

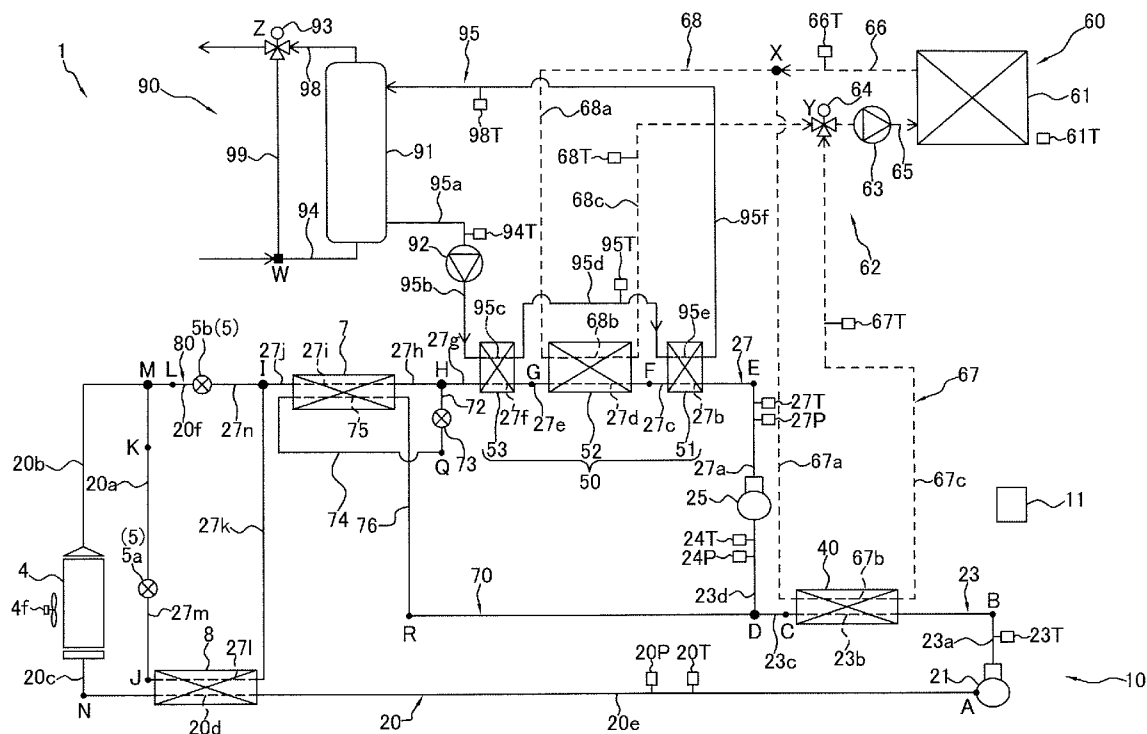


US 20120000237A1

(19) **United States**(12) **Patent Application Publication**
Yamada et al.(10) **Pub. No.: US 2012/0000237 A1**(43) **Pub. Date: Jan. 5, 2012**(54) **HEAT PUMP SYSTEM****Publication Classification**(75) Inventors: **Takuro Yamada**, Osaka (JP);
Noriyuki Okuda, Osaka (JP);
Shuji Fujimoto, Osaka (JP);
Atsushi Yoshimi, Osaka (JP)(51) **Int. Cl.**
F25B 30/00 (2006.01)(52) **U.S. Cl.** **62/324.6**(57) **ABSTRACT**(73) Assignee: **DAIKIN INDUSTRIES, LTD.**,
Osaka-shi, Osaka (JP)(21) Appl. No.: **13/256,270**(22) PCT Filed: **Mar. 10, 2010**(86) PCT No.: **PCT/JP2010/001698**§ 371 (c)(1),
(2), (4) Date: **Sep. 13, 2011**(30) **Foreign Application Priority Data**

Mar. 13, 2009 (JP) 2009-061902

A heat pump system includes a heat pump circuit, a heat load circuit, first and second heat exchangers, a flow rate adjustment element, and a controller. The heat pump circuit circulates primary refrigerant through a low and high stage-side compression elements, an expansion element and an evaporator. The heat load circuit circulates a first fluid and has a first and second branching portions, first and second branching channels, and a heat-load-processing section. The first and second heat exchangers perform heat exchange between the primary refrigerant and the first fluid. Flow rate of the first fluid in the first and/or second branching channel is adjustable. The controller performs flow rate adjustment control so as to maintain a state in which a predetermined temperature condition is satisfied, or to reduce a difference between the temperature of the first fluid flowing through portions of the first and second branching channels.



