return pipe (57) has two ends respectively connected to the oil separator (41) and the second main injection pipe (54n). The oil return pipe (57) is provided with a capillary tube (42).

<Temperature Sensor, Pressure Sensor>

[0055] The heat-source-side circuit (21) is provided with a plurality of temperature sensors (81a, 81b, 81c, 82) and a plurality of pressure sensors (85, 86, 87).

[0056] The discharge pipes (51a, 51b, 51c) of the discharge refrigerant pipe (51) are respectively provided with discharged refrigerant temperature sensors (81a, 81b, 81c). The first discharged refrigerant temperature sensor (81a) is attached to the first discharge pipe (51a) to measure the temperature of the refrigerant discharged from the first compressor (31a). The second discharged refrigerant temperature sensor (81b) is attached to the second discharge pipe (51b) to measure the temperature of the refrigerant discharged from the second compressor (31b). The third discharged refrigerant temperature sensor (81c) is attached to the third discharge pipe (51c) to measure the temperature of the refrigerant discharged from the third compressor (31c).

[0057] The heat-source-side liquid refrigerant pipe (53) is provided with the liquid refrigerant temperature sensor (82). The liquid refrigerant temperature sensor (82) is attached to the third heat-source-side liquid pipe (53c) to measure the temperature of the refrigerant flowing through the third heat-source-side liquid pipe (53c).

[0058] The discharge pressure sensor (85) is connected to the discharge collection pipe (51d) of the discharge refrigerant pipe (51) to measure the pressure of the refrigerant discharged from the compressors (31a-31c). The suction pressure sensor (86) is connected to the main suction pipe (52d) of the suction refrigerant pipe (52) to measure the pressure of the refrigerant yet to be sucked into the compressors (31a-31c). The liquid refrigerant pressure sensor (87) is connected to the third heat-source-side liquid pipe (53c) of the heat-source-side liquid refrigerant pipe (53) to measure the pressure of the refrigerant flowing through the third heat-source-side liquid pipe (53c).

-Utilization-Side Circuit-

[0059] Each utilization-side circuit (23) includes a utilization-side heat exchanger (61), a drain pan heater (71b), a utilization-side solenoid valve (62), and a utilization-side expansion valve (63). The utilization-side circuit (23) is provided with a utilization-side liquid refrigerant pipe (71) and a utilization-side gaseous refrigerant pipe (72).

<Utilization-Side Heat Exchanger>

[0060] The utilization-side heat exchanger (61) is a cross-fin, fin-and-tube heat exchanger, and exchanges heat between a refrigerant and inside air. The utilization-side fan (24) for supplying inside air to the utilization-side heat exchanger (61) is disposed near the utilization-side heat exchanger (61).

<Drain Pan Heater>

[0061] The drain pan heater (71b) is configured as a pipe provided for the drain pan (25) disposed below the utilization-side heat exchanger (61). The drain pan heater (71b) is used to heat the drain pan (25) to prevent drain water from being frozen.

<Utilization-Side Liquid Refrigerant Pipe, Utilization-Side Gaseous Refrigerant Pipe>

[0062] The utilization-side liquid refrigerant pipe (71) includes a first utilization-side liquid pipe (71a) and a second utilization-side liquid pipe (71c). One end of the first utilization-side liquid pipe (71a) is connected to the liquid interconnecting pipe (14), and the other end thereof is connected to one end of the drain pan heater (71b). The one end of the first utilization-side liquid pipe (71a) constitutes the liquid end of the utilization-side circuit (23). The second utilization-side liquid pipe (71c) has two ends respectively connected to the other end of the drain pan heater (71b) and the liquid end of the utilization-side heat exchanger (61).

[0063] One end of the utilization-side gaseous refrigerant pipe (72) is connected to the gas end of the utilization-side heat exchanger (61), and the other end thereof is connected to the gas interconnecting pipe (15). The other end of the utilization-side gaseous refrigerant pipe (72) constitutes the gas end of the utilization-side circuit (23).

<Utilization-Side Solenoid Valve, Utilization-Side Expansion Valve>

[0064] The utilization-side solenoid valve (62) and the utilization-side expansion valve (63) are provided on the second utilization-side liquid pipe (71c) of the utilization-side liquid refrigerant pipe (71). In the second utilization-side liquid pipe (71c), the utilization-side expansion valve (63) is disposed between the utilization-side solenoid valve (62) and the utilization-side heat exchanger (61).

[0065] The utilization-side solenoid valve (62) interrupts the passage of electric current through a solenoid to switch between an open state and a closed state. While the utilization-side solenoid valve (62) is in the open state, the utilization-side unit (12) is in a cooling state where the utilization-side heat exchanger (61) functions as an evaporator to cool inside air. While the utilization-side solenoid valve (62) is in the closed state, the utilization-side unit (12) is in a suspended state where the flow of a refrigerant through the utilization-side heat exchanger (61) is interrupted.

[0066] The utilization-side expansion valve (63) is an externally equalized thermostatic expansion valve. A feeler bulb (63a) of the utilization-side expansion valve (63) is attached near one end of the utilization-side gaseous refrigerant pipe (72) (near the utilization-side heat exchanger (61)). An equalizer (63b) of the utilization-side expansion valve (63) is connected to a portion of the utilization-side gaseous refrigerant pipe (72) near one end thereof.

-Main Controller-

[0067] As shown in FIG. 2, the main controller (90) of the heat-source-side unit (11) includes a compressor control section (91), an intermediate expansion valve control section (92), a subcooling expansion valve control section (93), and a heat-source-side expansion valve control section (94). The main controller (90) receives values input from the temperature sensors (81a, 81b, 81c, 82) and the pressure sensors (85, 86, 87) provided for the heat-source-side unit (11). The main controller (90) receives a thermo-off signal from the utilization-side controller (99) of each utilization-side unit (12). A control operation performed by the main controller (90) will be described later.

-Utilization-Side Controller-

[0068] Although not shown, each utilization-side unit (12) is provided with a sucked air temperature sensor. The sucked air temperature sensor measures the temperature of inside air that has not passed through the utilization-side heat exchanger (61) yet. The utilization-side controller (99) receives a value measured by the sucked air temperature sensor. The utilization-side controller (99) opens and closes the utilization-side solenoid valve (62) based on the value measured by the sucked air temperature sensor. The utilization-side controller (99) outputs a thermo-off signal if the utilization-side solenoid valve (62) is to be closed. An operation performed by the utilization-side controller (99) will be described later.

-Operation of Refrigeration Apparatus-

[0069] The refrigeration apparatus (10) operates in a selected one of a normal mode for cooling an internal space and a defrosting mode for melting frost formed on the utilization-side heat exchanger (61).

<Operation in Normal Mode>

[0070] The operation of the refrigeration apparatus (10) in the normal mode will be described with reference to FIG. 2. A refrigeration cycle is performed by circulating a refrigerant through the refrigerant circuit (20) operating in the normal mode, in which the heat-source-side heat exchanger (33) functions as a condenser, and the utilization-side heat exchangers (61) function as evaporators.

[0071] The operation performed in the normal mode while both of the utilization-side units (12) are in the cooling state and all of the compressors (31a-31c) are operating will now be exemplified.

[0072] As shown in FIG. 2, the four-way valve (32) is set to be in the first state during operation in the normal mode. The main controller (90) controls the subcooling expansion valve (35), the intermediate expansion valves (36a, 36b, 36c), and the heat-source-side expansion valve (38). An operation of the main controller (90) will be described later. In the case shown in FIG. 2, the utilization-side solenoid valves (62) of the utilization-side units (12) are set to be in the open state.

[0073] The refrigerant discharged from the compressors (31a-31c) passes through the oil separator (41) in the discharge refrigerant pipe (51), then flows through the four-way valve (32) into the heat-source-side heat exchanger (33), dissipates heat to outdoor air in the heat-source-side heat exchanger (33), and condenses. The refrigerant (high-pressure refrigerant) that has flowed out of the heat-source-side heat exchanger (33) passes through the first heat-source-side liquid pipe (53a), the receiver (37), and the second heat-source-side liquid pipe (53b) in this order, flows into the first channels (34a) of the subcooling heat exchanger (34), and is cooled by the refrigerant flowing through the second channels (34b) of the subcooling heat exchanger (34). Part of the subcooled liquid refrigerant that has flowed out of the first channels (34a) of the subcooling heat exchanger (34) into the third heat-source-side liquid pipe (53c) flows into the first main injection pipe (54m). The remaining part passes through the heat-source-side expansion valve (38) and the liquid stop valve (V1) in this order, and then flows into the liquid interconnecting pipe (14).

[0074] The refrigerant that has flowed into the liquid interconnecting pipe (14) is split into the utilization-side circuits (23) of the utilization-side units (12). In each utilization-side circuit (23), the refrigerant that has flowed into the first utilization-side liquid pipe (71a) passes through the drain pan heater (71b), and then flows through the second utilization-side liquid pipe (71c) into the utilization-side solenoid valve (62). The refrigerant that has passed through the utilization-side solenoid valve (62) expands when passing through the utilization-side expansion valve (63), and turns into a gas-liquid two-phase refrigerant, which then flows into the utilization-side heat exchanger (61). The refrigerant that has flowed into the utilization-side heat exchanger (61) absorbs heat from inside air to evaporate. As a result, the inside air is cooled. The utilization-side unit (12) sends the inside air cooled in the utilization-side heat exchanger (61) back to the internal space.

[0075] The refrigerant that has evaporated in the utilization-side heat exchanger (61) flows through the utilization-side gaseous refrigerant pipe (72) into the gas interconnecting pipe (15). Flows of the refrigerant from the utilization-side circuits (23) enter, and merge together in the gas interconnecting pipe (15). Then, the merged refrigerant flows into the heat-source-side circuit (21), passes through the gas stop valve (V2) and the four-way valve (32) in this order, and is then sucked through the suction refrigerant pipe (52) into the compressors (31a-31c).

[0076] Meanwhile, the refrigerant that has flowed into the first main injection pipe (54m) expands when passing through the subcooling expansion valve (35), and turns into a gas-liquid two-phase refrigerant, which then flows into the second channels (34b) of the subcooling heat exchanger (34), and absorbs heat from the refrigerant (high-pressure refrigerant) flowing through the first channels (34a) of the subcooling heat exchanger (34) to evaporate. The refrigerant that has flowed through the second channels (34b) of the subcooling heat exchanger (34) into the second main injection pipe (54n) is introduced into the intermediate ports of the compressors (31a-31c).

<Operation in Defrosting Mode>

[0077] The operation of the refrigeration apparatus (10) in the defrosting mode will be described with reference to FIG. 3. An operation is performed in the defrosting mode if a predetermined condition (e.g., a condition where a period of time during which an operation in the normal mode is continued has reached a predetermined period of time) is satisfied. A refrigeration cycle is performed by circulating a refrigerant through the refrigerant circuit (20) operating in the defrosting mode, in which the utilization-side heat exchangers (61) function as condensers, and the heat-source-side heat exchanger (33) functions as an evaporator.

[0078] As shown in FIG. 3, the four-way valve (32) is set to be in the second state during the operation in the defrosting mode. The main controller (90) controls the subcooling expansion valve (35), the intermediate expansion valves (36a, 36b, 36c), and the heat-source-side expansion valve (38). In each utilization-side unit (12), the utilization-side solenoid valve (62) is set to be in the open state, and the utilization-side fan (24) is at rest.

[0079] The refrigerant discharged from the compressors (31a-31c) passes through the four-way valve (32), then flows into the gas interconnecting pipe (15), and is split into the utilization-side circuits (23) of the utilization-side units (12). The refrigerant split into the utilization-side circuits (23) flows into the utilization-side heat exchangers (61), and dissipates heat to condense. In each utilization-side heat exchanger (61), frost formed on the utilization-side heat exchanger (61) is heated by the refrigerant, and melts.

[0080] Flows of the refrigerants that have passed through the utilization-side heat exchangers (61) of the utilization-side circuits (23) enter, and merge together in the liquid interconnecting pipe (14), and then the merged refrigerant flows into the heat-source-side circuit (21). The refrigerant that has flowed into the heat-source-side circuit (21) passes through the liquid stop valve (VI), the first connection pipe (55), and the receiver (37) in this order, and then flows into the first channels (34a) of the subcooling heat exchanger (34). Part of the refrigerant that has flowed out of the first channels (34a) of the subcooling heat exchanger (34) flows into the first main injection pipe (54m). The remaining part flows into the heat-source-side expansion valve (38).

[0081] The refrigerant that has flowed into the heat-source-side expansion valve (38) expands when passing through the heat-source-side expansion valve (38), and turns into a gas-liquid two-phase refrigerant, which then flows into the heat-source-side heat exchanger (33), and absorbs heat from outdoor air to evaporate. The refrigerant that has evaporated in the heat-source-side heat exchanger (33) passes through the four-way valve (32), flows into the suction refrigerant pipe (52), and is then sucked into the compressors (31a-31c).

[0082] Meanwhile, the refrigerant that has flowed into the first main injection pipe (54m) passes through the second channels (34b) of the subcooling heat exchanger (34), flows into the second main injection pipe (54n), and is then introduced into the intermediate ports of the compressors (31a-31c).

-Operation of Utilization-Side Controller-

[0083] As described above, in each utilization-side unit (12), the utilization-side controller (99) opens and closes the utilization-side solenoid valve (62) based on the value measured by the sucked air temperature sensor. The operation of this utilization-side controller (99) will be described.

[0084] The utilization-side controller (99) controls the utilization-side solenoid valve (62) such that a value Tr measured by the sucked air temperature sensor falls within the range from the set internal temperature Tr_set - 1°C to the set internal temperature Tr_set + 1°C (i.e., $Tr\_set - 1 \leq Tr \leq Tr\_set + 1$).

[0085] Suppose that the utilization-side solenoid valve (62) is in the open state. While the utilization-side solenoid valve (62) is open, the utilization-side unit (12) is in the cooling state. Specifically, a refrigerant flows into the utilization-side heat exchanger (61) to evaporate. As a result, inside air is cooled in the utilization-side heat exchanger (61). While the utilization-side solenoid valve (62) is open, the temperature of the inside air (i.e., the value Tr measured by the sucked air temperature sensor) gradually decreases. If the value Tr measured by the sucked air temperature sensor is below Tr_set - 1 (i.e., $Tr < Tr\_set - 1$), the utilization-side controller (99) switches the utilization-side solenoid valve (62) from the open state to the closed state. The utilization-side controller (99) that has just switched the utilization-side solenoid valve (62) from the open state to the closed state outputs, to the main controller (90), a thermo-off signal indicating that the utilization-side unit (12) has been suspended.

[0086] While the utilization-side solenoid valve (62) is closed, the utilization-side unit (12) is in the suspended state. Specifically, the flow of a refrigerant through the utilization-side heat exchanger (61) is interrupted, and inside air is not cooled in the utilization-side heat exchanger (61). While the utilization-side solenoid valve (62) is closed, the temperature of inside air (i.e., the value Tr measured by the sucked air temperature sensor) gradually increases. If the value Tr measured by the sucked air temperature sensor is above Tr_set + 1 (i.e., $Tr\_set + 1 < Tr$), the utilization-side controller (99) switches the utilization-side solenoid valve (62) from the closed state to the open state.

-Operation of Main Controller-

[0087] As described above, the main controller (90) includes the compressor control section (91), the intermediate expansion valve control section (92), the subcooling expansion valve control section (93), and the heat-source-side expansion valve control section (94). Operations performed by the compressor control section (91), the intermediate expansion valve control section (92), the subcooling expansion valve control section (93), and the heat-source-side expansion valve control section (94) will now be described. The main controller (90) operates the four-way valve (32) to switch between the normal mode and the defrosting mode, and controls the rotational speed of the heat-source-side fan (22).

<Operation of Compressor Control Section>

[0088] The compressor control section (91) adjusts the operational capacity of the first compressor (31a), and switches the second and third compressors (31b) and (31c) between an on state and an off state, such that a value measured by the suction pressure sensor (86) is equal to a predetermined target pressure.

[0089] If the cooling capability of each utilization-side unit (12) is excessively low with respect to a load required to cool inside air, the evaporating pressure of a refrigerant in the utilization-side heat exchanger (61) (i.e., the low pressure of the refrigeration cycle) increases. The low pressure of the refrigeration cycle is substantially equal to the value measured by the suction pressure sensor (86). Thus, if the value measured by the suction pressure sensor (86) is above the target pressure, the compressor control section (91) operates to increase the operational capacities of the compressors (31a-31c). Specifically, in this case, the compressor control section (91) operates to gradually increase the output frequency of the inverter to increase the operational capacity of the first compressor (31a), and to start a suspended one of the second and third compressors (31b) and (31c).

[0090] On the other hand, if the cooling capability of each utilization-side unit (12) is excessively high with respect to the load required to cool inside air, the evaporating pressure of a refrigerant in the utilization-side heat exchanger (61) (i.e., the low pressure of the refrigeration cycle) decreases. Thus, if the value measured by the suction pressure sensor (86) is below the target pressure, the compressor control section (91) operates to reduce the operational capacities of the compressors (31a-31c). Specifically, in this case, the compressor control section (91) operates to gradually reduce the output frequency of the inverter to reduce the operational capacity of the first compressor (31a), and to suspend an operating one of the second and third compressors (31b) and (31c).

<Operation of Intermediate Expansion Valve Control Section>

[0091] The intermediate expansion valve control section (92) adjusts the degrees of opening of the intermediate expansion valves (36a-36c). The intermediate expansion valve control section (92) adjusts the degree of opening of the first intermediate expansion valve (36a) based on values measured by the first discharged refrigerant temperature sensor (81a) and the discharge pressure sensor (85), adjusts the degree of opening of the second intermediate expansion valve (36b) based on values measured by the second discharged refrigerant temperature sensor (81b) and the discharge pressure sensor (85), and adjusts the degree of opening of the third intermediate expansion valve (36c) based on values measured by the third discharged refrigerant temperature sensor (81c) and the discharge pressure sensor (85).

[0092] An operation in which the intermediate expansion valve control section (92) adjusts the degree of opening of the first intermediate expansion valve (36a) will now be described. The intermediate expansion valve control section (92) further adjusts the degrees of opening of the second and third intermediate expansion valves (36b) and (36c) in the same way.

[0093] If the value measured by the first discharged refrigerant temperature sensor (81a) is above a predetermined upper-limit temperature, the intermediate expansion valve control section (92) operates to increase the degree of opening of the first intermediate expansion valve (36a) in order to reduce the value measured by the first discharged refrigerant temperature sensor (81a).

[0094] On the other hand, if the value measured by the first discharged refrigerant temperature sensor (81a) is below the predetermined upper-limit temperature, the intermediate expansion valve control section (92) adjusts the degree of opening of the first intermediate expansion valve (36a) such that the superheat of the refrigerant discharged from the first compressor (31a) is equal to a predetermined target discharge superheat. Specifically, the intermediate expansion valve control section (92) calculates the superheat of the refrigerant discharged from the first compressor (3 1a), based on the values measured by the first discharged refrigerant temperature sensor (81a) and the discharge pressure sensor (85). If the calculated superheat is above the target discharge superheat, the intermediate expansion valve control section (92) increases the degree of opening of the first intermediate expansion valve (36a). If the calculated superheat is below the target discharge superheat, the intermediate expansion valve control section (92) reduces the degree of opening of the first intermediate expansion valve (36a).