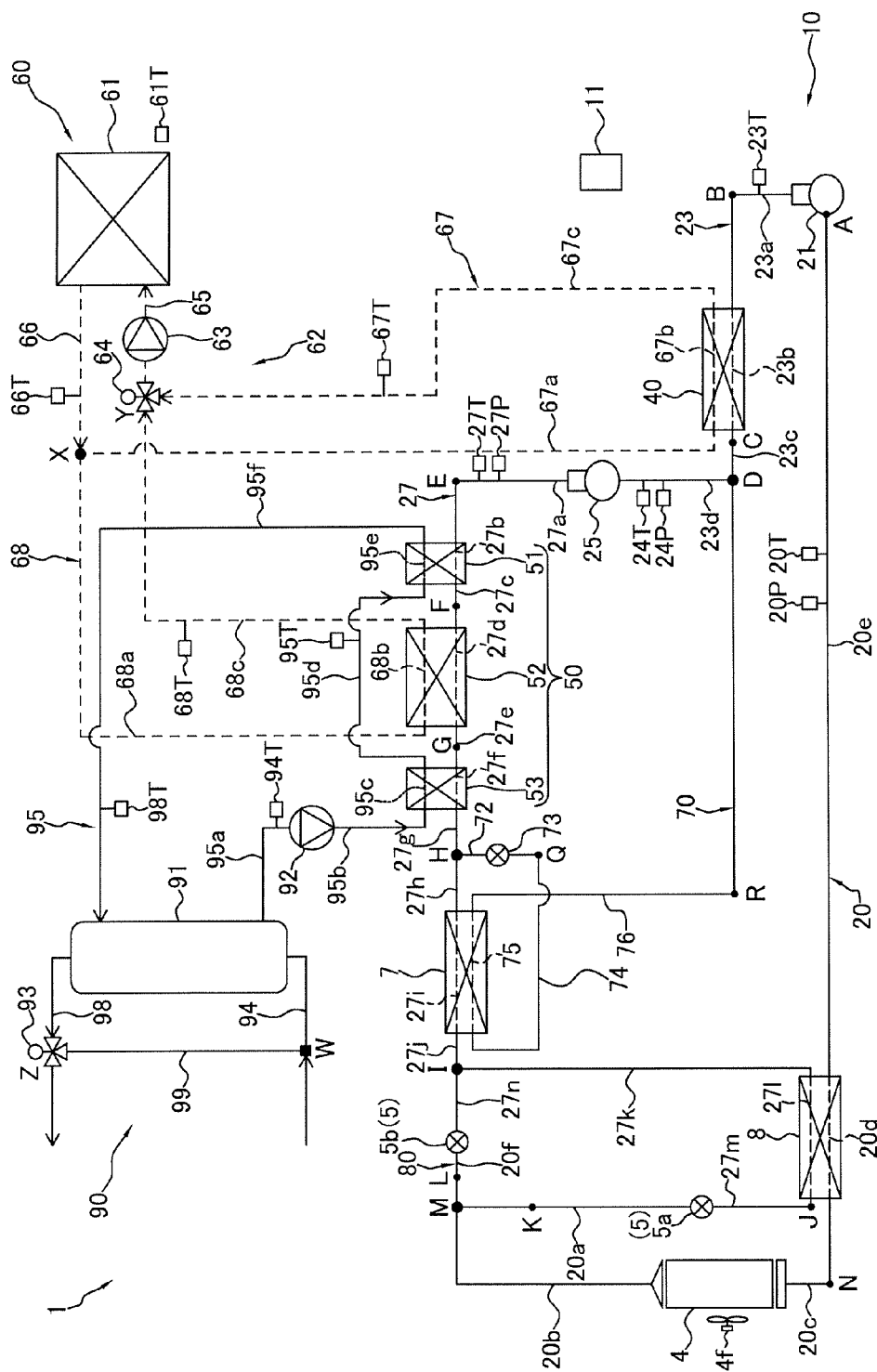


FIG. 1

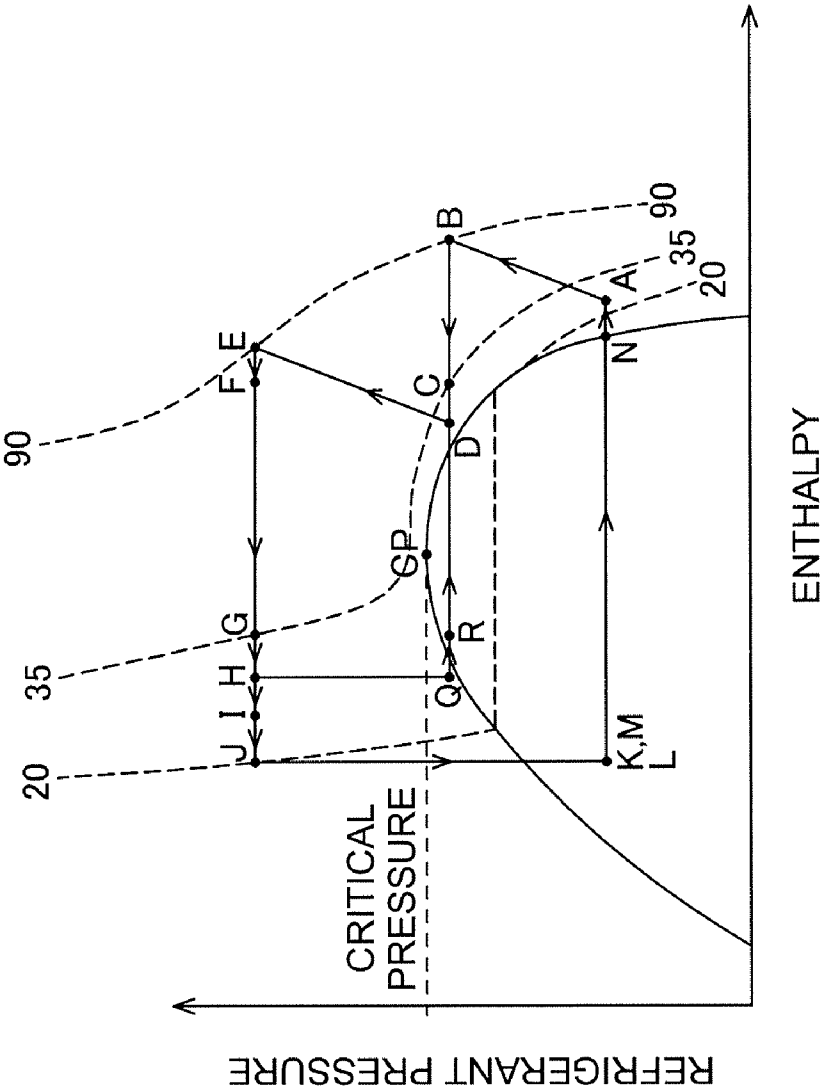


FIG. 2

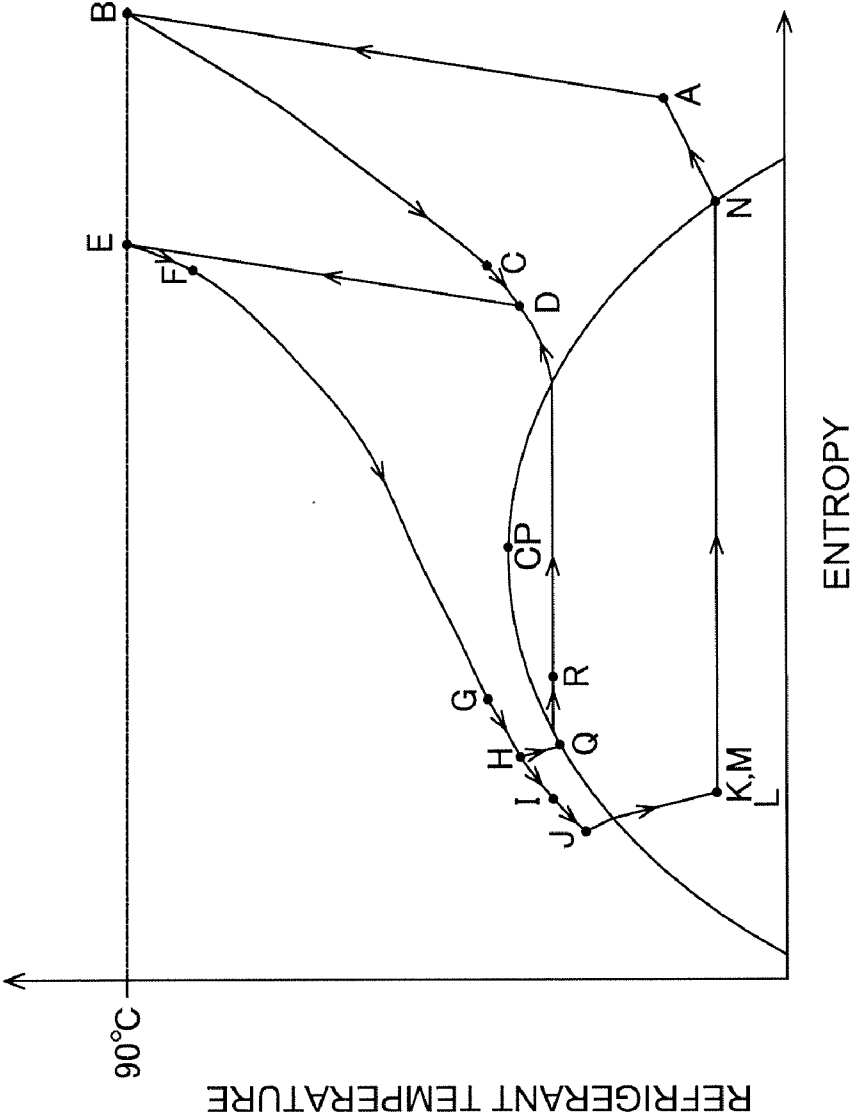


FIG. 3

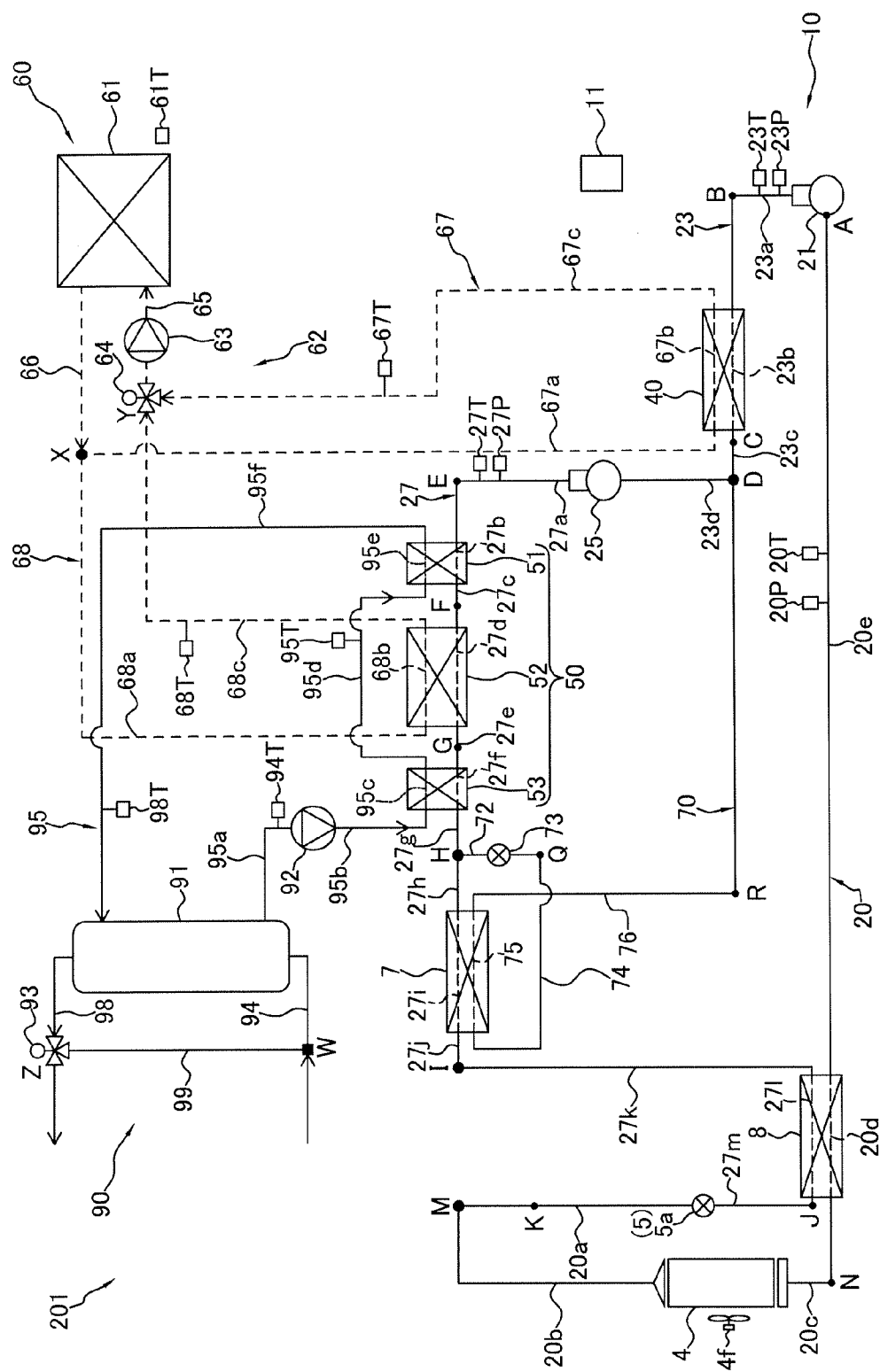


FIG. 4

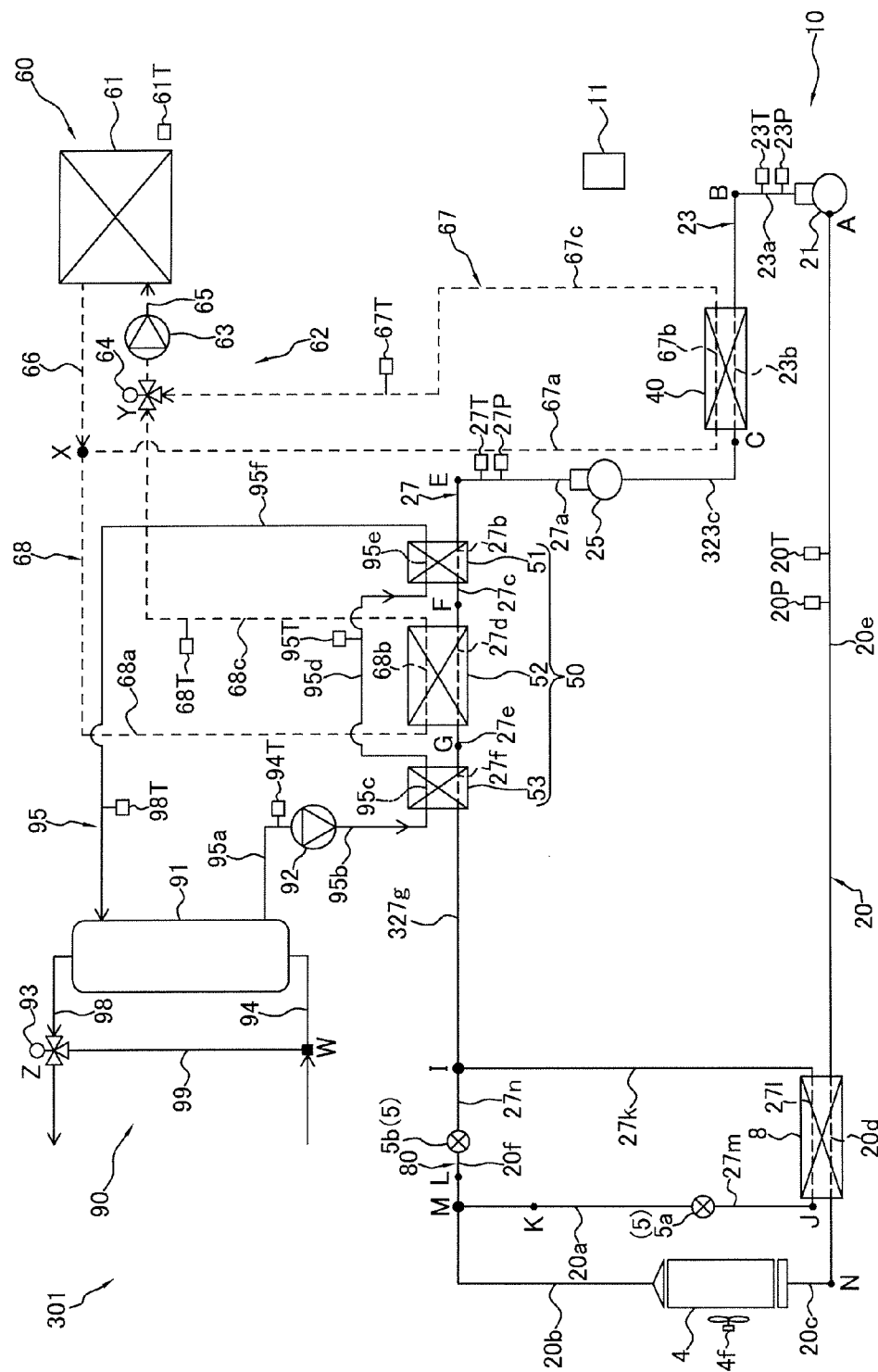


FIG. 5

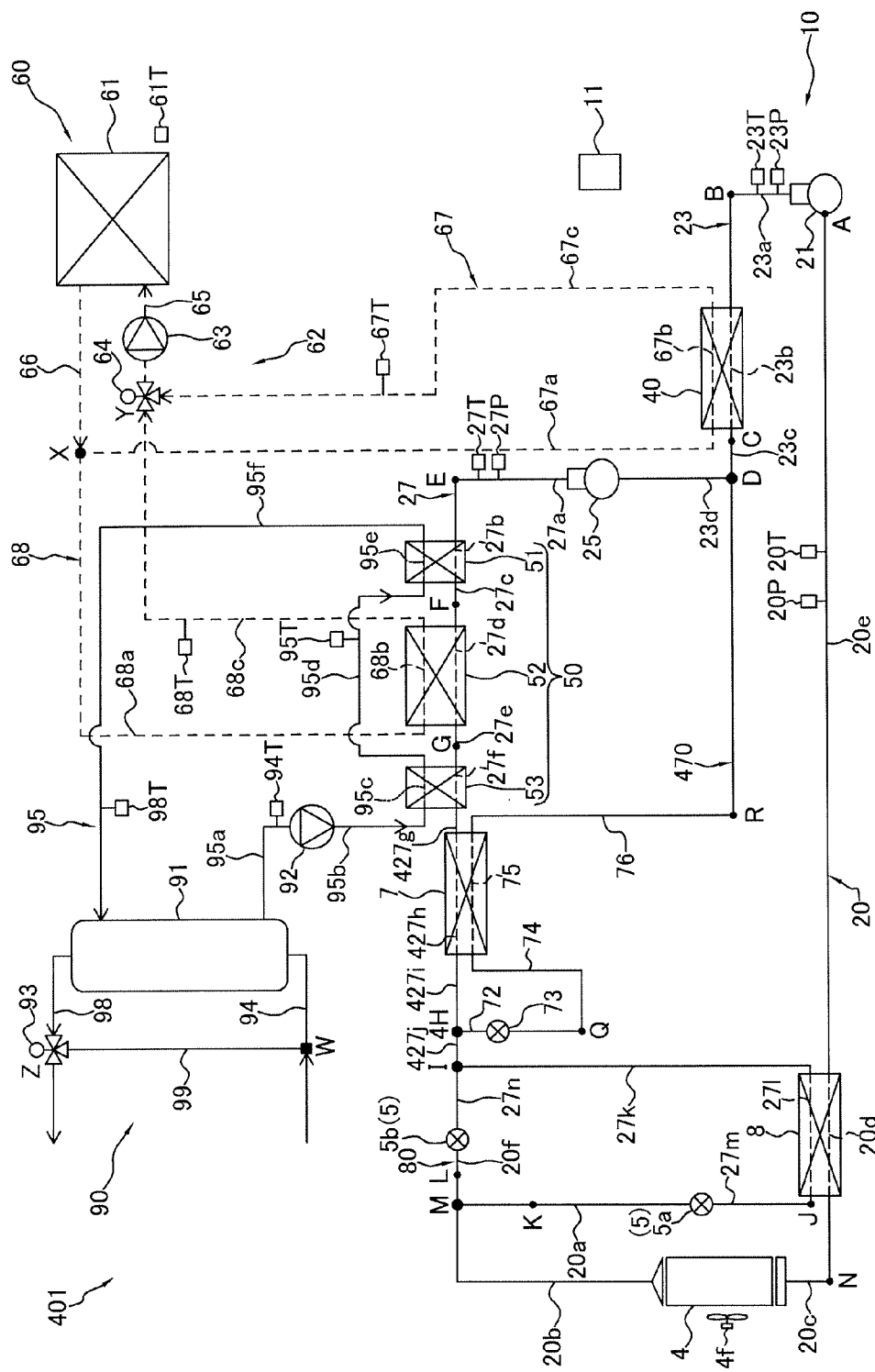


FIG. 6

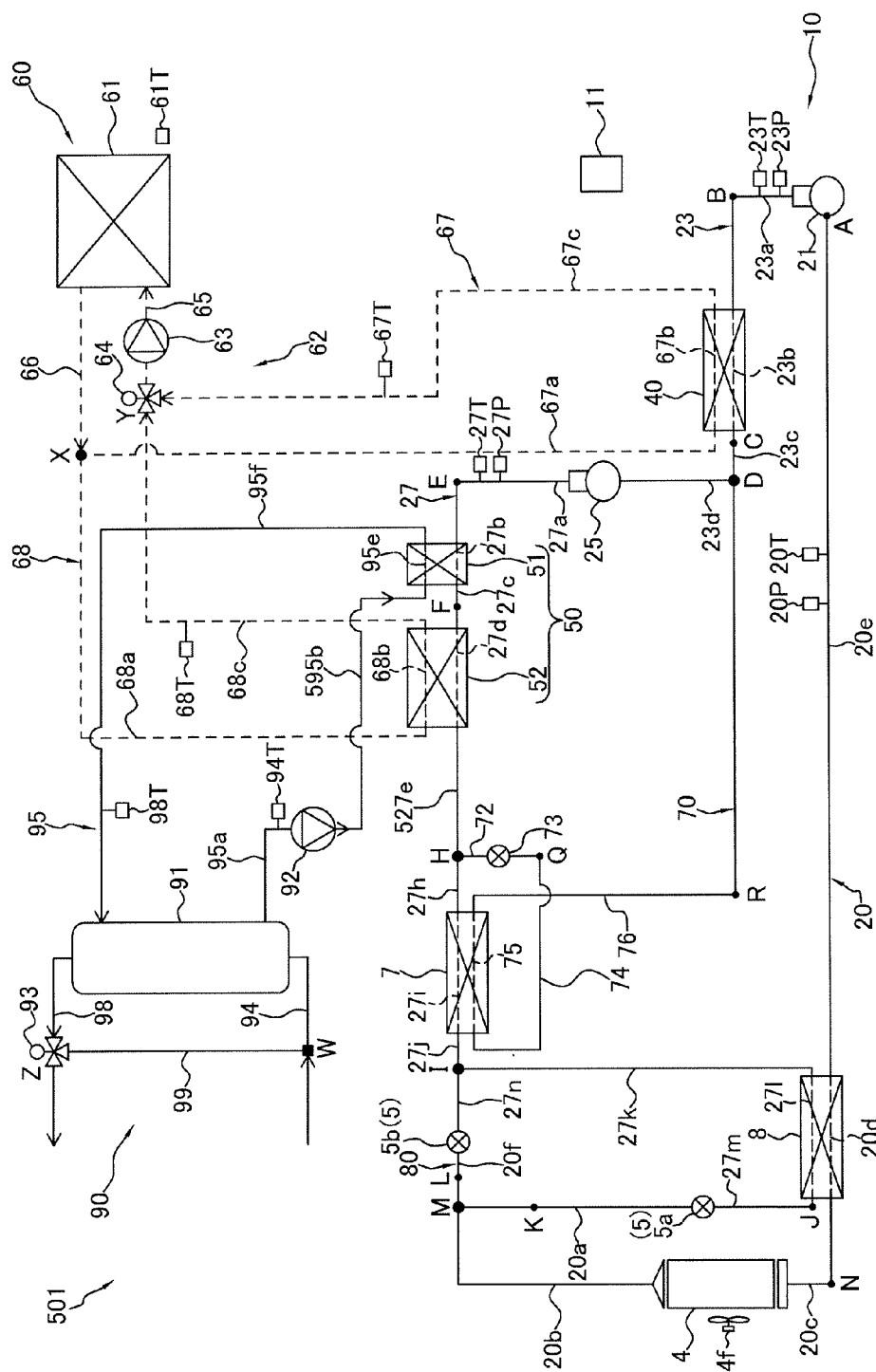


FIG. 7

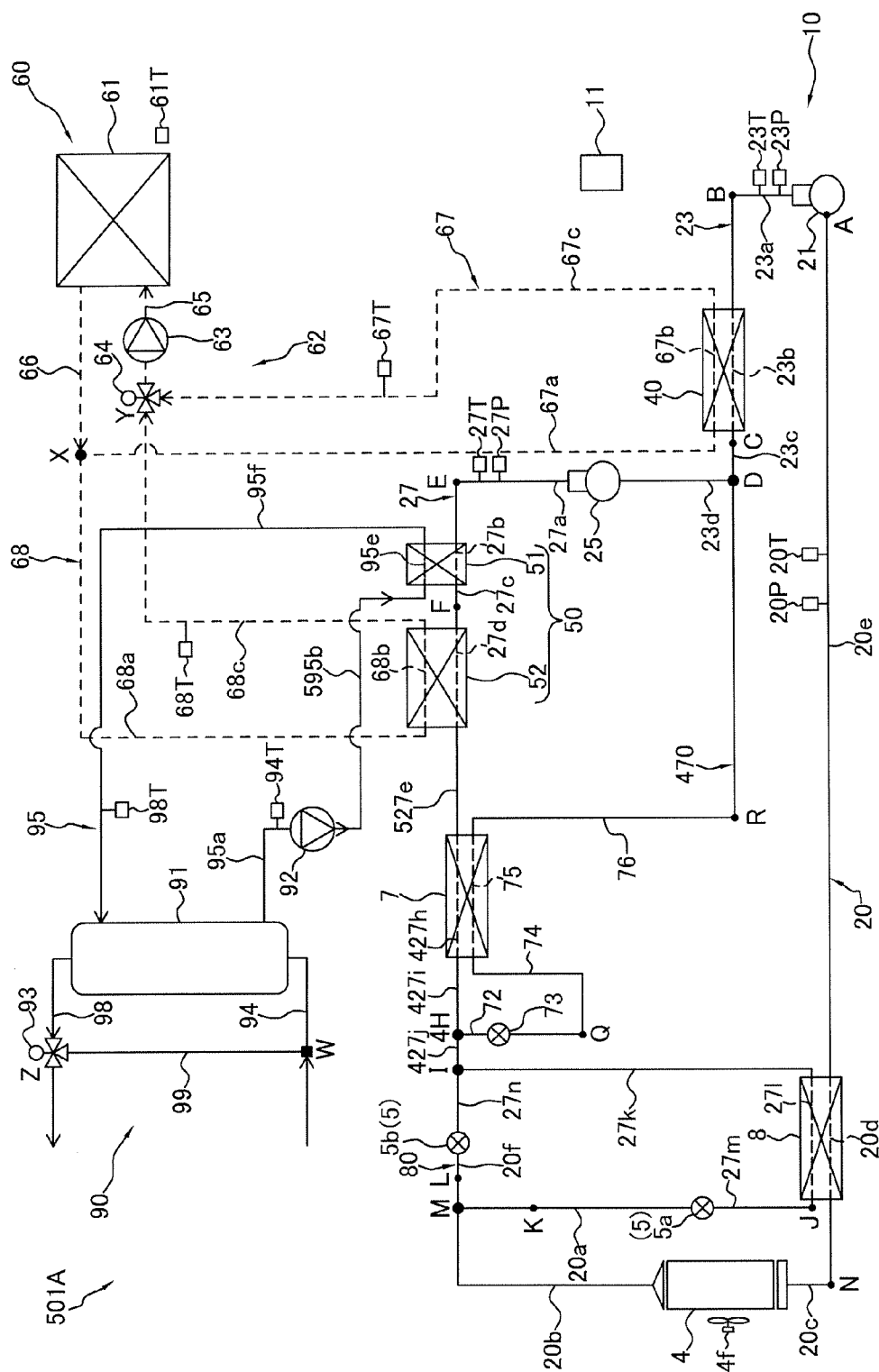


FIG. 8

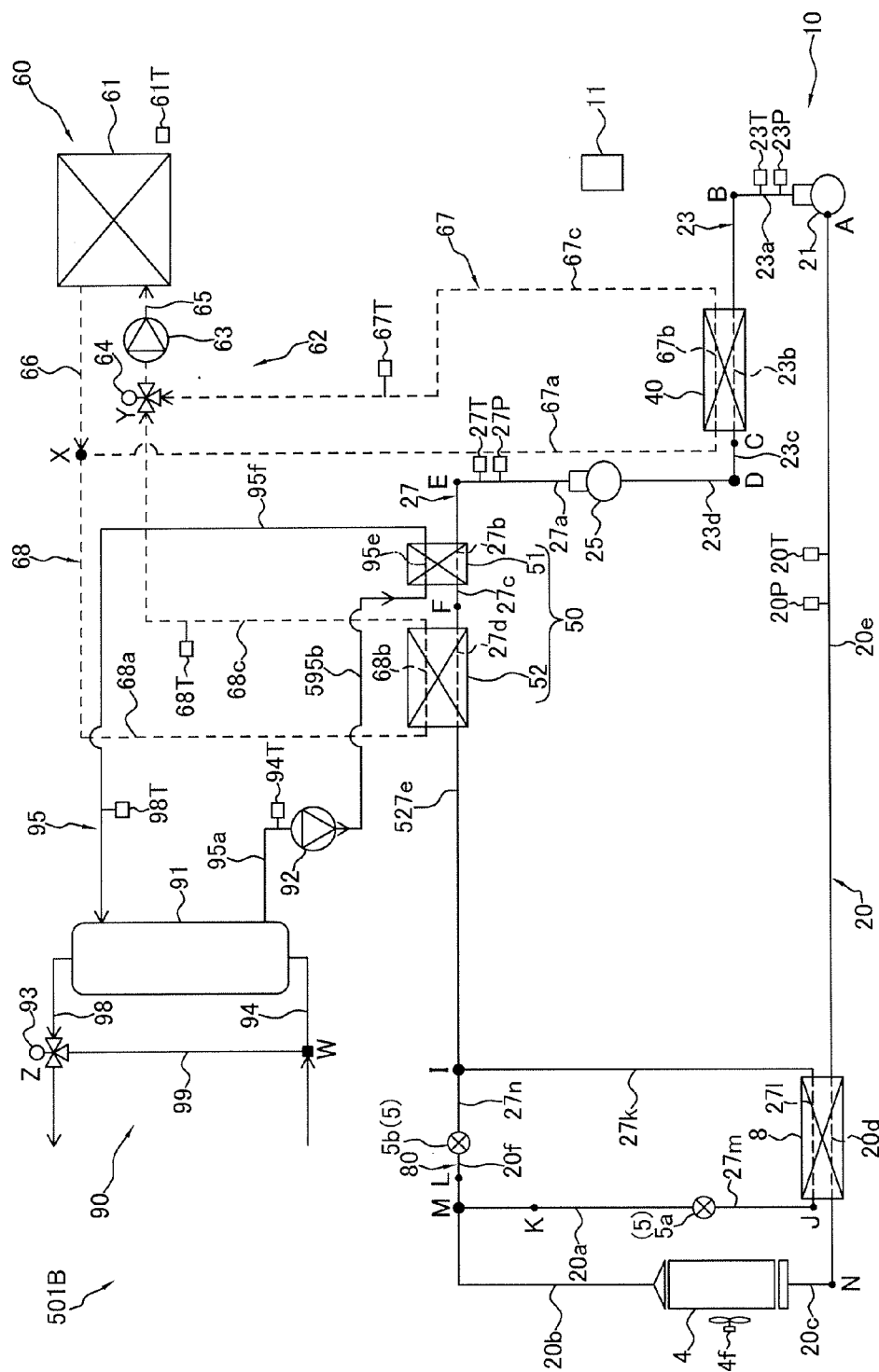


FIG. 9

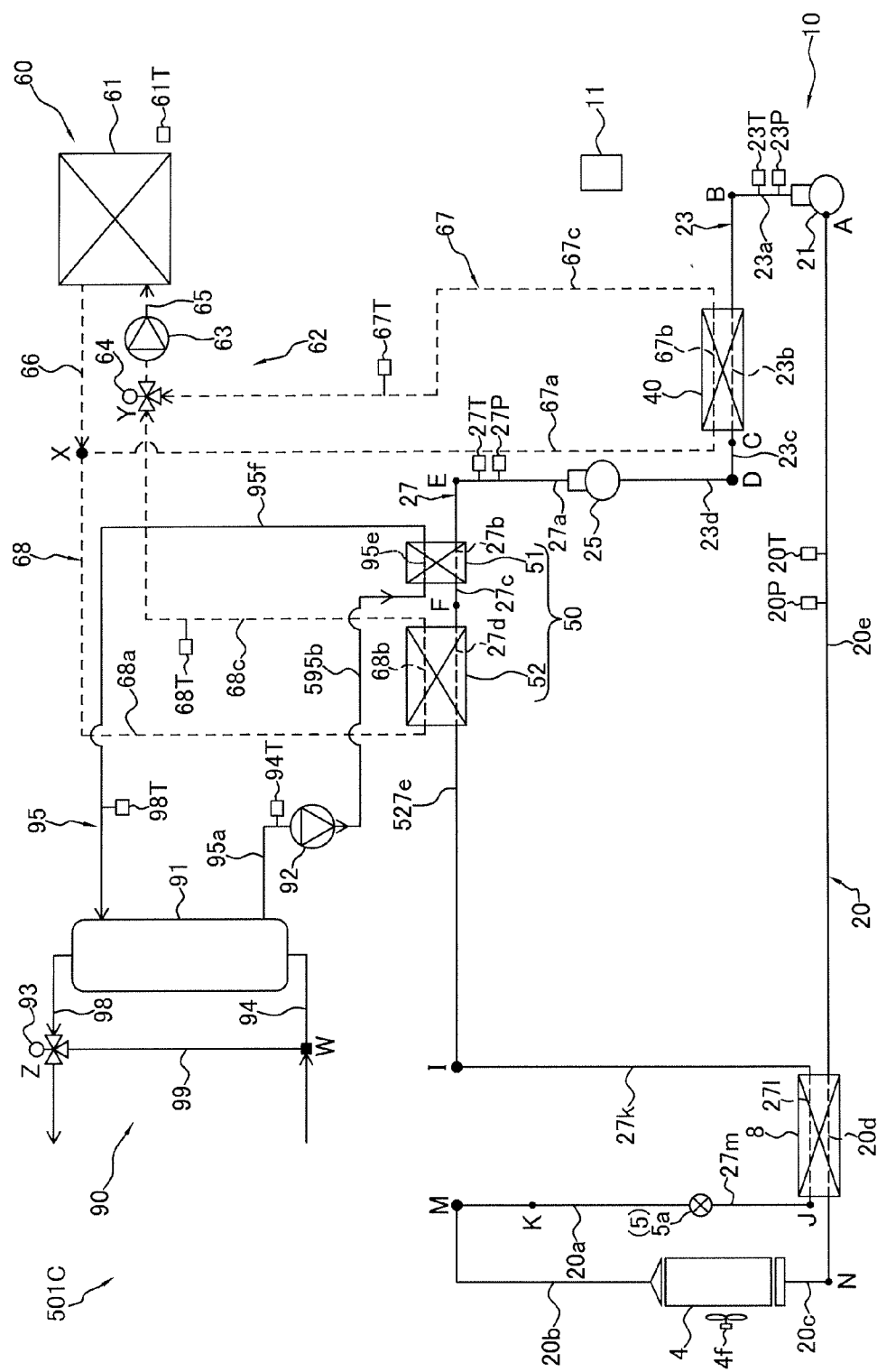


FIG. 10

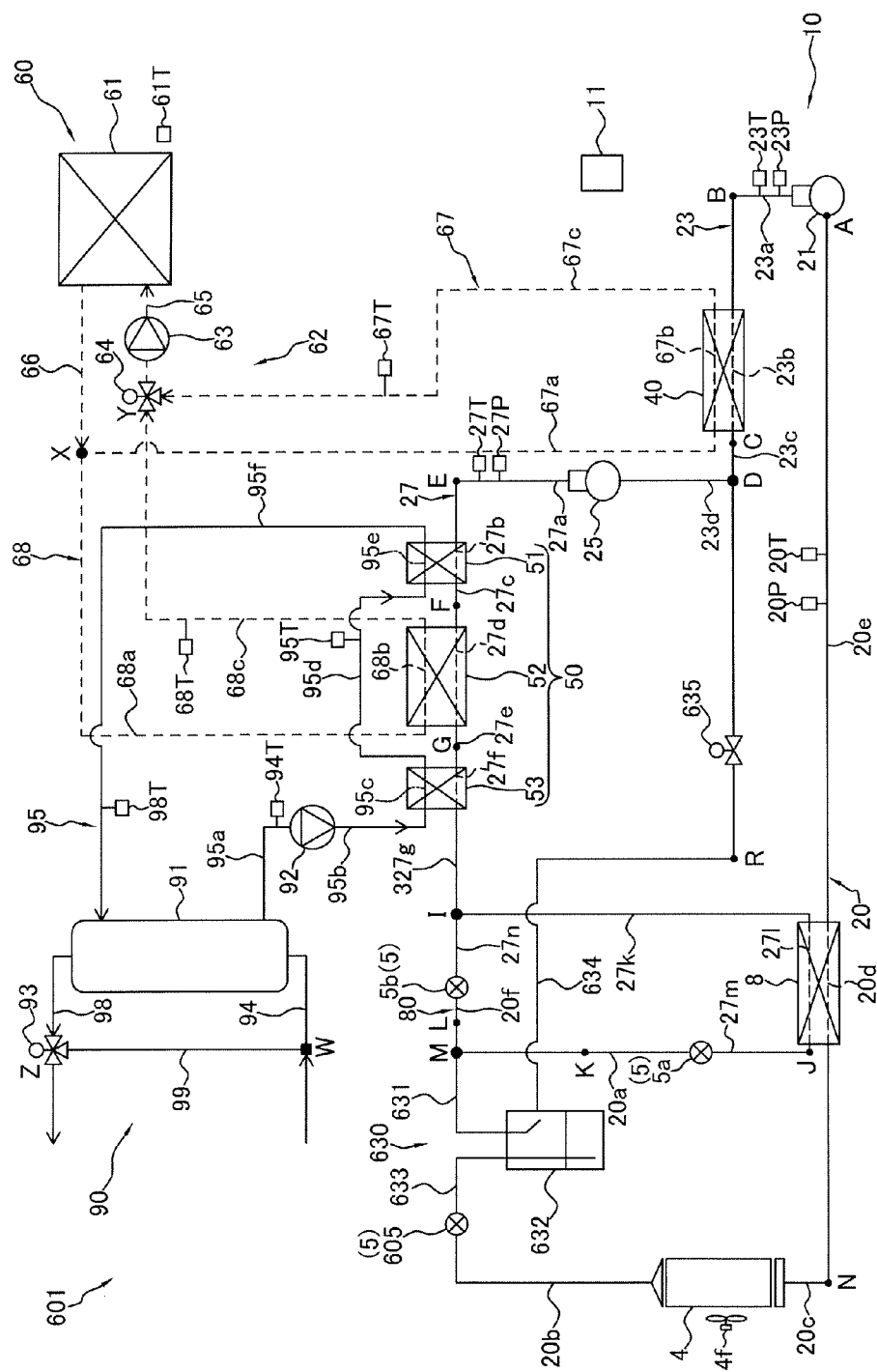


FIG. 11

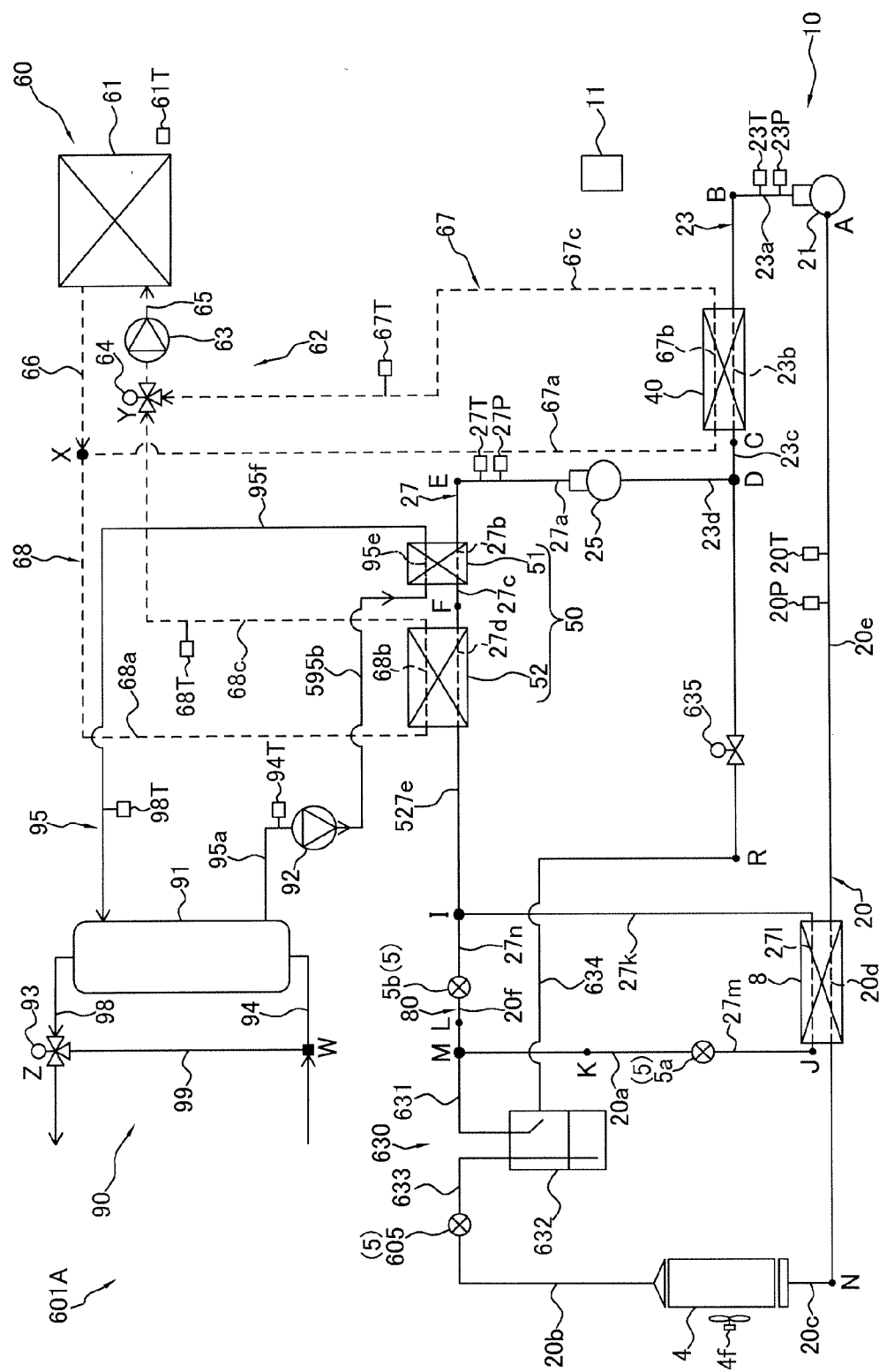


FIG. 12

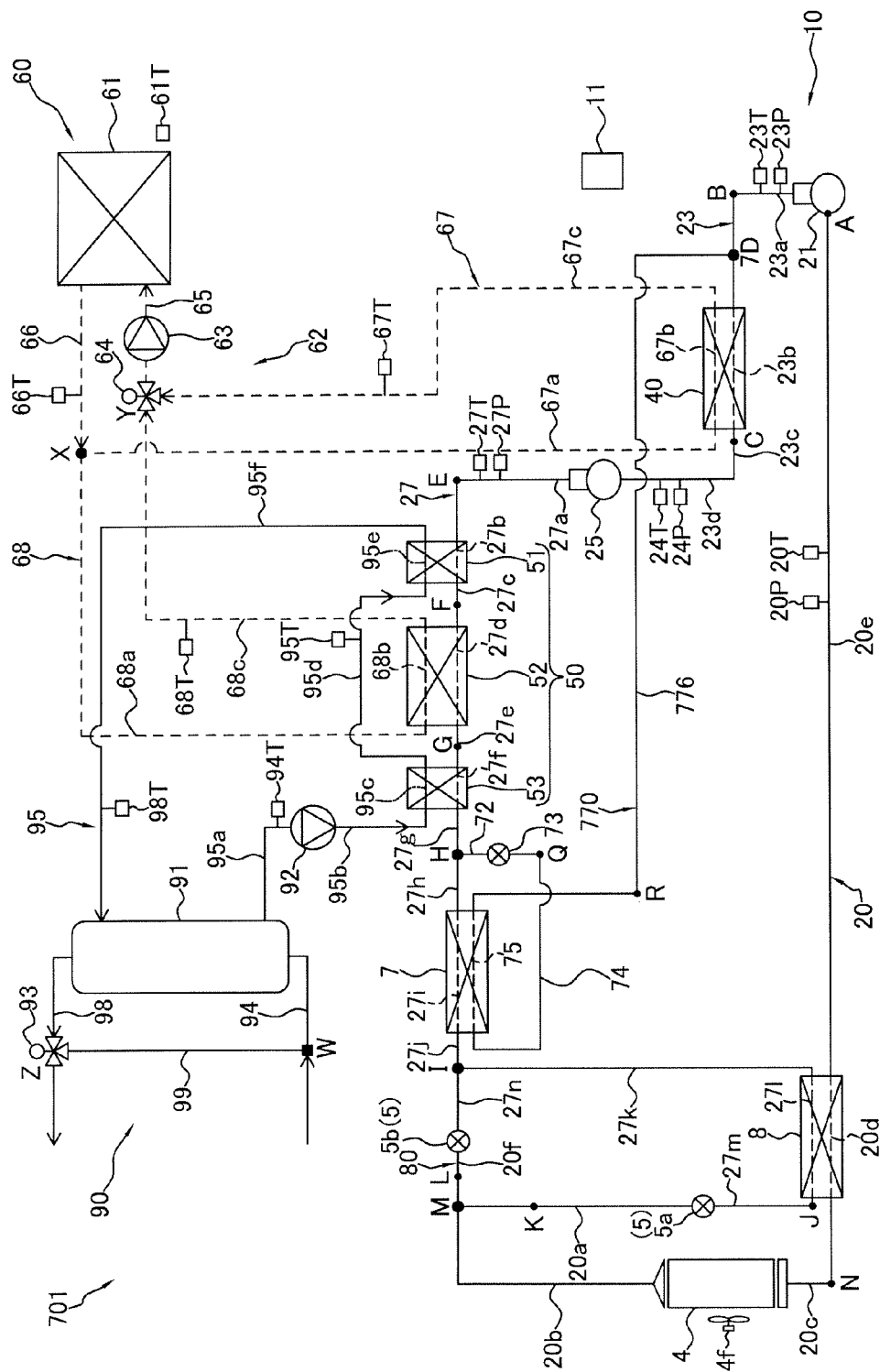


FIG. 13

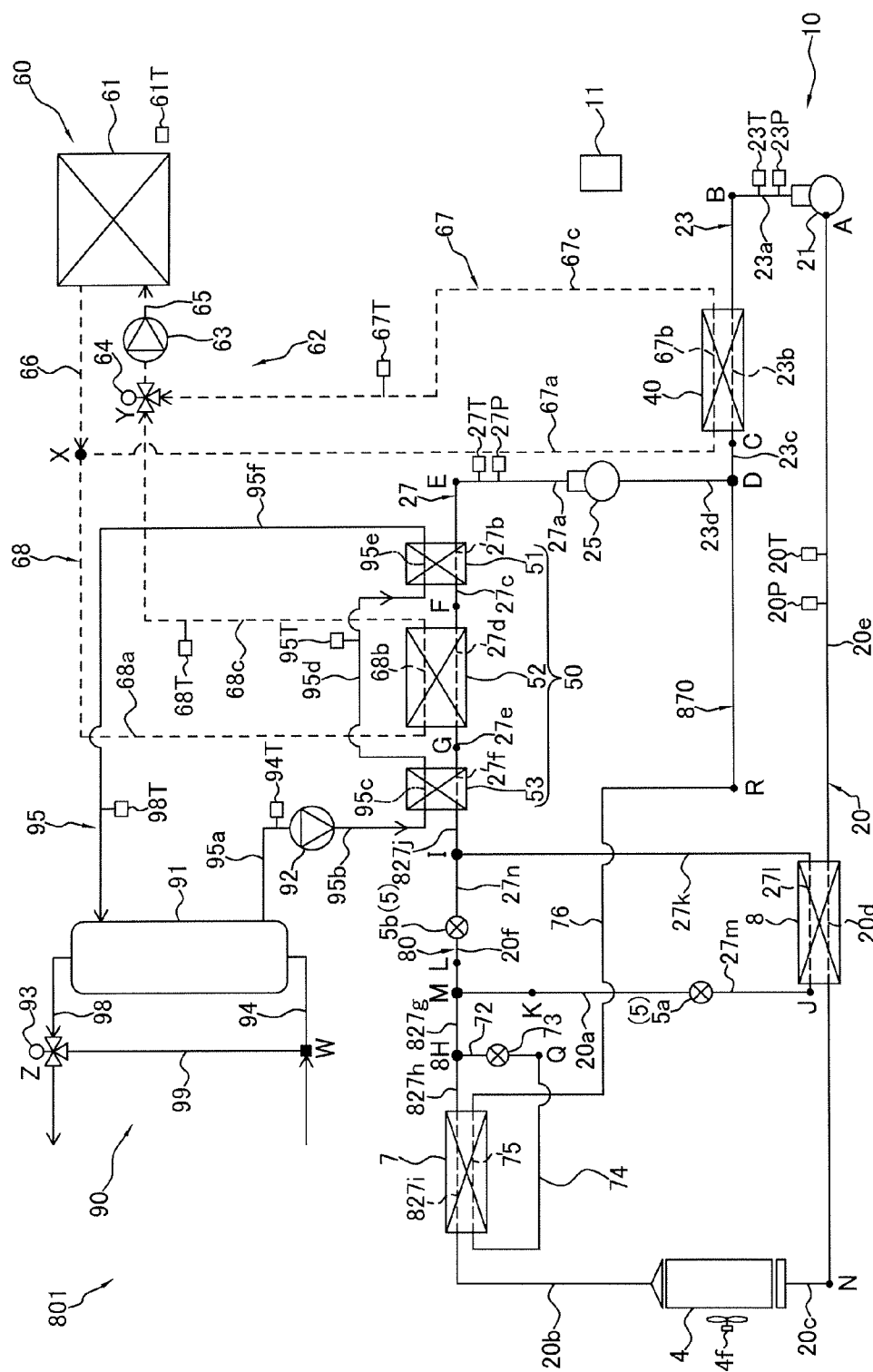


FIG. 14

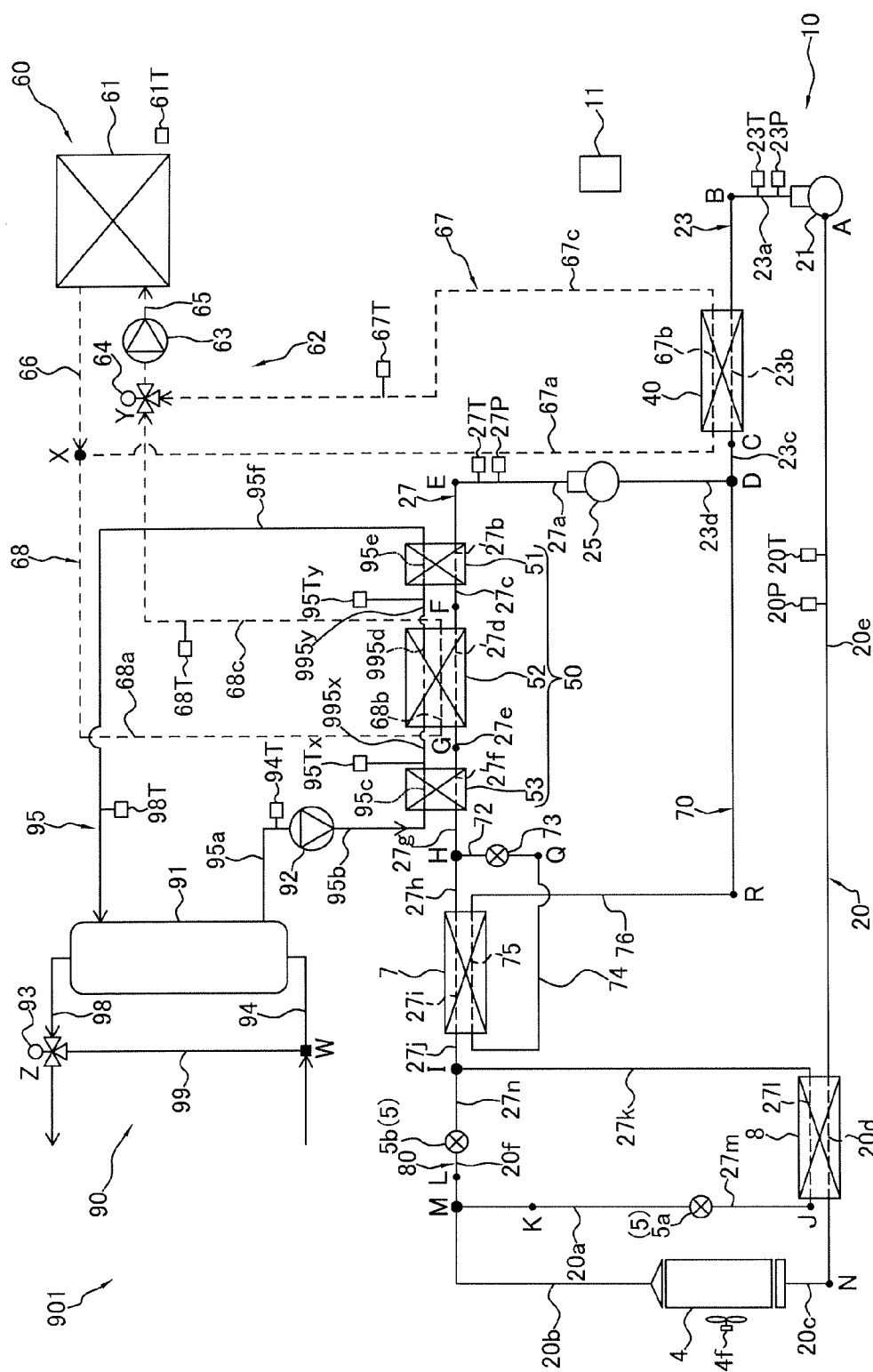


FIG. 15

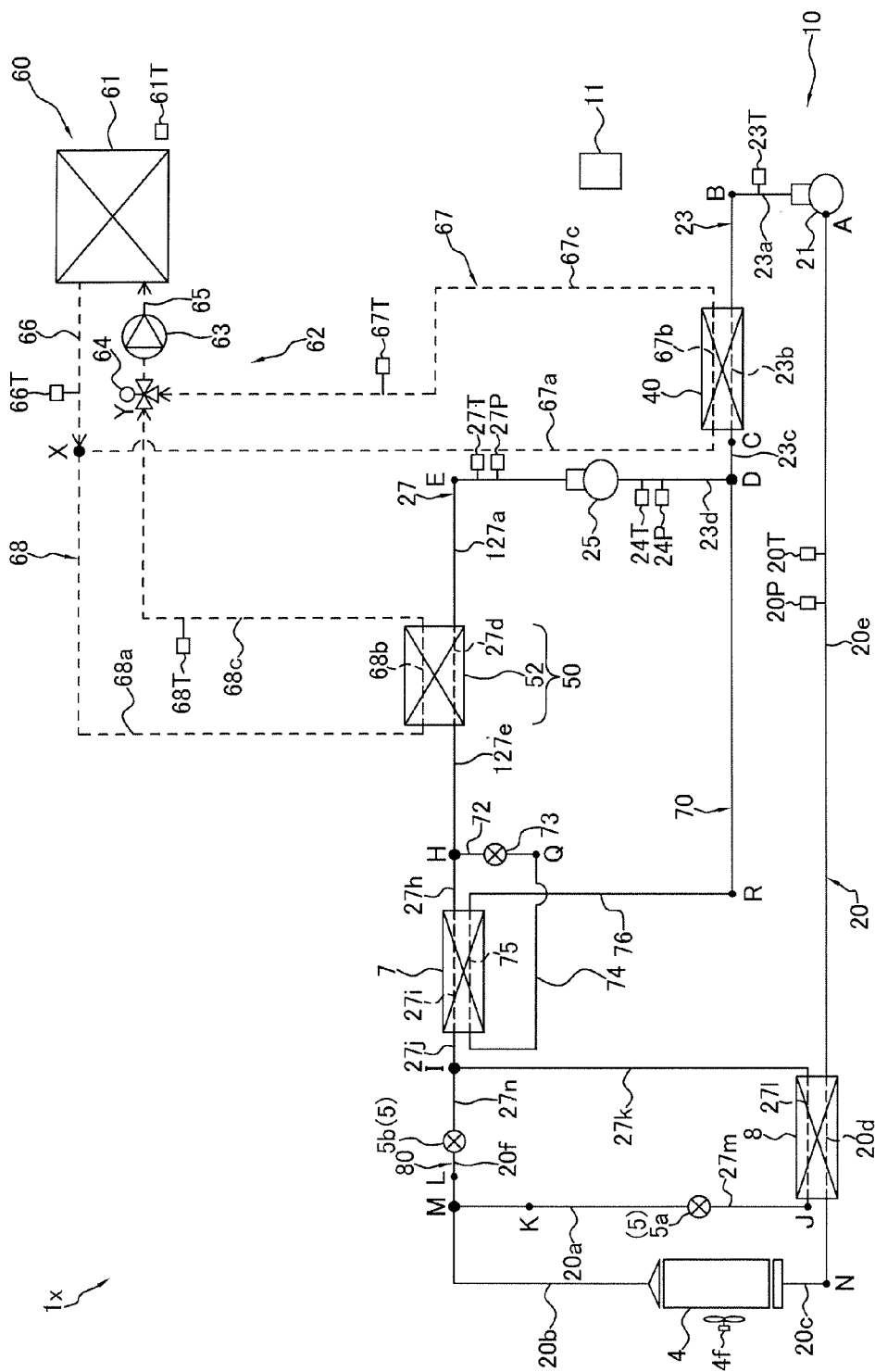


FIG. 16

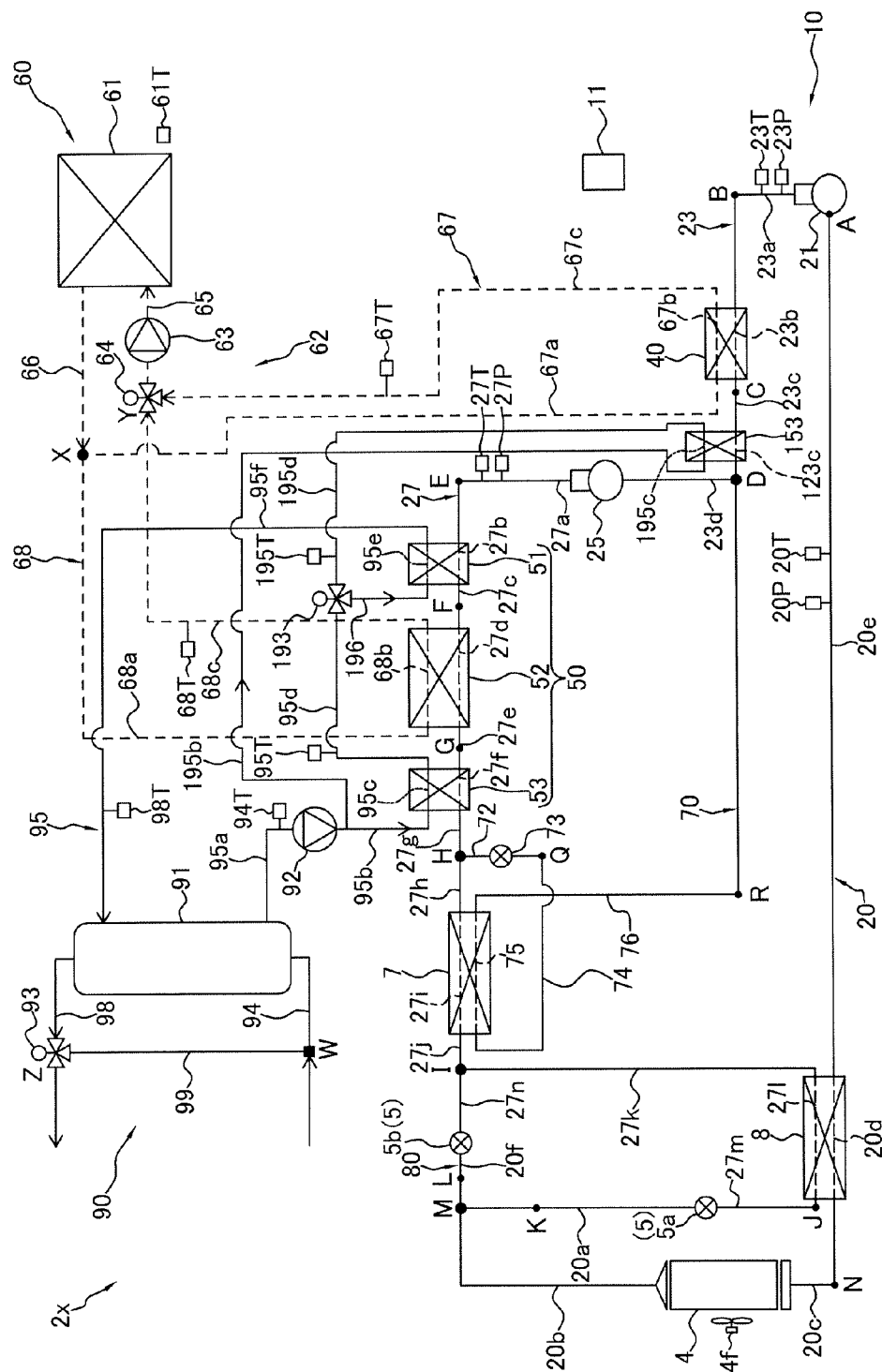


FIG. 17

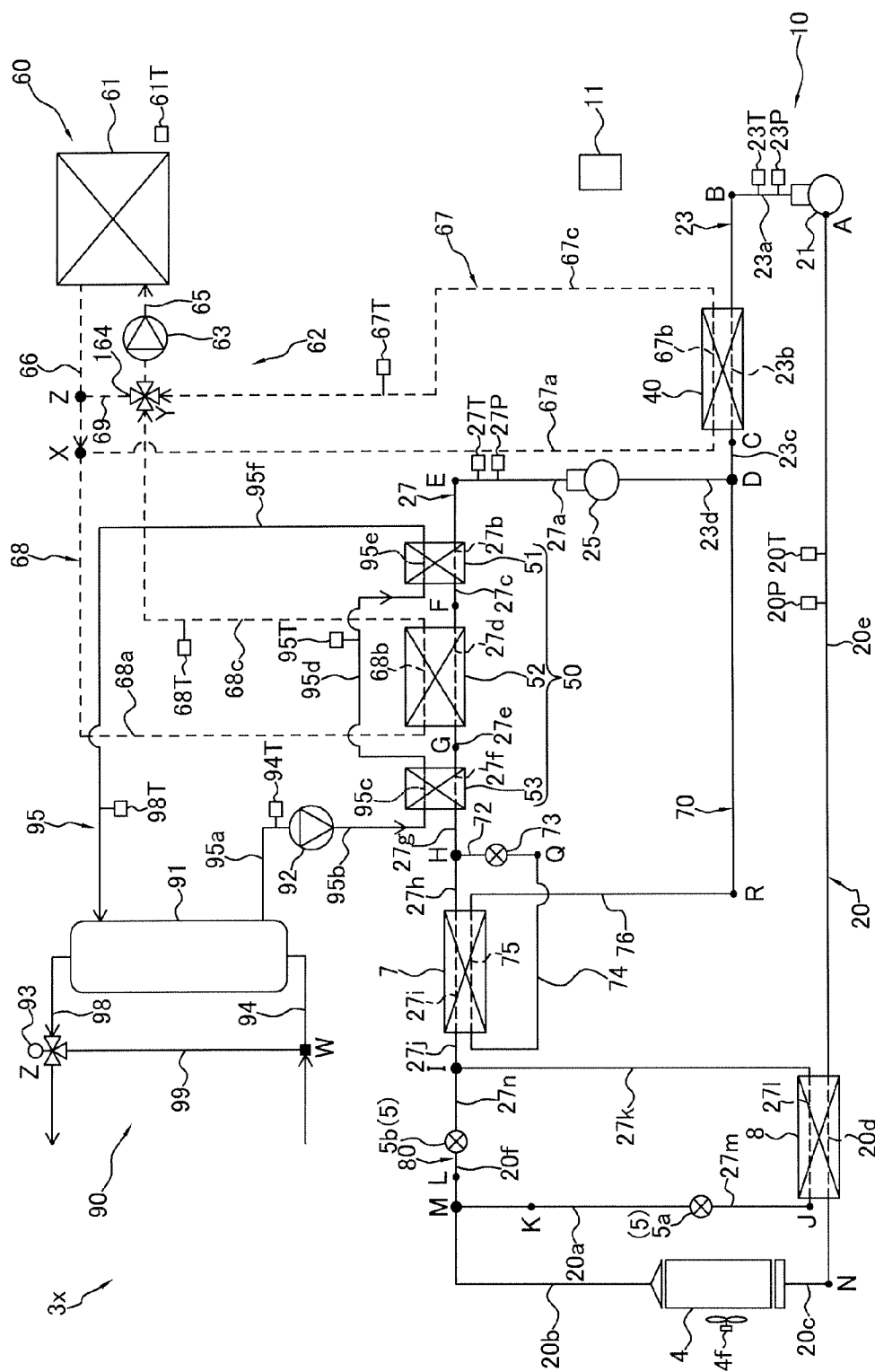


FIG. 18

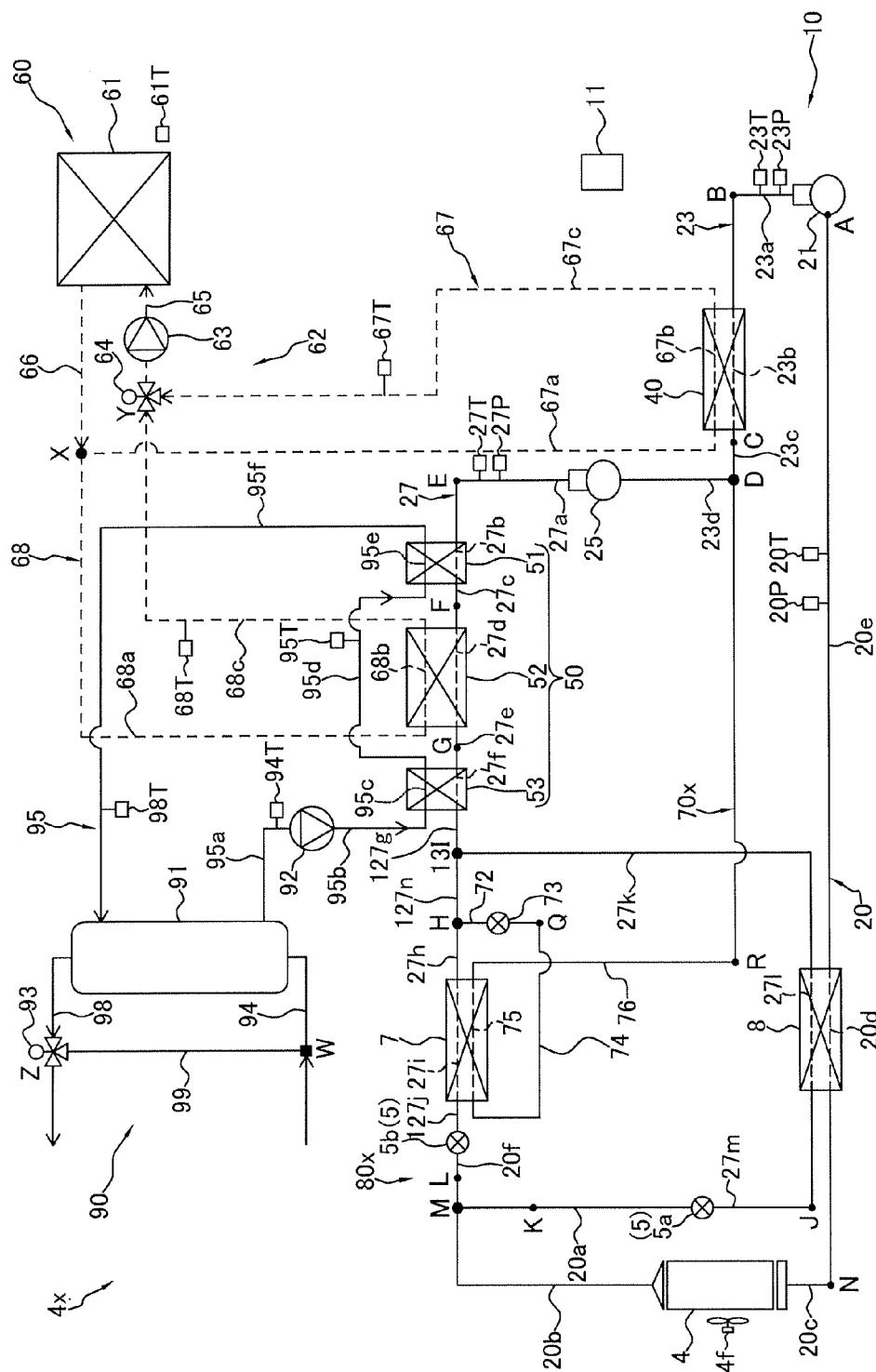


FIG. 19

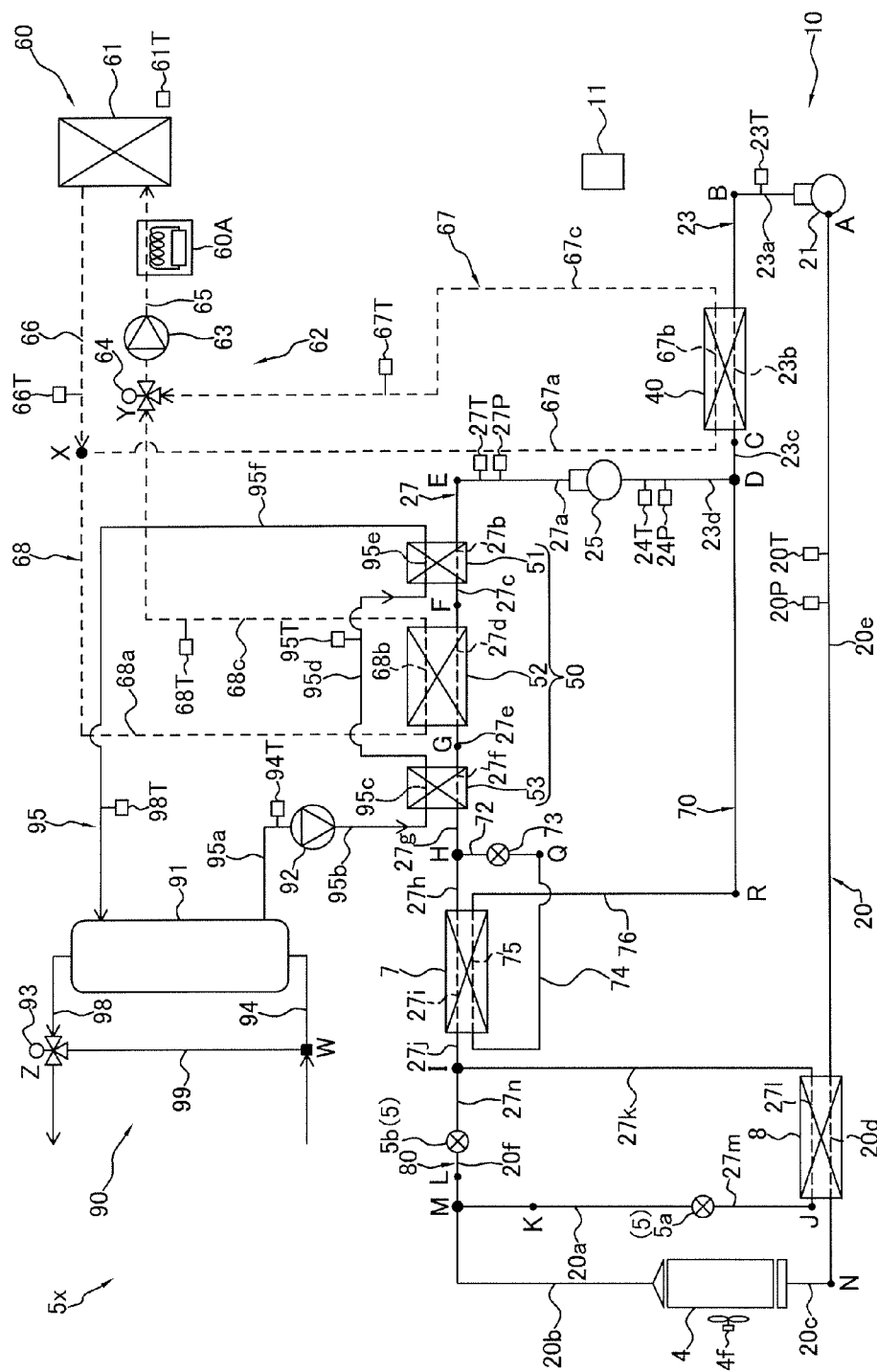


FIG. 20

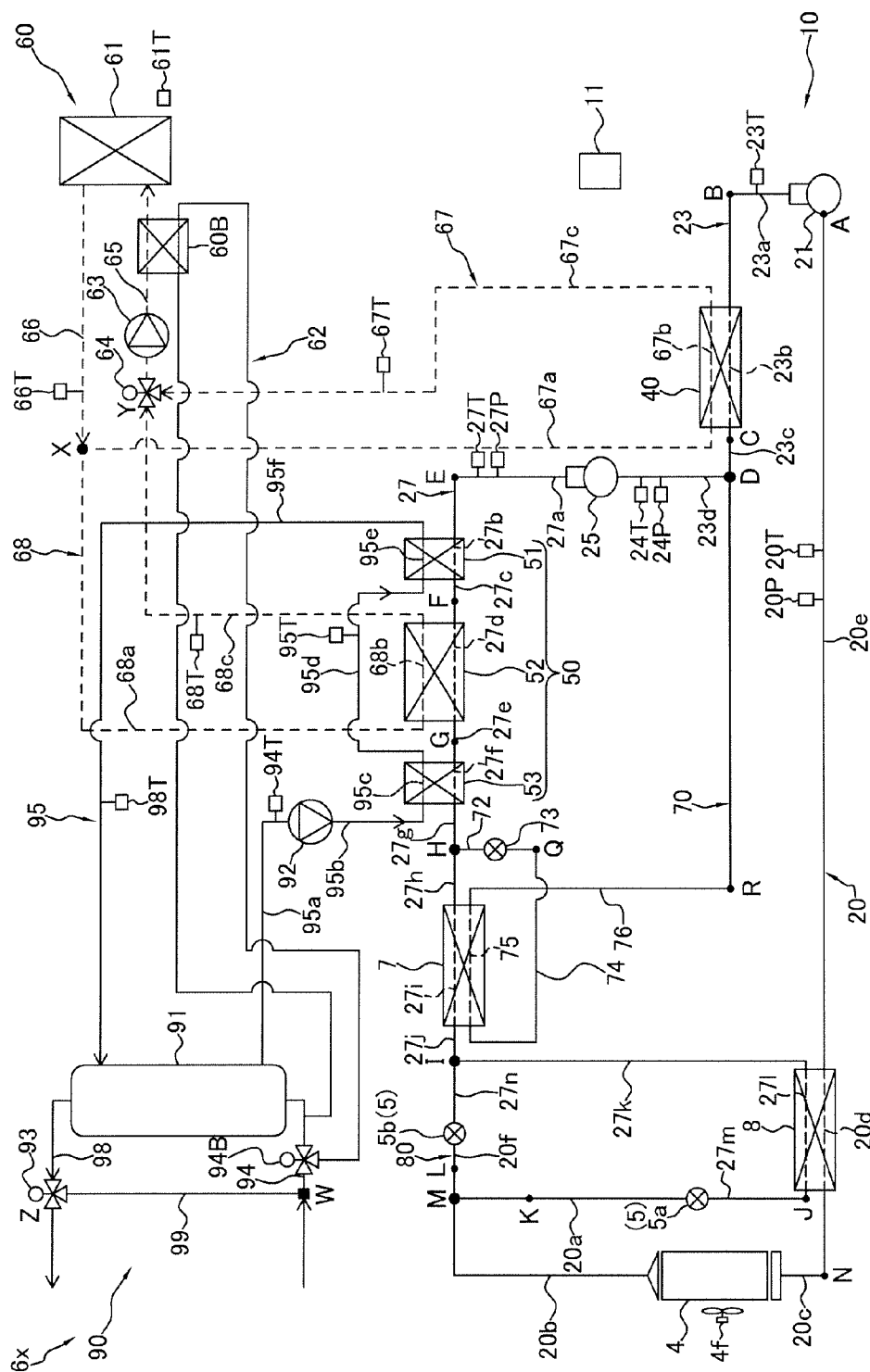


FIG. 21

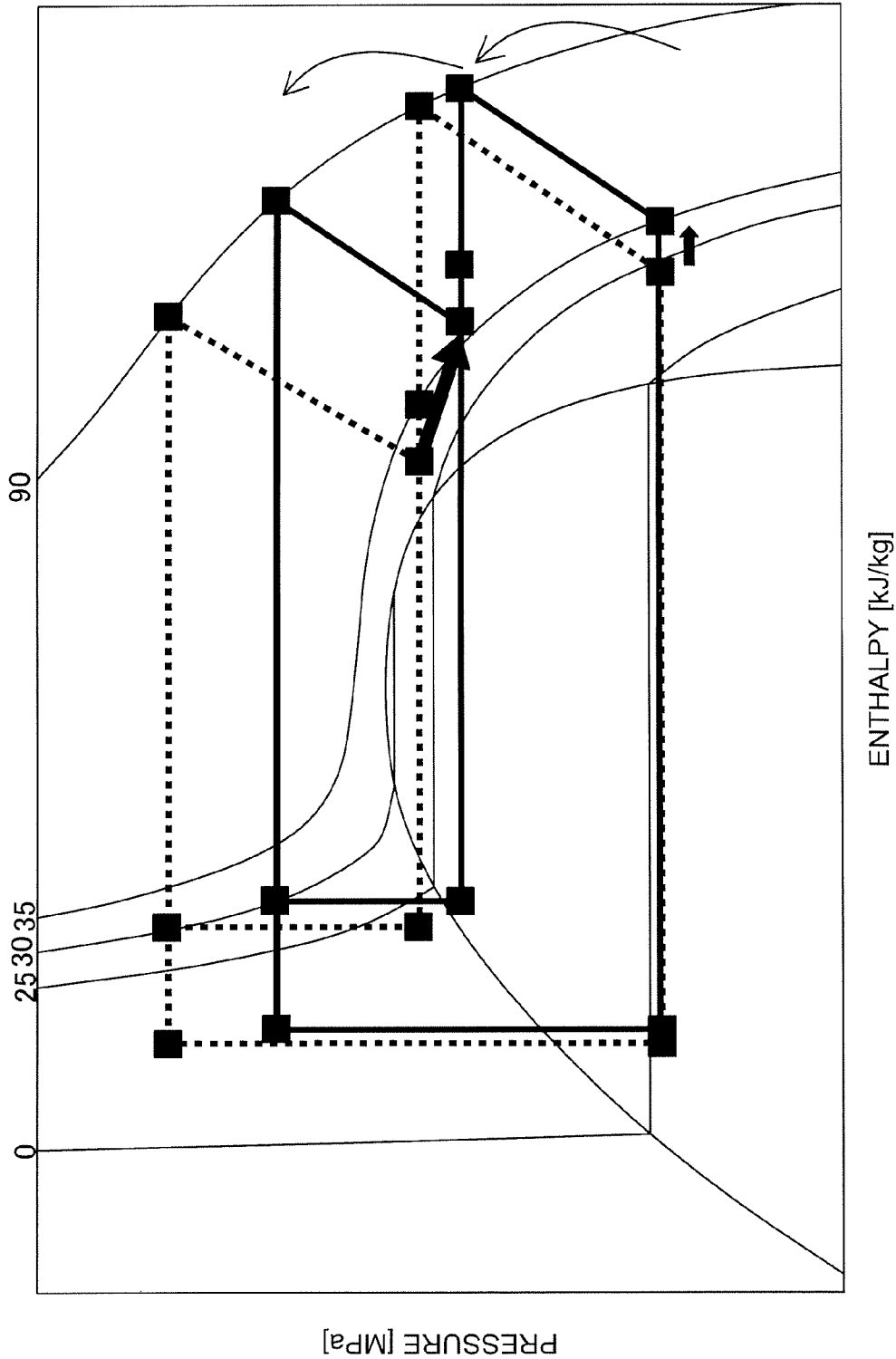


FIG. 22

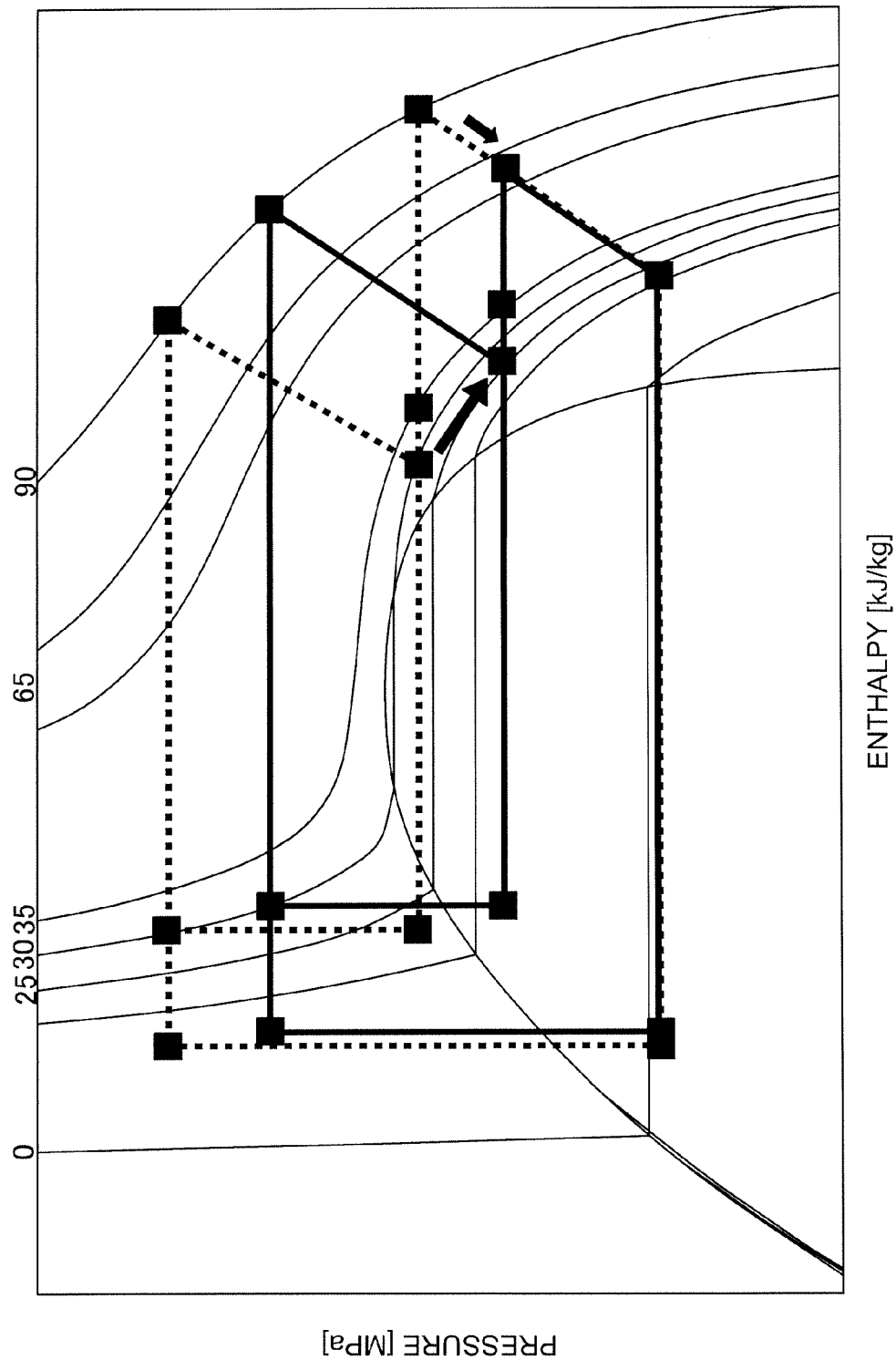


FIG. 23

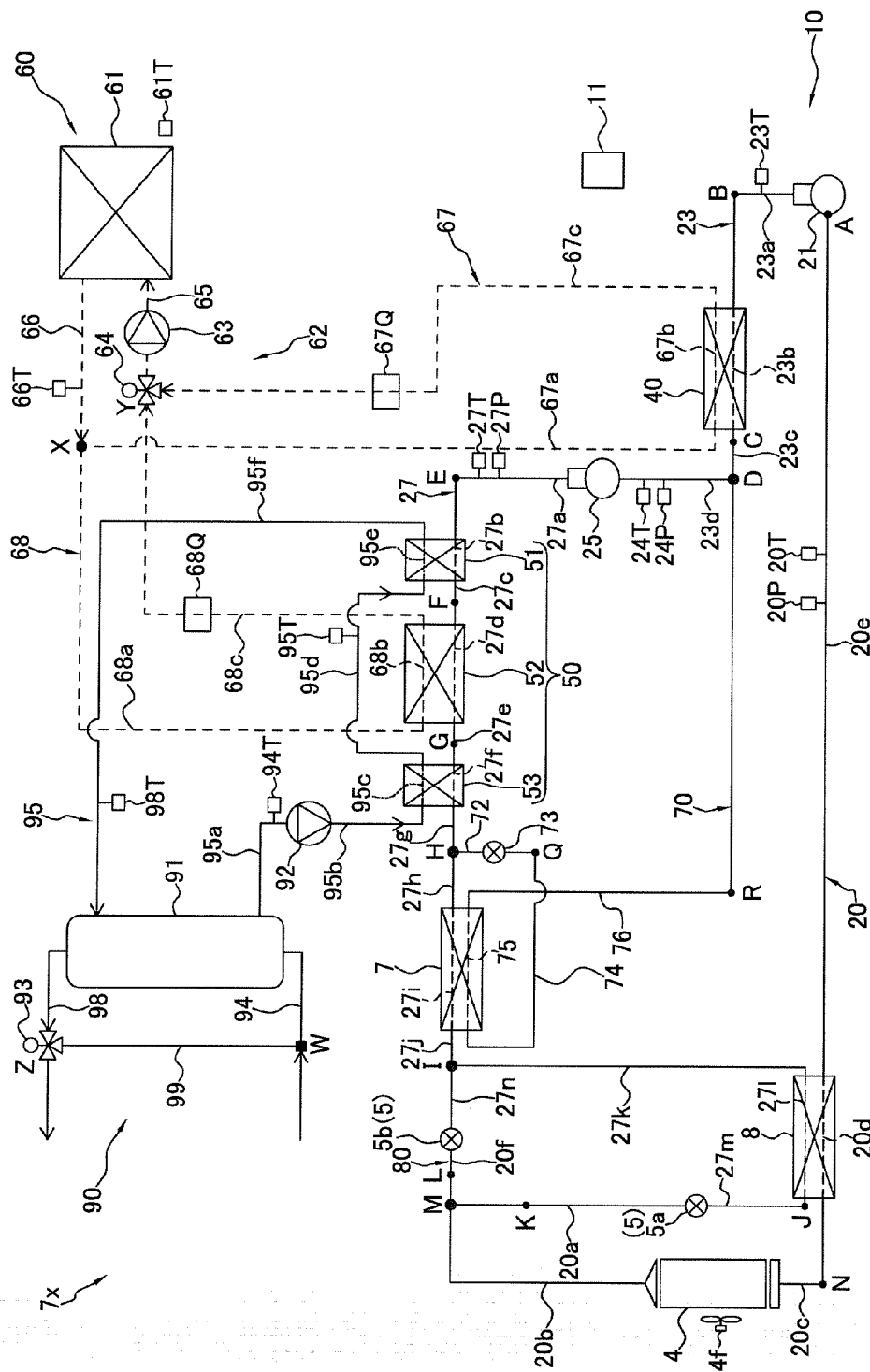