[0095] If one or more of the compressors (31a-31c) respectively associated with the intermediate expansion valves (36a-36c) are operating, the intermediate expansion valve control section (92) adjusts the degree of opening of the associated intermediate expansion valve(s) (36a-36c). If one or more of the compressors (31a-31c) respectively associated with the intermediate expansion valves (36a-36c) are at rest, the intermediate expansion valve control section (92) keeps the associated intermediate expansion valve(s) (36a-36c) fully closed. Specifically, the intermediate expansion valve control section (92) adjusts the degree of opening of the second intermediate expansion valve (36b) while the second compressor (31b) is operating. The intermediate expansion valve control section (92) keeps the second intermediate expansion valve (36b) fully closed while the second compressor (31b) is at rest. The intermediate expansion valve control section (92) adjusts the degree of opening of the third intermediate expansion valve (36c) while the third compressor (31c) is operating. The intermediate expansion valve control section (92) keeps the third intermediate expansion valve (36c) fully closed while the third compressor (31c) is at rest.

<Operation of Subcooling Expansion Valve Control Section>

[0096] The subcooling expansion valve control section (93) adjusts the degree of opening of the subcooling expansion valve (35) according to the temperature of a liquid refrigerant delivered from the heat-source-side unit (11) to the liquid interconnecting pipe (14) during the operation in the normal mode. The temperature of the liquid refrigerant delivered from the heat-source-side unit (11) to the liquid interconnecting pipe (14) during the operation in the normal mode is substantially equal to the value measured by the liquid refrigerant temperature sensor (82). Thus, the subcooling expansion valve control section (93) adjusts the degree of opening of the subcooling expansion valve (35) such that the value measured by the liquid refrigerant temperature sensor (82) is equal to a predetermined target liquid refrigerant temperature (e.g., 20°C). If an operation is performed in the normal mode, and the heat-source-side expansion valve (38) is in a fully-open state, the degree of subcooling of the liquid refrigerant delivered from the heat-source-side unit (11) to the liquid interconnecting pipe (14) is generally about 0°C to 20°C.

[0097] Specifically, if the value measured by the liquid refrigerant temperature sensor (82) is above the target liquid refrigerant temperature, the subcooling expansion valve control section (93) reduces the degree of opening of the subcooling expansion valve (35), and reduces the temperature of the refrigerant delivered from the subcooling expansion valve (35) to the second channels (34b) of the subcooling heat exchanger (34). On the other hand, if the value measured by the liquid refrigerant temperature sensor (82) is below the target liquid refrigerant temperature, the subcooling expansion valve control section (93) increases the degree of opening of the subcooling expansion valve (35), and increases the temperature of the refrigerant delivered from the subcooling expansion valve (35) to the second channels (34b) of the subcooling heat exchanger (34).

<Operation of Heat-Source-Side Expansion Valve Control Section>

[0098] The heat-source-side expansion valve control section (94) performs control to prevent a liquid hammer phenomenon. The control is performed by controlling the degree of opening of the heat-source-side expansion valve (38) in order to prevent the liquid hammer phenomenon caused by the utilization-side solenoid valves (62) of the utilization-side units (12) that has just switched from the closed state to the open state. How the heat-source-side expansion valve control section (94) performs control to prevent the liquid hammer phenomenon will now be described with reference to the flowchart shown in FIG. 5.

[0099] In step ST1, the heat-source-side expansion valve control section (94) determines whether or not the refrigeration apparatus (10) is operating in the normal mode. If the refrigeration apparatus (10) is operating in the normal mode, the process proceeds to step ST2, and the heat-source-side expansion valve control section (94) continues performing control to prevent the liquid hammer phenomenon. On the other hand, if the refrigeration apparatus (10) is not operating in the normal mode (i.e., if the refrigeration apparatus (10) is operating in the defrosting mode, or if all of the compressors (31a-31c) are in a standby mode in which they are at rest), the control to prevent the liquid hammer phenomenon is terminated.

[0100] In step ST2, the heat-source-side expansion valve control section (94) loads a value HP measured by the discharge pressure sensor (85) and a value LP measured by the suction pressure sensor (86).

[0101] In subsequent step ST3, the heat-source-side expansion valve control section (94) determines whether or not at least one of the utilization-side units (12) is suspended. As described above, if the utilization-side solenoid valve (62) switches from the open state to the closed state to allow an associated one of the utilization-side units (12) to be suspended, the utilization-side controller (99) provided for the associated utilization-side unit (12) outputs, to the main controller (90), a thermo-off signal indicating that the associated utilization-side unit (12) has been suspended. Thus, the heat-source-side expansion valve control section (94) determines whether or not it has received a thermo-off signal or signals from one or some of the utilization-side units (12).

[0102] If none of the utilization-side units (12) outputs a thermo-off signal during the operation in the normal mode, a determination is made that all of the utilization-side units (12) are in the cooling state. In this case, the utilization-side solenoid valves (62) of all of the utilization-side units (12) are open. This prevents the liquid hammer phenomenon from being caused by the utilization-side solenoid valves (62) that has just switched from the closed state to the open state.

[0103] For this reason, if a thermo-off signal is not received from any of the utilization-side units (12), the process proceeds to step ST5, and the heat-source-side expansion valve control section (94) performs a degree-of-opening maintaining operation. For example, if the heat-source-side expansion valve (38) is not fully opened, the heat-source-side expansion valve control section (94) increases the degree of opening of the heat-source-side expansion valve (38) so that the heat-source-side expansion valve (38) is fully opened, and keeps the heat-source-side expansion valve (38) fully opened. If the heat-source-side expansion valve (38) has already been fully opened, the heat-source-side expansion valve control section (94) continues to keep the heat-source-side expansion valve (38) fully opened. The termination of step ST5 allows the heat-source-side expansion valve control section (94) to temporarily terminate the control to prevent the liquid hammer phenomenon.

[0104] On the other hand, if a thermo-off signal or signals are received from one or some of the utilization-side units (12), the utilization-side solenoid valve(s) (62) of the utilization-side unit(s) (12) that has outputted the thermo-off signal(s) is closed. Thus, if no countermeasure is taken, a liquid hammer phenomenon may occur when the utilization-side solenoid valve(s) (62) opens. However, if the difference (HP - LP) between the value HP measured by the discharge pressure sensor (85) and the value LP measured by the suction pressure sensor (86) is sufficiently small, no liquid hammer phenomenon occurs when the utilization-side solenoid valve(s) (62) opens.

[0105] Thus, if a thermo-off signal or signals are received from one or some of the utilization-side units (12), the process in the heat-source-side expansion valve control section (94) proceeds to step ST4. In step ST4, the heat-source-side expansion valve control section (94) determines whether or not the difference (HP - LP) between the value HP measured by the discharge pressure sensor (85) and the value LP measured by the suction pressure sensor (86) is larger than or equal to a predetermined upper-limit pressure difference ΔPmax (e.g., 2MPa).

[0106] If the difference (HP - LP) is smaller than the upper-limit pressure difference ΔPmax, the possibility that when the utilization-side solenoid valve (62) opens, the liquid hammer phenomenon may occur is very low. Thus, if the difference (HP - LP) is smaller than the upper-limit pressure difference ΔPmax, the process in the heat-source-side expansion valve control section (94) proceeds to step ST5, and the heat-source-side expansion valve control section (94) performs the degree-of-opening maintaining operation. Then, the control to prevent the liquid hammer phenomenon terminates temporarily. The degree-of-opening maintaining operation is performed as described above.

[0107] On the other hand, if the difference (HP - LP) is larger than or equal to the upper-limit pressure difference ΔPmax, the liquid hammer phenomenon is highly likely to occur when the utilization-side solenoid valve(s) (62) of a suspended one or ones of the utilization-side units (12) opens. Thus, in this case, the heat-source-side expansion valve control section (94) performs a pressure control operation in subsequent steps ST6 through ST8. This pressure control operation is performed to allow the degree of opening of the heat-source-side expansion valve (38) to be lower than that of the fully opened valve, and to allow the pressure of a refrigerant flowing through the liquid interconnecting pipe (14) to be lower than the condensing pressure of a refrigerant in the heat-source-side heat exchanger (33) (i.e., the high pressure of the refrigeration cycle). This pressure control operation is also performed to control the degree of opening of the heat-source-side expansion valve (38) to prevent the pressure of a refrigerant in the liquid interconnecting pipe (14) from exceeding a predetermined upper limit.

[0108] In step ST6, the heat-source-side expansion valve control section (94) loads a value Ps measured by the liquid refrigerant pressure sensor (87) and a value TL measured by the liquid refrigerant temperature sensor (82).

[0109] In subsequent step ST7, the heat-source-side expansion valve control section (94) determines a target pressure Ps_t that is a target pressure of a refrigerant flowing through the liquid interconnecting pipe (14). The target pressure Ps_t is the upper limit of the pressure of a refrigerant in the liquid interconnecting pipe (14) during a pressure control operation.

[0110] Specifically, in step ST7, the heat-source-side expansion valve control section (94) calculates a value Ps_1 (= LP + 1) obtained by adding a predetermined lower-limit pressure difference ΔPmin (e.g., 1 MPa) to the value LP measured by the suction pressure sensor (86), and a pressure value Ps_2 determined such that the degree of subcooling of a liquid refrigerant delivered from the heat-source-side unit (11) to the liquid interconnecting pipe (14) is equal to a predetermined target degree of subcooling SC_t (e.g., 3°C). The pressure value Ps_2 is a pressure determined such that the saturation temperature is equal to (TL + SC_t), and is calculated based on the properties of a refrigerant charged into the refrigerant circuit (20).

[0111] The heat-source-side expansion valve control section (94) selects a greater one of the calculated values Ps_1 and Ps_2 as the target pressure Ps_t. Specifically, the heat-source-side expansion valve control section (94) sets the target pressure Ps_t such that the difference (Ps - LP) between the value Ps measured by the liquid refrigerant pressure sensor (87) and the value LP measured by the suction pressure sensor (86) is larger than or equal to the lower-limit pressure difference ΔPmin, and the degree of subcooling of the liquid refrigerant delivered from the heat-source-side unit (11) to the liquid interconnecting pipe (14) is equal to the target degree of subcooling SC_t. Note that the difference (Ps - LP) is substantially equal to the difference in pressure between a refrigerant flowing through the liquid interconnecting pipe (14) and a refrigerant flowing through the gas interconnecting pipe (15).