FIG. 24

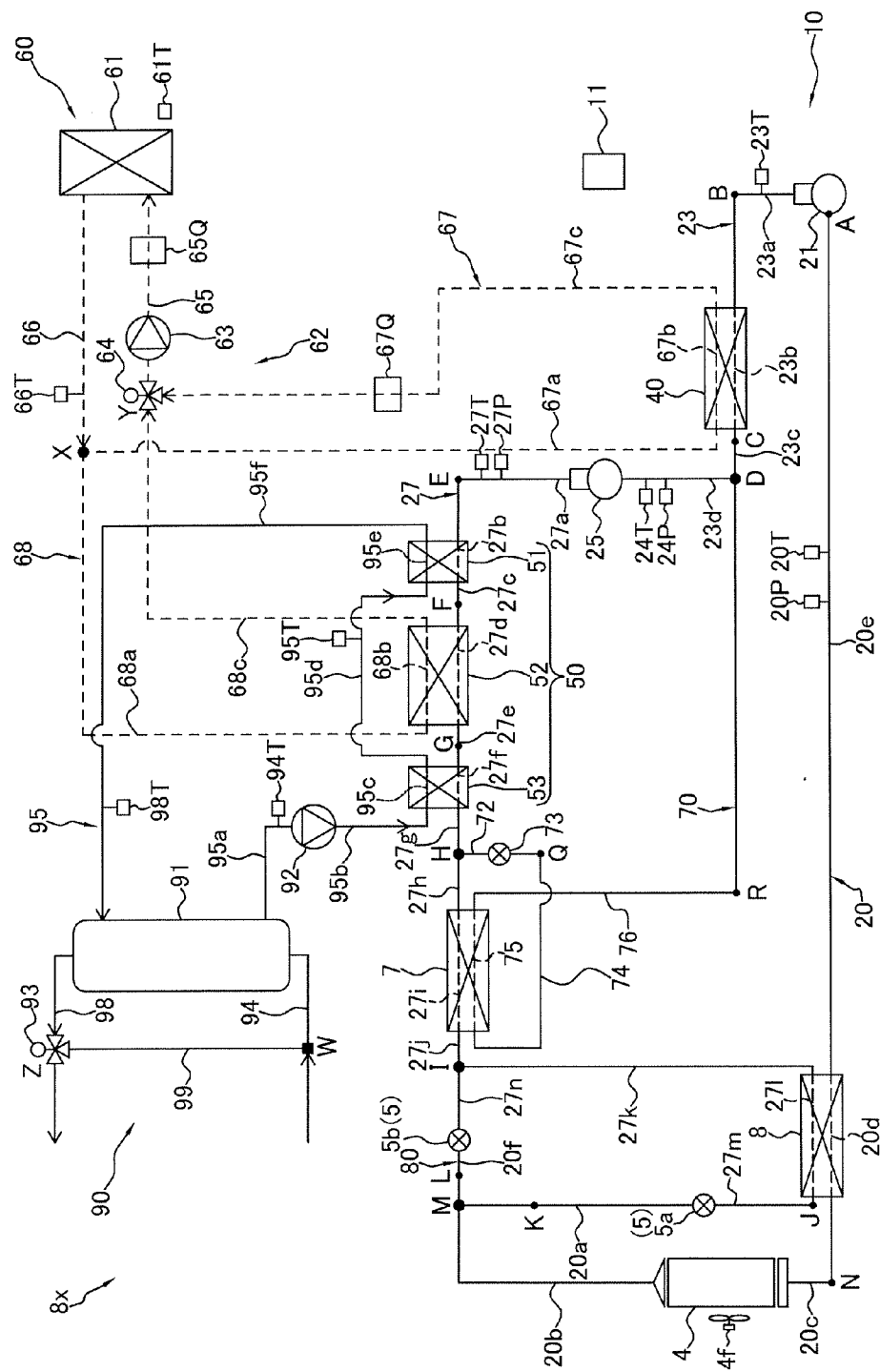


FIG. 25

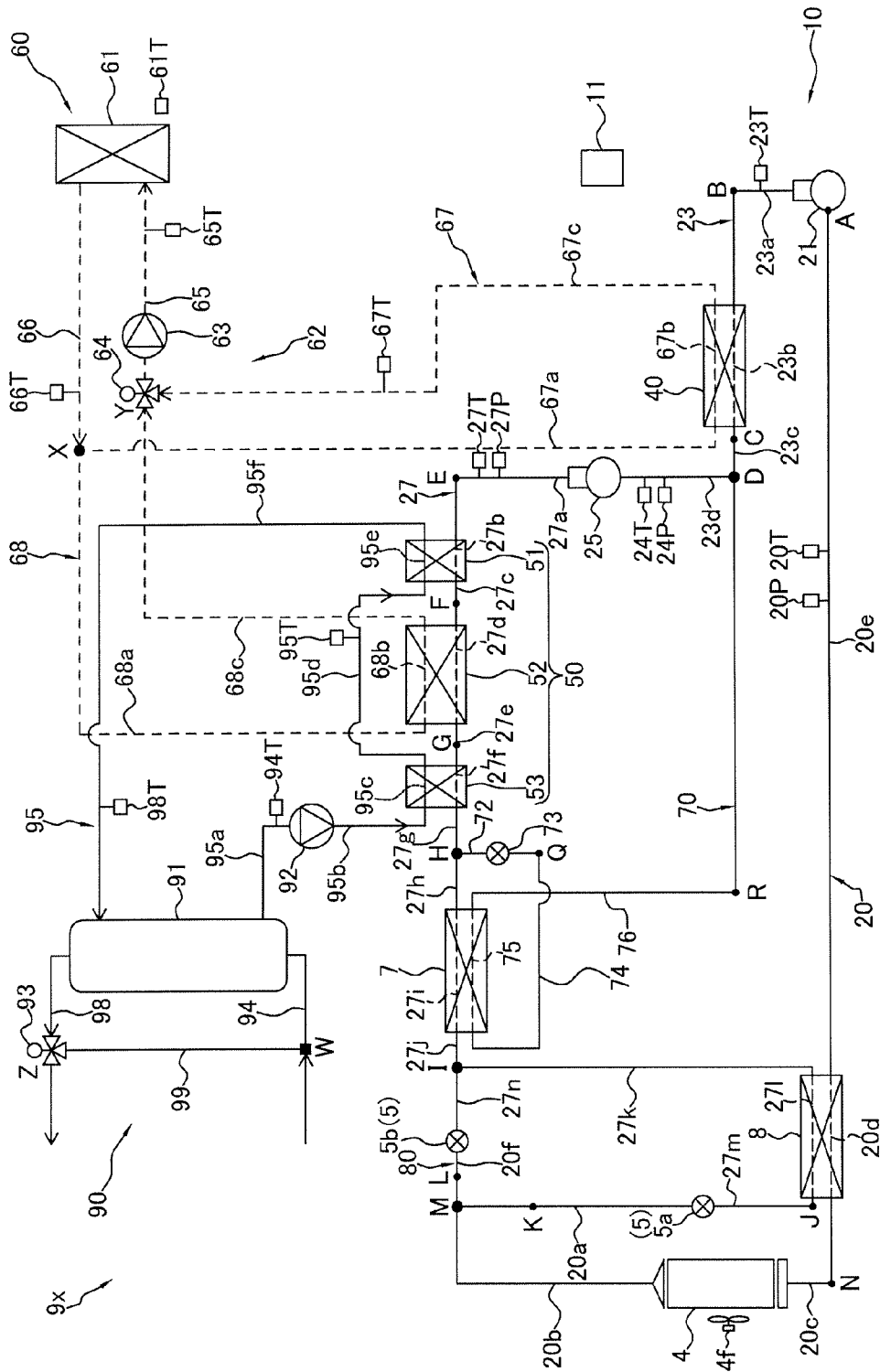


FIG. 26

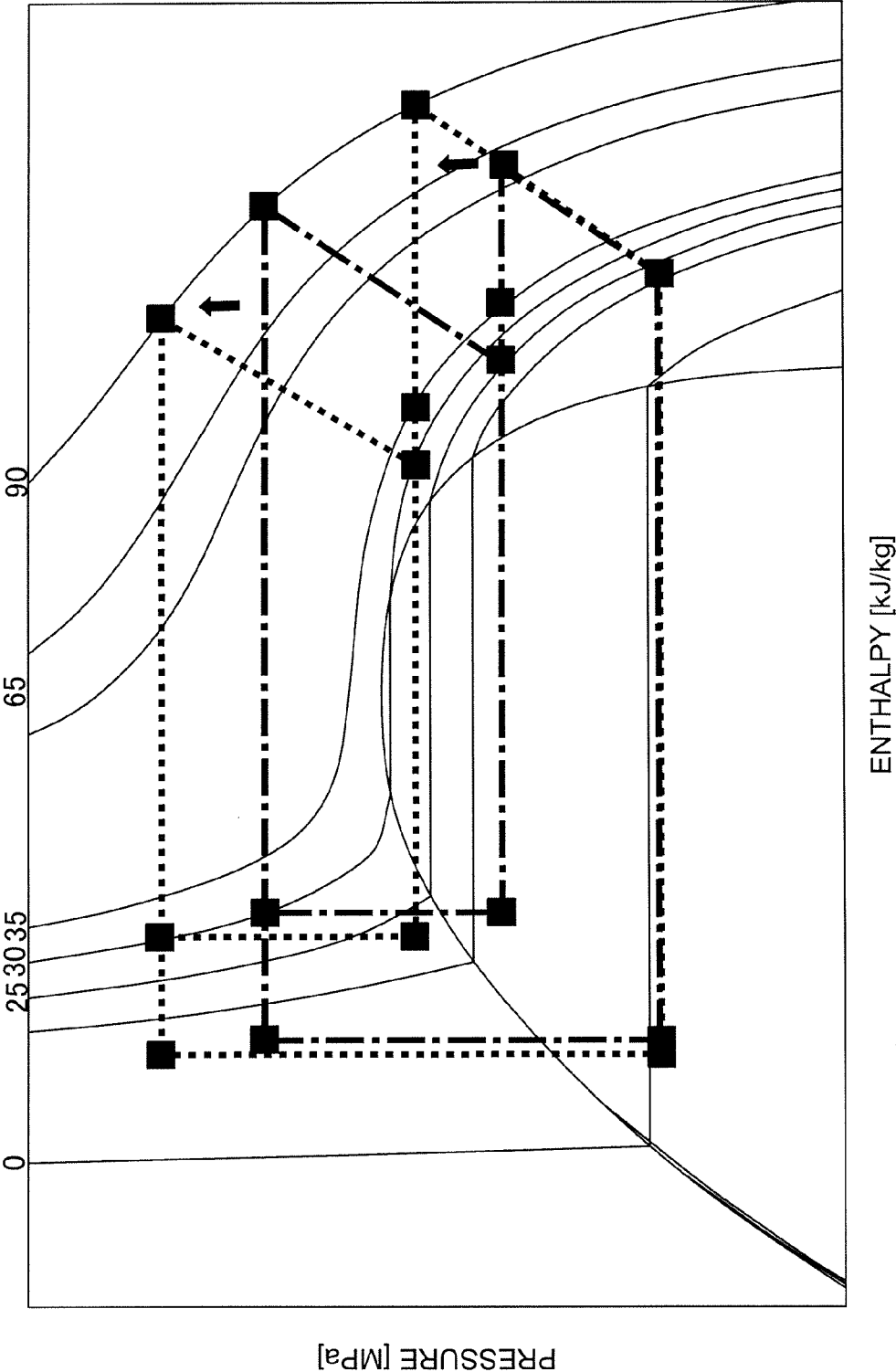


FIG. 27

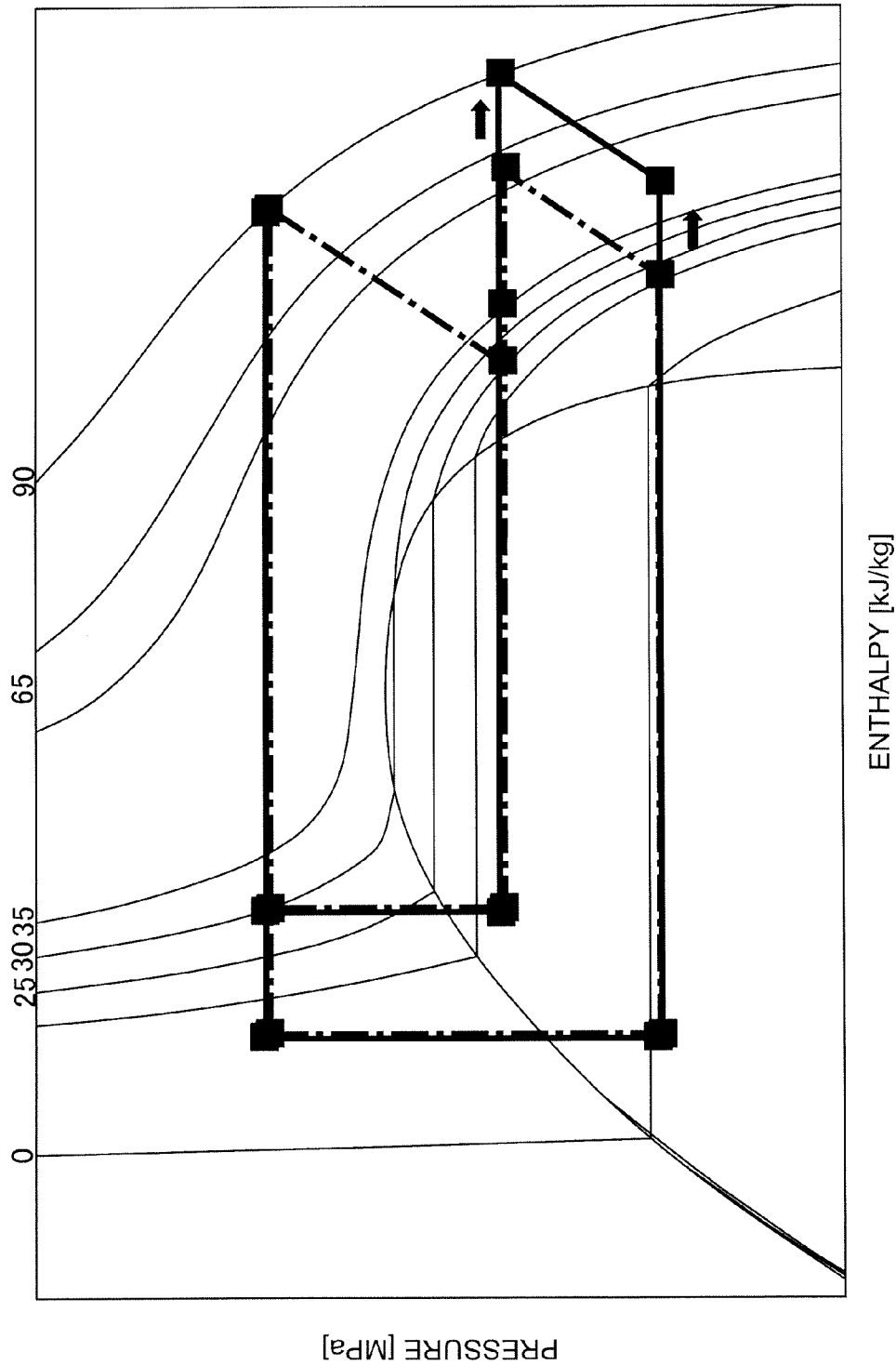


FIG. 28

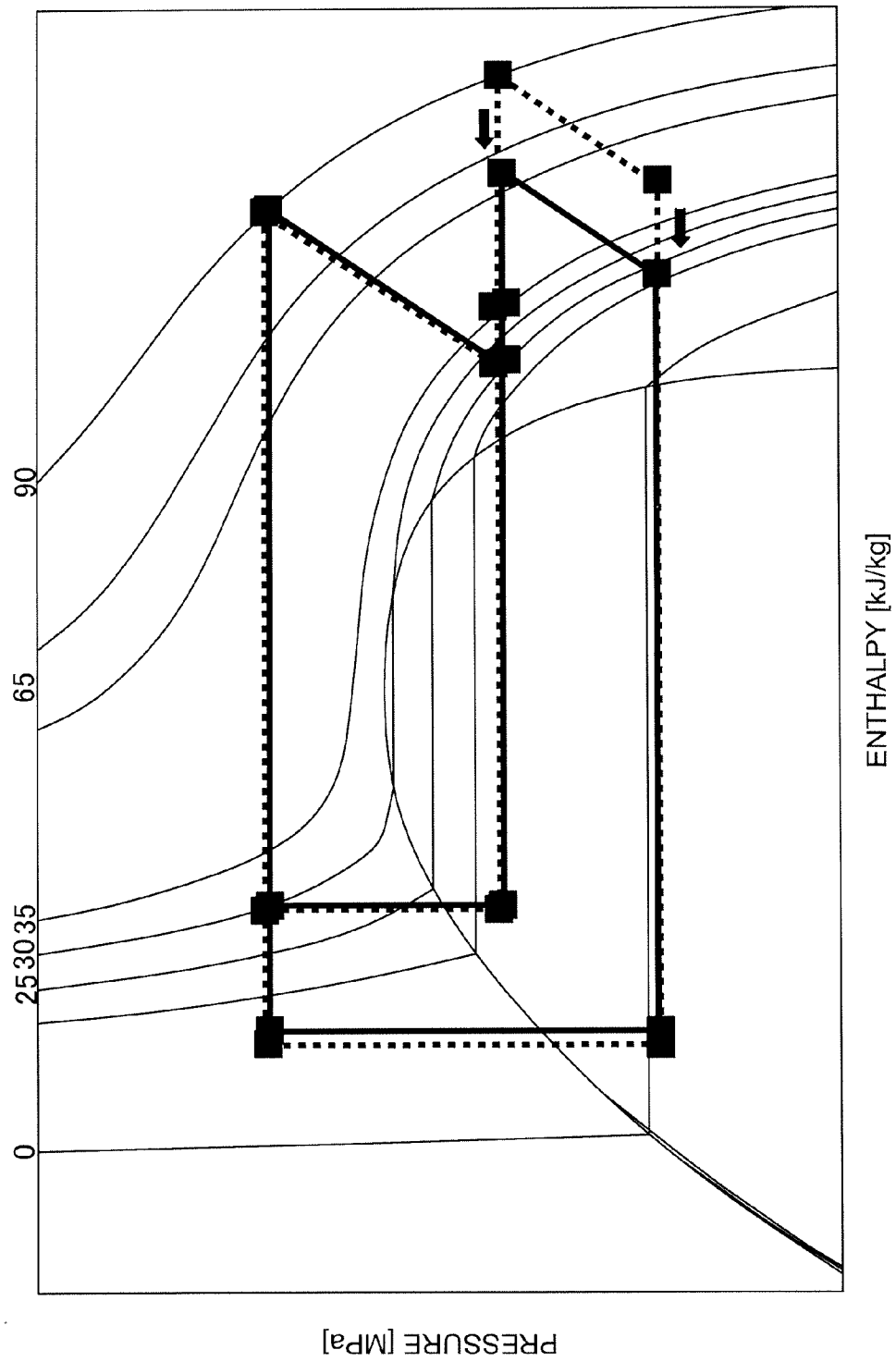


FIG. 29

HEAT PUMP SYSTEM

TECHNICAL FIELD

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

BACKGROUND ART

[0002] There are known conventional systems that perform an air-warming operation using a heat pump cycle through which a primary refrigerant circulates, and a secondary-side cycle through which a secondary refrigerant circulates.

[0003] For example, with the heat pump-type air conditioner described in Patent Document 1 (Japanese Laid-open Patent Application Publication No. 2004-177067), high pressure-side primary refrigerant and low pressure-side primary refrigerant are made to undergo heat exchange, and the heating of the secondary refrigerant for air warming is aided using the heat of the low pressure-side primary refrigerant thus warmed, whereby an improvement in efficiency is ensured.

SUMMARY OF THE INVENTION

Technical Problem