[0112] In subsequent step ST8, the heat-source-side expansion valve control section (94) adjusts the degree of opening of the heat-source-side expansion valve (38) such that the value Ps measured by the liquid refrigerant pressure sensor (87) is equal to the target pressure Ps_t. Specifically, if the value Ps measured by the liquid refrigerant pressure sensor (87) is above the target pressure Ps_t, the heat-source-side expansion valve control section (94) reduces the degree of opening of the heat-source-side expansion valve (38). On the other hand, if the value Ps measured by the liquid refrigerant pressure sensor (87) is below the target pressure Ps_t, the heat-source-side expansion valve control section (94) increases the degree of opening of the heat-source-side expansion valve (38). In this manner, the heat-source-side expansion valve control section (94) performs the pressure control operation to substantially keep the pressure of a refrigerant in the liquid interconnecting pipe (14) at the target pressure Ps_t.

[0113] The target degree of subcooling SC_t is set to be lower than the "degree of subcooling of the liquid refrigerant delivered from the heat-source-side unit (11) to the liquid interconnecting pipe (14) immediately before the heat-source-side expansion valve control section (94) starts performing the pressure control operation." Specifically, the heat-source-side expansion valve control section (94) adjusts the degree of opening of the heat-source-side expansion valve (38) such that the degree of subcooling of the liquid refrigerant supplied to the liquid interconnecting pipe (14) by the heat-source-side unit (11) is equal to the predetermined target degree of subcooling SC_t, which is lower than the degree of subcooling of the liquid refrigerant supplied to the liquid interconnecting pipe (14) by the heat-source-side unit (11) during the degree-of-opening maintaining operation. Thus, the pressure control operation of the heat-source-side expansion valve control section (94) allows the degree of opening of the heat-source-side expansion valve (38) to be lower than that of the fully opened valve.

[0114] If the pressure control operation of the heat-source-side expansion valve control section (94) allows the degree of opening of the heat-source-side expansion valve (38) to be lower than that of the fully opened valve, the pressure of the refrigerant supplied through the liquid interconnecting pipe (14) to the utilization-side units (12) after passing through the heat-source-side expansion valve (38) is lower than in a situation where the heat-source-side expansion valve (38) is fully opened. Such a reduction in the pressure of the refrigerant supplied from the liquid interconnecting pipe (14) to the utilization-side units (12) reduces the difference in pressure between two ends of each of the closed utilization-side solenoid valves (62), and simultaneously reduces the density of the refrigerant supplied from the liquid interconnecting pipe (14) to the utilization-side units (12).

[0115] The larger the difference in pressure between the two ends of each of the closed utilization-side solenoid valves (62) is, the more likely the liquid hammer phenomenon is to be caused by the utilization-side solenoid valves (62) of the utilization-side units (12) that has just switched from the closed state to the open state. In addition, the higher the density of the refrigerant supplied from the liquid interconnecting pipe (14) to the utilization-side units (12) is, the more likely the liquid hammer phenomenon is to occur.

[0116] To address this problem, if one or more of the utilization-side units (12) is in the suspended state where the associated utilization-side solenoid valve(s) (62) is closed, and the difference (HP - LP) is larger than or equal to the upper-limit pressure difference ΔPmax, the heat-source-side expansion valve control section (94) performs the pressure control operation to allow the degree of opening of the heat-source-side expansion valve (38) to be lower than that of the fully opened valve. As a result, the difference in pressure between the two ends of the closed utilization-side solenoid valve (62) is smaller, and the density of the refrigerant supplied from the liquid interconnecting pipe (14) to the utilization-side units (12) is lower, than in a situation where the heat-source-side expansion valve (38) is kept fully open. This reduces the possibility of the liquid hammer phenomenon caused by the utilization-side solenoid valve (62) that has just switched from the closed state to the open state.

[0117] If any one of a condition where all of the utilization-side units (12) are in the cooling state during the pressure control operation, and a condition where the difference (HP - LP) is smaller than the upper-limit pressure difference ΔPmax during the pressure control operation is satisfied, the heat-source-side expansion valve control section (94) terminates the pressure control operation, and performs the degree-of-opening maintaining operation.

-Advantages of Embodiment-

[0118] If, in the refrigeration apparatus (10) of this embodiment, one or some of the utilization-side units (12) are suspended, and the difference (Ps - LP) in pressure between the refrigerant in the liquid interconnecting pipe (14) and the refrigerant in the gas interconnecting pipe (15) is larger than or equal to the predetermined upper-limit pressure difference ΔPmax, the heat-source-side expansion valve control section (94) of the main controller (90) performs the pressure control operation to allow the degree of opening of the heat-source-side expansion valve (38) to be lower than that during the degree-of-opening maintaining operation. This allows the pressure and density of a liquid refrigerant present near one end of the closed utilization-side solenoid valve (62) of a suspended one of the utilization-side units (12) to be lower than in a situation where the heat-source-side expansion valve (38) is kept fully open. Specifically, the difference in pressure between the two ends of the closed utilization-side solenoid valve (62) of the suspended utilization-side unit (12) is smaller, and the density of the liquid refrigerant present near the one end of the closed utilization-side solenoid valve (62) is lower, than in a situation where the heat-source-side expansion valve (38) is kept fully open. Thus, adjusting the degree of opening of the heat-source-side expansion valve (38) of the heat-source-side unit (11) can prevent the liquid hammer phenomenon caused by the utilization-side solenoid valve(s) (62) that has just switched from the closed state to the open state. Thus, this embodiment can prevent the liquid hammer phenomenon without adding new members to the refrigeration apparatus (10).

[0119] In the refrigeration apparatus (10) of this embodiment, the degree of opening of the heat-source-side expansion valve (38) is controlled to prevent the liquid hammer phenomenon. For this reason, unlike a known situation where the use of an electric heater prevents the liquid hammer phenomenon, preventing the liquid hammer phenomenon hardly leads to an increase in the consumed power. Thus, this embodiment can prevent the liquid hammer phenomenon while avoiding an increase in the running cost of the refrigeration apparatus (10).

[0120] In such a situation, while the heat-source-side expansion valve control section (94) of the main controller (90) is performing the pressure control operation, one or some of the utilization-side units (12) are in the suspended state. In this state, the other utilization-side unit(s) (12) is in the cooling state. If the difference in pressure between the refrigerant in the liquid interconnecting pipe (14) and the refrigerant in the gas interconnecting pipe (15) is too small in this state, the flow rate of a refrigerant passing through the utilization-side expansion valve(s) (63) of the utilization-side unit(s) (12) in the cooling state may be too low. This may prevent the utilization-side unit(s) (12) in the cooling state from having adequate cooling capability.

[0121] To address this problem, according to this embodiment, the difference (Ps - LP) in pressure between the refrigerant in the liquid interconnecting pipe (14) and the refrigerant in the gas interconnecting pipe (15) during the pressure control operation of the heat-source-side expansion valve control section (94) can be kept at values greater than or equal to the lower-limit pressure difference ΔPmin. Thus, this embodiment allows the flow rate of a refrigerant passing through the utilization-side expansion valve(s) (63) of the utilization-side unit(s) (12) in the cooling state to be high enough even during the pressure control operation of the heat-source-side expansion valve control section (94), and allows the utilization-side unit(s) (12) in the cooling state to have adequate cooling capability.

[0122] In such a situation, if the refrigerant introduced from the liquid interconnecting pipe (14) into the utilization-side units (12) turns into a gas-liquid two-phase refrigerant, the gas-liquid two-phase refrigerant passes through the utilization-side expansion valves (63). Consequently, the flow rate of a refrigerant passing through the utilization-side expansion valves (63) may become insufficient. This may prevent the utilization-side heat exchangers (61) from having adequate cooling capability.

[0123] In the refrigeration apparatus (10) of this embodiment, the heat-source-side expansion valve control section (94) that is performing the pressure control operation adjusts the degree of opening of the heat-source-side expansion valve (38) such that the degree of subcooling of the liquid refrigerant supplied to the liquid interconnecting pipe (14) by the heat-source-side unit (11) during the pressure control operation is equal to the target degree of subcooling SC_t. This allows a liquid single-phase refrigerant to be reliably supplied from the liquid interconnecting pipe (14) to the utilization-side unit(s) (12) in the cooling state during the pressure control operation of the controller (90). In addition, a liquid single-phase refrigerant can be reliably supplied from the liquid interconnecting pipe (14) also to the utilization-side units (12) in each of which the utilization-side solenoid valve (62) has just switched from the closed state to the open state. Thus, this embodiment allows the flow rate of the refrigerant passing through the utilization-side expansion valve(s) (63) of the utilization-side unit(s) (12) in the cooling state to be high enough, and allows the associated utilization-side heat exchanger(s) (61) to have adequate cooling capability.

-First Variation of First Embodiment-

[0124] As described above, the heat-source-side expansion valve control section (94) of the main controller (90) according to this embodiment performs, as a degree-of-opening maintaining operation, an operation to keep the heat-source-side expansion valve (38) fully open. Alternatively, the heat-source-side expansion valve control section (94) may perform, as the degree-of-opening maintaining operation, an operation to keep the degree of opening of the heat-source-side expansion valve (38) at a fixed degree of opening almost equal to that of the fully open valve.

-Second Variation of First Embodiment-

[0125] As described above, the heat-source-side expansion valve control section (94) of the main controller (90) according to this embodiment performs, as a pressure control operation, an operation to adjust the degree of opening of the heat-source-side expansion valve (38) such that the pressure of the refrigerant in the liquid interconnecting pipe (14) is equal to the target pressure Ps_t. Alternatively, the heat-source-side expansion valve control section (94) may perform, as the pressure control operation, an operation to adjust the degree of opening of the heat-source-side expansion valve (38) such that the pressure of the refrigerant in the liquid interconnecting pipe (14) is lower than or equal to the target pressure Ps_t. If the value Ps measured by the liquid refrigerant pressure sensor (87) is below the target pressure Ps_t, the heat-source-side expansion valve control section (94) of this variation maintains the degree of opening of the heat-source-side expansion valve (38) without increasing this degree of opening.