[0004] The heat pump-type air conditioner described in Patent Document 1 (Japanese Laid-open Patent Application Publication No. 2004-177067) described above envisions a single-stage compression type having a single compression mechanism, and the drive force required in the compression mechanism is high.

[0005] An object of the present invention is to provide a heat pump system that can improve cycle efficiency in heat load processing performed by the secondary refrigerant.

Solution to Problem

[0006] A heat pump system of a first aspect of the present invention comprises a heat pump circuit, a first heat load circuit, a first heat exchanger, a second heat exchanger, a first flow rate adjustment mechanism, and a controller. The heat pump circuit has at least a low-stage-side compression mechanism, a high-stage-side compression mechanism, an expansion mechanism, and an evaporator. The heat pump circuit circulates a primary refrigerant. The first heat load circuit has a first branching portion, a second branching portion, a first branching channel, a second branching channel, and a first heat-load-processing section. The first branching channel connects the first branching portion and the second branching portion. The second branching channel connects the first branching portion and the second branching portion without merging with the first branching channel. The first heat load circuit circulates a first fluid. The first heat exchanger performs heat exchange between the primary refrigerant flowing from a discharge side of the low-stage-side compression mechanism toward an intake side of the high-stage-side compression mechanism and the first fluid flowing through the first branching channel. The second heat exchanger performs heat exchange between the primary refrigerant flowing from the high-stage-side compression mechanism toward the expansion mechanism and the first fluid flowing through the second branching channel. The first flow rate adjustment mechanism is capable of adjusting at least one flow rate among the flow rate of the first fluid in the first branching channel and the flow rate of the first fluid in the second branching channel. The controller performs flow rate adjustment control for operating the first flow rate adjustment

mechanism. In the flow rate adjustment control, the first flow rate adjustment mechanism is operated so as to maintain a state in which predetermined temperature conditions are satisfied, including a case in which the ratio between the temperature of the first fluid flowing