<<Second Embodiment>>

[0126] A second embodiment will be described. A main controller (90) of a refrigeration apparatus (10) according to this embodiment makes a heat-source-side expansion valve control section (94) perform a different operation from that performed in the first embodiment. The difference of the operation performed by the heat-source-side expansion valve control section (94) of this embodiment from that in the first embodiment will now be described.

[0127] The heat-source-side expansion valve control section (94) of this embodiment is configured to perform a pressure control operation under normal conditions. The pressure control operation under normal conditions also represents that the degree of opening of a heat-source-side expansion valve (38) is controlled such that the pressure of a refrigerant in a liquid interconnecting pipe (14) is lower than or equal to a predetermined upper-limit liquid pressure. If the heat-source-side expansion valve control section (94) does not receive a thermo-off signal from any of utilization-side units (12), it performs a pressure control operation under normal conditions instead of a degree-of-opening maintaining operation.

[0128] The heat-source-side expansion valve control section (94) performs, as the pressure control operation under normal conditions, an operation to control the degree of opening of the heat-source-side expansion valve (38) such that a value Ps measured by a liquid refrigerant pressure sensor (87) is less than or equal to a predetermined upper limit. Specifically, if a value HP measured by a discharge pressure sensor (85) is above the upper-limit liquid pressure, the heat-source-side expansion valve control section (94) reduces the degree of opening of the heat-source-side expansion valve (38) such that the value Ps measured by the liquid refrigerant pressure sensor (87) is less than or equal to the upper-limit liquid pressure. As a result, the value Ps measured by the liquid refrigerant pressure sensor (87) is kept at values less than or equal to the upper-limit liquid pressure.

[0129] In this embodiment, when a refrigeration apparatus is renewed, already-existing interconnecting pipes (14, 15) of the refrigeration apparatus yet to be renewed may be used to provide a new refrigeration apparatus (10). If the refrigeration apparatus yet to be renewed and the renewed refrigeration apparatus employ different refrigerants (e.g., if the renewed refrigeration apparatus employs a refrigerant R22, and the refrigeration apparatus (10) yet to be renewed employs a refrigerant R410A), the upper limit of the high pressure of a refrigeration cycle performed by the renewed refrigeration apparatus (10) may be higher than that of the high pressure of a refrigeration cycle performed by the refrigeration apparatus yet to be renewed. In such a case, the upper limit of the high pressure of the refrigeration cycle performed by the renewed new refrigeration apparatus (10) may be above the allowable pressure of the already-existing liquid interconnecting pipe (14). To address this problem, the refrigeration apparatus (10) of this embodiment performs a pressure control operation under normal conditions to allow the pressure of a refrigerant in the liquid interconnecting pipe (14) to be lower than or equal to the allowable pressure even in such a situation. Thus, the upper-limit liquid pressure during the pressure control operation under normal conditions is set to be lower than or equal to the allowable pressure of the liquid interconnecting pipe

(14).

[0130] In this embodiment, the pressure control operation performed by the heat-source-side expansion valve control section (94) is different from that in the first embodiment. The difference of the pressure control operation performed by the heat-source-side expansion valve control section (94) from that in the first embodiment will now be described.

[0131] The pressure control operation performed by the heat-source-side expansion valve control section (94) of this embodiment is different from that in the first embodiment in terms of how the operation is performed in step ST7 of FIG. 5. In step ST7 of FIG. 5, the heat-source-side expansion valve control section (94) of this embodiment compares a "greater one of a pressure value Ps_1 and a pressure value Ps_2" to a "previously stored upper-limit liquid pressure," and determines a less one of these values to be the target pressure Ps_t. Specifically, the heat-source-side expansion valve control section (94) of this embodiment sets the target pressure Ps_t such that the pressure of the refrigerant in the liquid interconnecting pipe (14) is always kept at pressures lower than or equal to the upper-limit liquid pressure even during the pressure control operation.

<<Other Embodiments>>

[0132] In the refrigeration apparatus (10) of each of the first and second embodiments, the subcooling expansion valve control section (93) of the main controller (90) may be configured to perform a flow-rate maintaining operation described below. The difference of an operation performed by a subcooling expansion valve control section (93) of this variation from that in the first embodiment will now be described.

[0133] The flow-rate maintaining operation is performed to allow the mass flow rate of a refrigerant supplied from a liquid interconnecting pipe (14) to one or some of utilization-side units (12) in a cooling state to be high enough even in a situation where when a heat-source-side expansion valve control section (94) performs a pressure control operation (i.e., when the utilization-side unit(s) (12) is in a suspended state), the difference in pressure between a refrigerant in the liquid interconnecting pipe (14) and a refrigerant in a gas interconnecting pipe (15) is smaller than a predetermined reference pressure difference (in this variation, 10 MPa). To maintain the flow rate, the subcooling expansion valve control section (93) of this embodiment performs, as the flow-rate maintaining operation, an operation to control the degree of opening of the subcooling expansion valve (35) to reduce the temperature of a liquid refrigerant delivered from the subcooling heat exchanger (34) to the liquid interconnecting pipe (14).

[0134] As described above, the subcooling expansion valve control section (93) controls the degree of opening of the subcooling expansion valve (35) such that a value TL measured by a liquid refrigerant temperature sensor (82) is equal to a target liquid refrigerant temperature. During the flow-rate maintaining operation, the subcooling expansion valve control section (93) sets a target liquid refrigerant temperature T_t as described below.

[0135] Specifically, the subcooling expansion valve control section (93) loads a value measured by a liquid refrigerant pressure sensor (87), a value measured by a suction pressure sensor (86), and the degree of opening of a utilization-side expansion valve (63) of (each of) the utilization-side unit(s) (12) in the cooling state every X seconds (e.g., five seconds). The subcooling expansion valve control section (93) determines, in step ST11, whether or not the value (Ps - LP) is less than the reference pressure difference, based on a value Ps currently measured by the liquid refrigerant pressure sensor (87) and a value LP currently measured by the suction pressure sensor (86).

[0136] If the value (Ps - LP) is greater than or equal to the reference pressure difference, the subcooling expansion valve control section (93) determines that the mass flow rate of a refrigerant flowing through the liquid interconnecting pipe (14) is high enough, and terminates the flow-rate maintaining operation. On the other hand, if the value (Ps - LP) is less than the reference pressure, the process in the subcooling expansion valve control section (93) proceeds to step ST12.

[0137] In step ST12, the subcooling expansion valve control section (93) calculates the mass flow rate of a refrigerant that flowed through the liquid interconnecting pipe (14) X seconds before the present moment, based on a value Ps' measured by the liquid refrigerant pressure sensor (87), a value TL' measured by the liquid refrigerant temperature sensor (82), a value LP' measured by the suction pressure sensor (86), and the degree of opening VO' of the utilization-side expansion valve (63) of (each of) the utilization-side unit(s) (12) in the cooling state, which were obtained X seconds before the present moment. That is to say, the subcooling expansion valve control section (93) calculates the mass flow rate of a refrigerant passing through the utilization-side expansion valve (63) of (each of) the utilization-side unit(s) (12) in the cooling state, and determines the sum of these mass flow rates to be the mass flow rate of the refrigerant that flowed through the liquid interconnecting pipe (14) X seconds before the present moment.

[0138] Specifically, the subcooling expansion valve control section (93) calculates the density p' of the refrigerant that flowed through the liquid interconnecting pipe (14) X seconds before the present moment, based on the measured value Ps', the measured value TL', and the properties of the refrigerant. The subcooling expansion valve control section (93) calculates the volume flow rates VL of refrigerants that flowed through the utilization-side expansion valves (63) X seconds before the present moment, based on the measured value Ps', the measured value LP', and the degree of opening VO'. The subcooling expansion valve control section (93) individually calculates the mass flow rates Gev' ($Gev' = p' \times VL'$) of refrigerants that flowed through the utilization-side expansion valves (63) X seconds before the present moment by multiplying the volume flow rates VL' by the density p', and determines the sum of the mass flow rates Gev' of the refrigerants through the utilization-side expansion valves (63) to be the mass flow rate G' (= ΣGev') of the refrigerant that flowed through the liquid interconnecting pipe (14) X seconds before the present moment. If only one of the utilization-side units (12) is suspended, the mass flow rate Gev' of the refrigerant through the utilization-side expansion valve (63) of the suspended utilization-side unit (12) is the mass flow rate G' (= Gev') of the refrigerant that flowed through the liquid interconnecting pipe (14) X seconds before the present moment.

[0139] In subsequent step ST13, the subcooling expansion valve control section (93) calculates an equivalent density p, based on the value Ps measured by the liquid refrigerant pressure sensor (87), the value LP measured by the suction pressure sensor (86), and the degree of opening VO of the utilization-side expansion valve (63) of (each of) the utilization-side unit(s) (12) in the cooling state, which are obtained at the present moment.

[0140] Specifically, the subcooling expansion valve control section (93) individually calculates the volume flow rates VLev of a refrigerant that is flowing through the utilization-side expansion valves (63) at the present moment, based on the measured value Ps, the measured value LP, and the degree of opening VO, and determines the sum of the volume flow rates VLev of the refrigerant through the utilization-side expansion valves (63) to be the volume flow rate VL (= ΣVLev) of a refrigerant that is flowing through the liquid interconnecting pipe (14) at the present moment. If only one of the utilization-side units (12) is suspended, the mass flow rate Vev of the refrigerant through the utilization-side expansion valve (63) of the suspended utilization-side unit (12) is the volume flow rate VL (= VLev) of the refrigerant that is flowing through the liquid interconnecting pipe (14) at the present moment. Then, the subcooling expansion valve control section (93) determines a value obtained by dividing the mass flow rate G' of the refrigerant that flowed through the liquid interconnecting pipe (14) X seconds before the present moment by the volume flow rate VL of the refrigerant that is flowing through the liquid interconnecting pipe (14) at the present moment, to be the equivalent density ρ (ρ = G'/VL).

[0141] In subsequent step ST14, the subcooling expansion valve control section (93) sets the target liquid refrigerant temperature T_t. Specifically, the subcooling expansion valve control section (93) calculates a temperature at which when the refrigerant pressure is equal to the measured value Ps, the refrigerant density is equal to the equivalent density p, based on the value Ps measured by the liquid refrigerant pressure sensor (87) at the present moment, the equivalent density ρ calculated in step ST13, and the properties of refrigerant, and determines the calculated temperature to be the target liquid refrigerant temperature T_t.

[0142] In subsequent step ST15, the subcooling expansion valve control section (93) controls the degree of opening of the subcooling expansion valve (35) such that a value measured by the liquid refrigerant temperature sensor (82) is equal to the target liquid refrigerant temperature T_t. If the value measured by the liquid refrigerant temperature sensor (82) is higher than the target liquid refrigerant temperature T_t, the subcooling expansion valve control section (93) reduces the degree of opening of the subcooling expansion valve (35). As a result, the temperature of a refrigerant supplied to the second channels (34b) of the subcooling heat exchanger (34) after passing through the subcooling expansion valve (35) decreases, the temperature of a refrigerant flowing out of the first channels (34a) of the subcooling heat exchanger (34) decreases, and the density of a refrigerant supplied from the subcooling heat exchanger (34) to the liquid interconnecting pipe (14) increases. Thus, even if the value (Ps - TL) is less than the reference pressure difference, and the volume flow rate of the refrigerant supplied from the subcooling heat exchanger (34) to the liquid interconnecting pipe (14) is low, the mass flow rate of the refrigerant supplied from the subcooling heat exchanger (34) to the liquid interconnecting pipe (14) is high enough. This allows the utilization-side units (12) to have adequate cooling capability.

INDUSTRIAL APPLICABILITY

[0143] As can be seen from the foregoing description, the present invention is useful for a refrigeration apparatus that circulates a refrigerant through a refrigerant circuit to perform a refrigeration cycle.

DESCRIPTION OF REFERENCE CHARACTERS

[0144]

10 Refrigeration Apparatus
11 Heat-Source-Side Unit
12 Utilization-Side Unit
14 Liquid Interconnecting Pipe
15 Gas Interconnecting Pipe
20 Refrigerant Circuit
31a First Compressor
31b Second Compressor
31c Third Compressor
33 Heat-Source-Side Heat Exchanger
34 Subcooling Heat Exchanger
35 Heat-Source-Side Expansion Valve
53c Third Heat-Source-Side Liquid Pipe (Pipe)
61 Utilization-Side Heat Exchanger
63 Utilization-Side Expansion Valve
62 Utilization-Side Solenoid Valve
90 Controller

Claims

1. A refrigeration apparatus comprising:

a refrigerant circuit (20) including a heat-source-side unit (11) and a plurality of utilization-side units (12), the heat-source-side unit (11) and the utilization-side units (12) being connected together through a liquid interconnecting pipe (14) and a gas interconnecting pipe (15), the utilization-side units (12) being arranged in parallel,
the refrigeration apparatus circulating a refrigerant through the refrigerant circuit (20) to perform a refrigeration cycle, wherein
the heat-source-side unit (11) includes a compressor (31a-31c), a heat-source-side heat exchanger (33), and a heat-source-side expansion valve (38) provided on a pipe (53c) configured to deliver the refrigerant condensed in the heat-source-side heat exchanger (33) to the liquid interconnecting pipe (14),
each of the utilization-side units (12) includes a utilization-side heat exchanger (61), a utilization-side expansion valve (63), and a utilization-side solenoid valve (62), which are arranged in series, and is switchable between a cooling state where the utilization-side solenoid valve (62) opens to allow the utilization-side heat exchanger (61) to function as an evaporator, and a suspended state where the utilization-side solenoid valve (62) closes to interrupt flow of the refrigerant through the utilization-side heat exchanger (61), and
the refrigeration apparatus further includes: a controller (90) configured to perform a pressure control operation to control a degree of opening of the heat-source-side expansion valve (38) to prevent a pressure of the refrigerant in the liquid interconnecting pipe (14) from exceeding a predetermined upper-limit value while one or some of the utilization-side units (12) are in the suspended state.

2. The refrigeration apparatus of claim 1, wherein
while one or some of the utilization-side units (12) are in the suspended state, and a difference in pressure between the refrigerant in the liquid interconnecting pipe (14) and the refrigerant in the gas interconnecting pipe (15) is larger than or equal to a predetermined upper-limit pressure difference, the controller (90) performs the pressure control operation.

3. The refrigeration apparatus of claim 2, wherein
the controller (90) that is performing the pressure control operation adjusts the degree of opening of the heat-source-side expansion valve (38) such that the difference in pressure between the refrigerant in the liquid interconnecting pipe (14) and the refrigerant in the gas interconnecting pipe (15) is larger than or equal to a lower-limit pressure difference that is smaller than the upper-limit pressure difference.

4. The refrigeration apparatus of any one of claims 1-3, wherein
the heat-source-side unit (11) includes a subcooling heat exchanger (34) configured to exchange heat between a liquid refrigerant condensed in the heat-source-side heat exchanger (33) and delivered to the liquid interconnecting pipe (14) and a cooling fluid to cool the liquid refrigerant.

5. The refrigeration apparatus of claim 4, wherein
the heat-source-side unit (11) includes
a subcooling pipe (54m) configured to supply part of the refrigerant condensed in the heat-source-side heat exchanger (33) as the cooling fluid to the subcooling heat exchanger (34), and
a subcooling expansion valve (35) provided on the subcooling pipe (54m), and the controller (90) is configured to control a degree of opening of the subcooling expansion valve (35) to reduce a temperature of a liquid refrigerant delivered from the subcooling heat exchanger (34) to the liquid interconnecting pipe (14) if a difference in pressure between the refrigerant in the liquid interconnecting pipe (14) and the refrigerant in the gas interconnecting pipe (15) is smaller than a predetermined reference pressure difference.

FIG.1

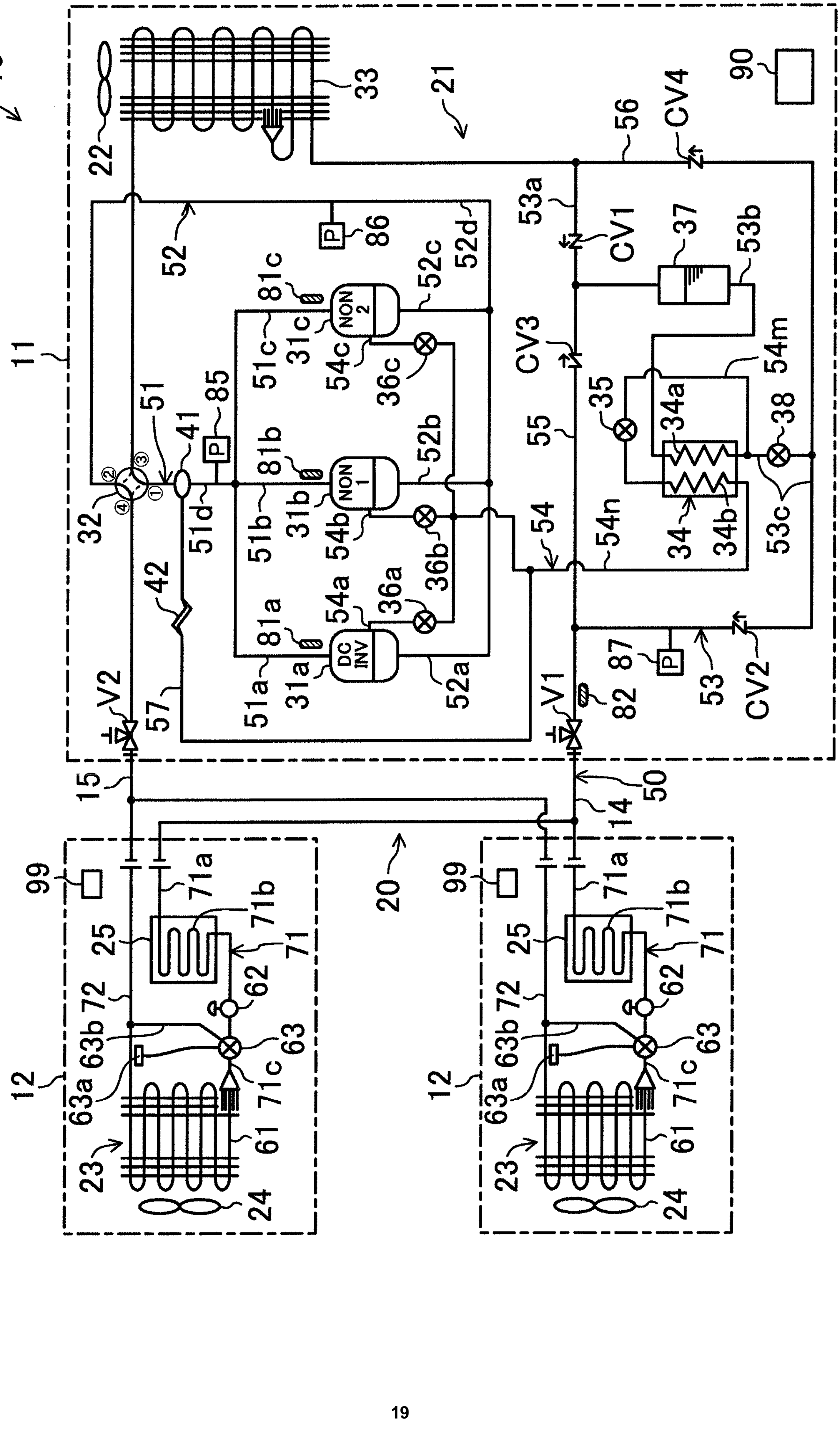
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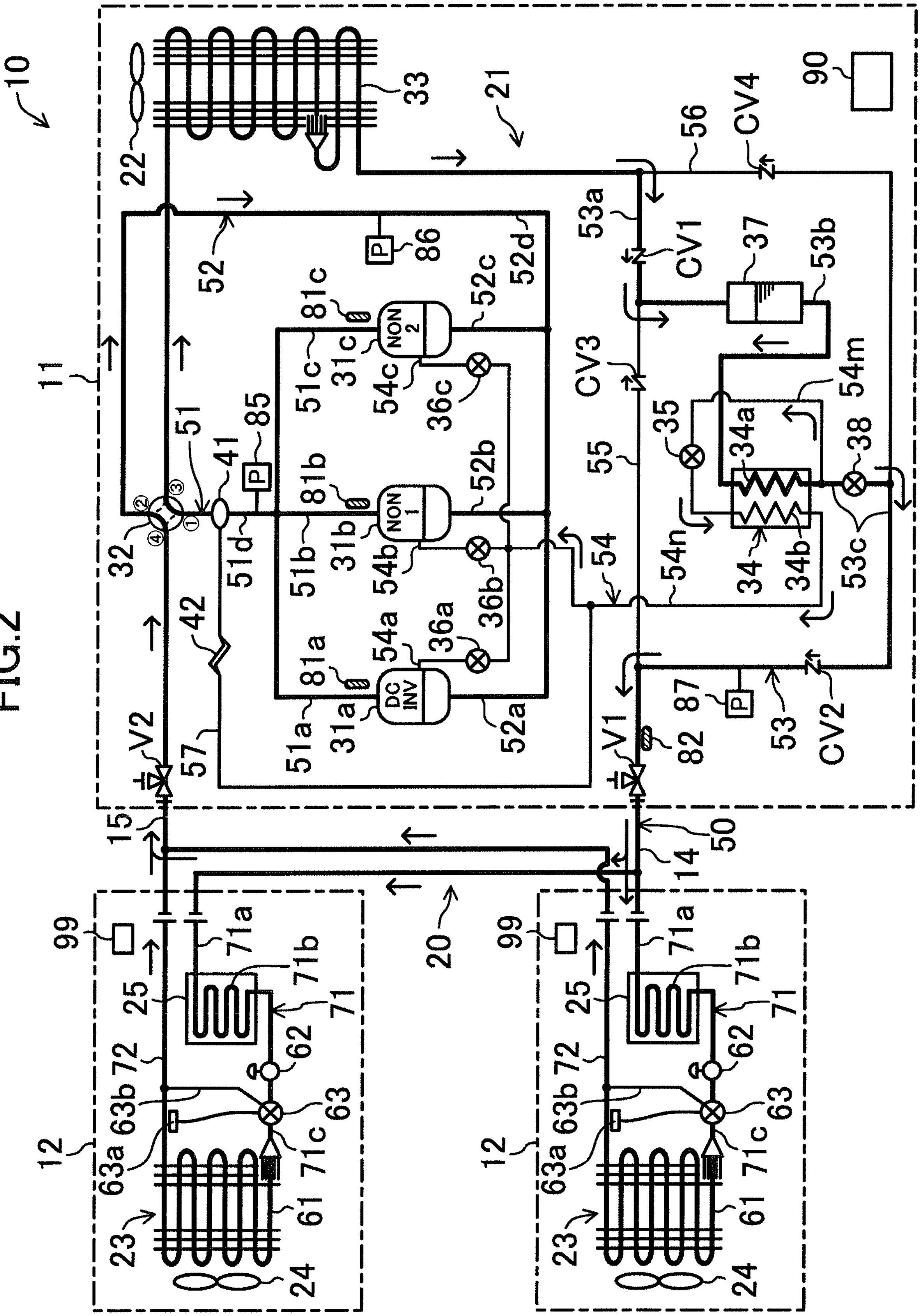
FIG.2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63a
63b
71
71a
71b
71c
72
81a
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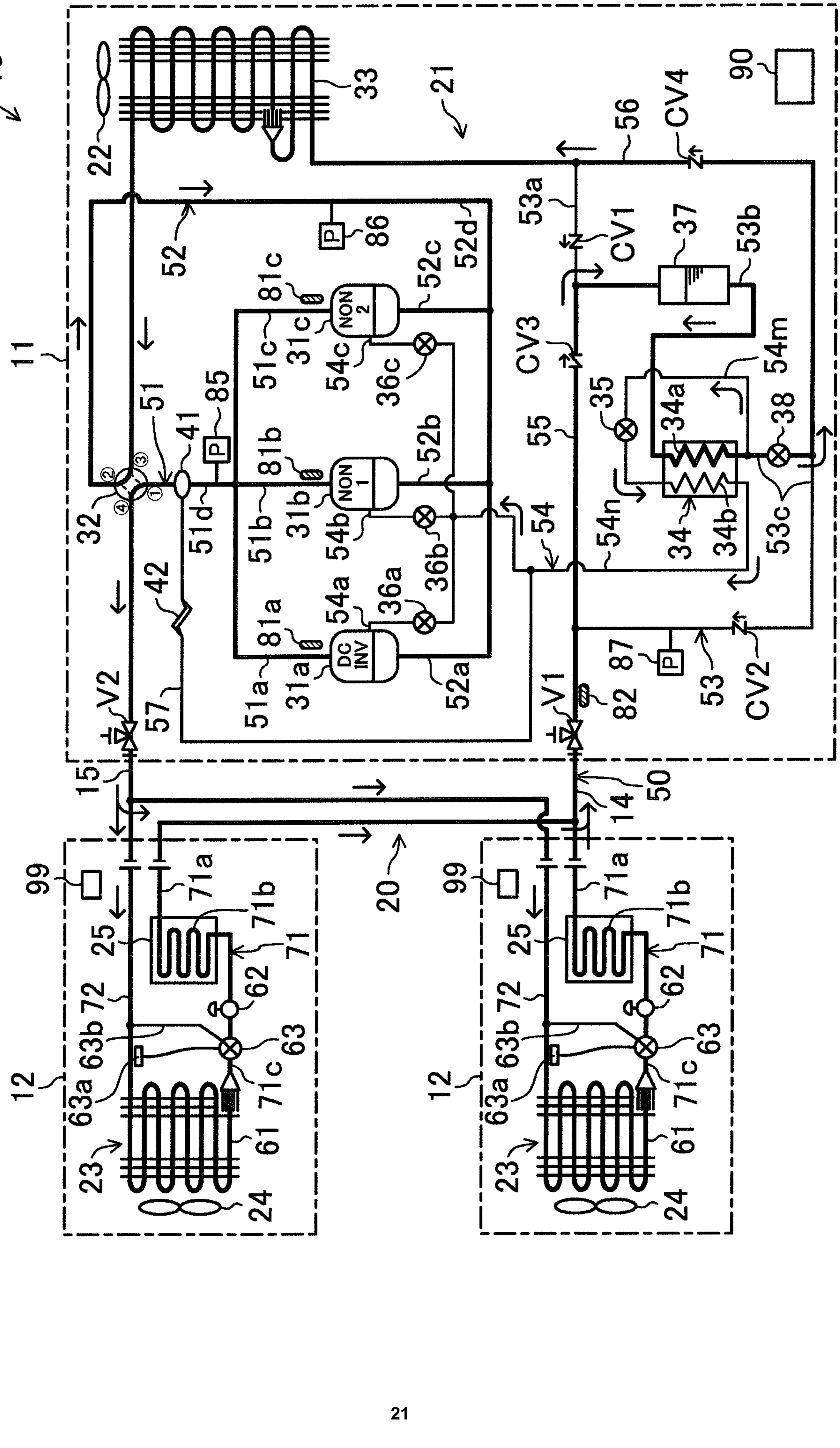
99
CV1
CV2
CV3
CV4
V1
V2
DC INV
NON 1
NON 2
P

FIG.3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25
32
33
34
34a
34b
35
36a
36b
36c
37
38
41
42
50
51
51a
51b
51c
51d
52
52a
52b
52c
52d
53
53a
53b
53c
54
54a
54b
54c
54m
54n
55
56
57
61
62
63
63a
63b
71
71a
71b
71c
72
81a
81b
81c
82
85
86
87
90
99
31a
31b
31c
DC INV
NON 1
NON 2
P
V1
V2
CV1
CV2
CV3
CV4

FIG.4

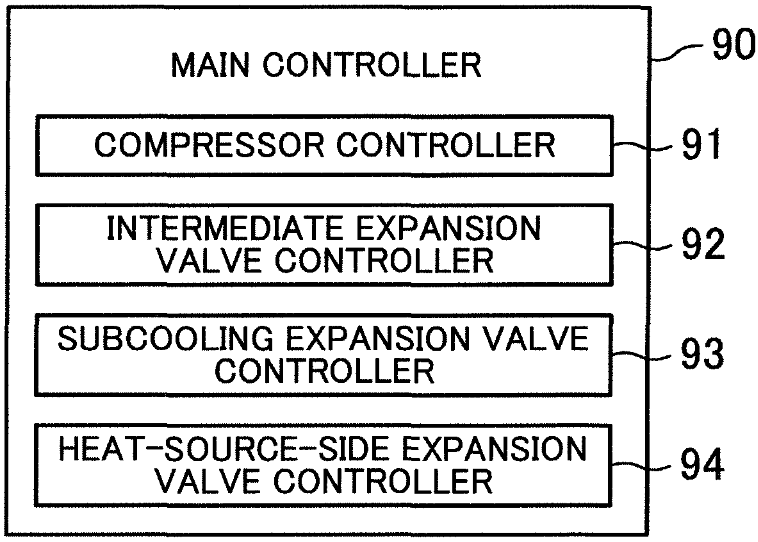

MAIN CONTROLLER
90
COMPRESSOR CONTROLLER
91
INTERMEDIATE EXPANSION VALVE CONTROLLER
92
SUBCOOLING EXPANSION VALVE CONTROLLER
93
HEAT-SOURCE-SIDE EXPANSION VALVE CONTROLLER
94

FIG.5

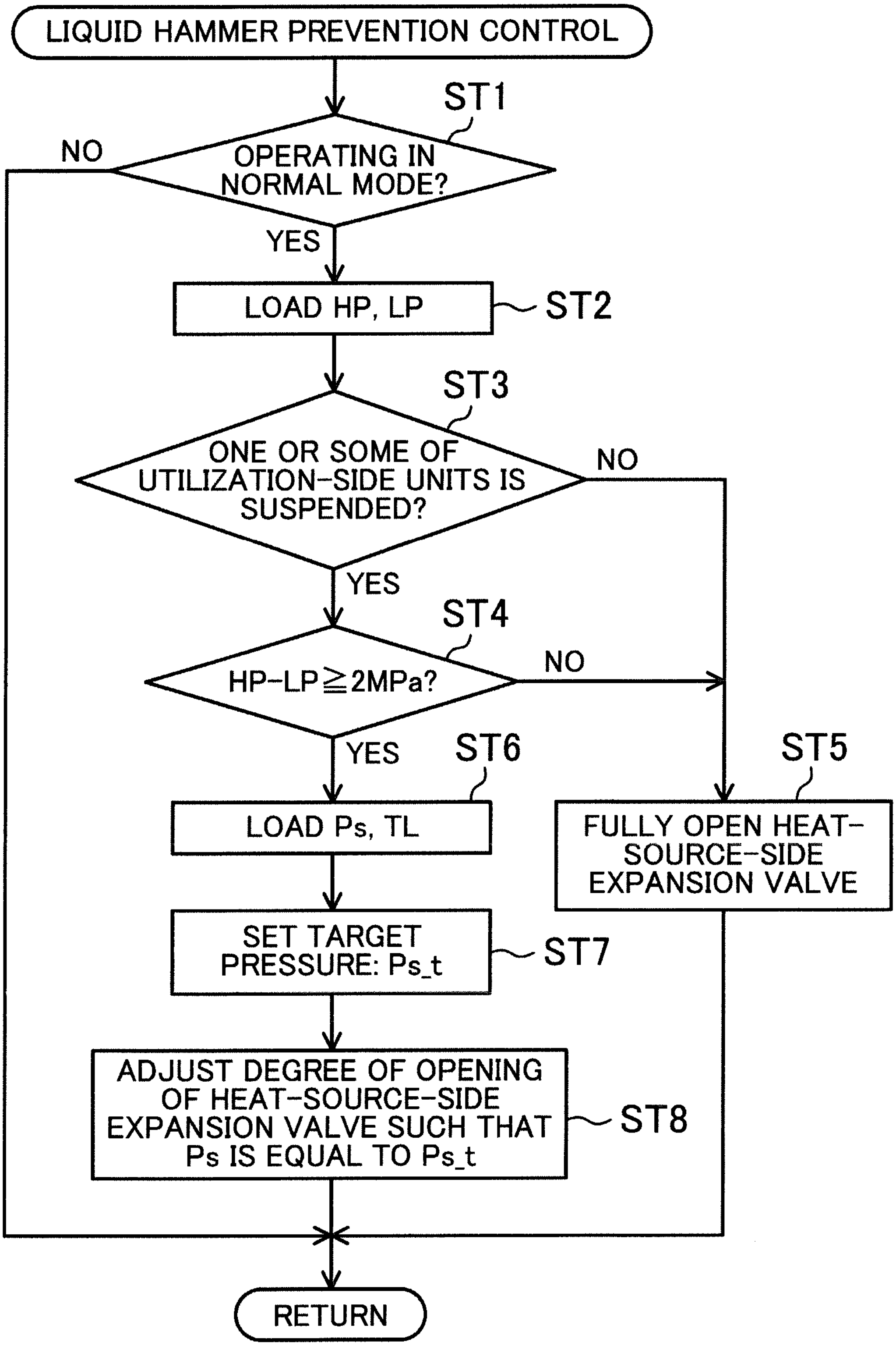
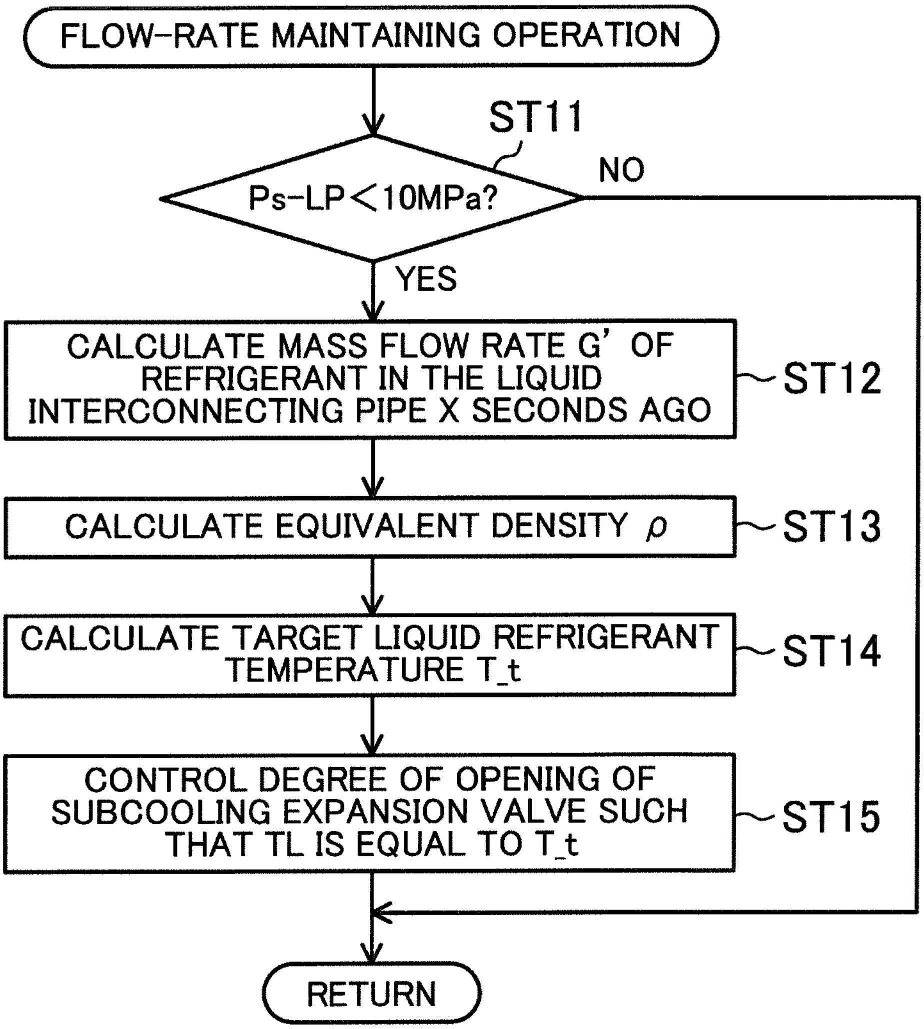

LIQUID HAMMER PREVENTION CONTROL
ST1
OPERATING IN NORMAL MODE?
NO
YES
LOAD HP, LP
ST2
ST3
ONE OR SOME OF UTILIZATION-SIDE UNITS IS SUSPENDED?
NO
YES
ST4
HP-LP≧2MPa?
NO
YES
ST6
LOAD Ps, TL
ST5
FULLY OPEN HEAT-SOURCE-SIDE EXPANSION VALVE
SET TARGET PRESSURE: Ps_t
ST7
ADJUST DEGREE OF OPENING OF HEAT-SOURCE-SIDE EXPANSION VALVE SUCH THAT Ps IS EQUAL TO Ps_t
ST8
RETURN

FIG.6


FLOW-RATE MAINTAINING OPERATION
ST11
Ps-LP<10MPa?
NO
YES
CALCULATE MASS FLOW RATE G' OF REFRIGERANT IN THE LIQUID INTERCONNECTING PIPE X SECONDS AGO
ST12
CALCULATE EQUIVALENT DENSITY ρ
ST13
CALCULATE TARGET LIQUID REFRIGERANT TEMPERATURE T_t
ST14
CONTROL DEGREE OF OPENING OF SUBCOOLING EXPANSION VALVE SUCH THAT TL IS EQUAL TO T_t
ST15
RETURN

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2016/002239

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

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

B. FIELDS SEARCHED

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

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	1996-2016
Kokai Jitsuyo Shinan Koho	1971-2016	Toroku Jitsuyo Shinan Koho	1994-2016

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 A	WO 2012/172599 A1 (Mitsubishi Electric Corp.), 20 December 2012 (20.12.2012), paragraphs [0001] to [0088]; fig. 1 to 5 1 & US 2014/0083126 A1 paragraphs [0001] to [0149]; fig. 1 to 5 & EP 2722616 A1 & CN 104204691 A	1-2,4 3,5
Y	JP 8-233379 A (Mitsubishi Heavy Industries, Ltd.), 13 September 1996 (13.09.1996), paragraphs [0006] to [0008], [0017] to [0027]; fig. 1 to 2 (Family: none)	1-2,4

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

* Special categories of cited documents:
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"E" earlier application or patent but published on or after the international filing date
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"O" document referring to an oral disclosure, use, exhibition or other means
"P" document published prior to the international filing date but later than the priority date claimed
"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
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"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
19 July 2016 (19.07.16)	26 July 2016 (26.07.16)

Name and mailing address of the ISA/	Authorized officer
Japan Patent Office 3-4-3,Kasumigaseki,Chiyoda-ku, Tokyo 100-8915,Japan	Telephone No.

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INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2016/002239

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 2-187567 A (Mitsubishi Electric Corp.), 23 July 1990 (23.07.1990), specification, page 1, lower left column, line 16 to page 4, upper left column, line 18; fig. 1 to 4 (Family: none)	1-2,4
A	JP 2013-68344 A (Daikin Industries, Ltd.), 18 April 2013 (18.04.2013), paragraphs [0001] to [0117]; fig. 1 to 7 (Family: none)	1-5

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REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP H11325654 B **[0005]**