through a portion of the first branching channel that has passed through the first heat exchanger and the temperature of the first fluid flowing through a portion of the second branching channel that has passed through the second heat exchanger is 1; or to reduce the difference between the temperature of the first fluid flowing through a portion of the first branching channel that has passed through the first heat exchanger and the temperature of the first fluid flowing through a portion of the second branching channel that has passed through the second heat exchanger. A compression mechanism may be further provided apart from the high-stage-side compression mechanism and the low-stage-side compression mechanism, and it is apparent that multistage compression systems are also included in the scope of the present invention.

[0007] According to the heat pump system of the aspect of the present invention described above, the difference between the ambient temperature and the temperature of the first fluid heated in the first heat exchanger, and the difference between the ambient temperature and the temperature of the first fluid heated in the second heat exchanger can be prevented from increasing in the case that the heat amount of the secondary refrigerant fed to the first heat-load-processing section is the same. It is therefore possible to minimize the total of the radiation loss by release from the first fluid, which has been heated in the first heat exchanger, prior to arriving at the first heat-load-processing section and the radiation loss by release from the first fluid, which has been heated in the second heat exchanger, prior to arriving at the first heat-load-processing section. It is thereby possible to improve the efficiency of the heat pump system in terms of processing the heat load in the first heat load heat exchanger.

[0008] The heat pump system of the second aspect of the present invention is the heat pump system of the first aspect, wherein the controller controls output of the low-stage-side compression mechanism and the high-stage-side compression mechanism so that the temperature of the primary refrigerant that flows into the first heat exchanger and the temperature of the primary refrigerant that flows into the second heat exchanger both become a temperature equal to or greater than a first heat-load-corresponding temperature requested in the first heat-load-processing section, while causing the temperature of the primary refrigerant flowing to the first heat exchanger to become a temperature equal to or greater than the first fluid flowing to the first heat exchanger, and while causing the temperature of the primary refrigerant flowing to the second heat exchanger to become a temperature equal to or greater than the first fluid flowing to the second heat exchanger.

[0009] According to the heat pump system of the aspect described above, it is possible to reliably increase the temperature of the first fluid by using the primary refrigerant that flows into the first heat exchanger, without a reduction in the temperature of the first fluid that flows into the first heat exchanger. The discharge refrigerant temperature of the high-stage-side compression mechanism can be prevented from increasing abnormally. Similarly, it is possible to reliably increase the temperature of the first fluid by using the primary refrigerant that flows into the second heat exchanger, without a reduction in the temperature of the first fluid that flows into

the second heat exchanger. It is possible to adapt to the heat load in the first heat load heat exchanger by using only the heat amount obtained by the first fluid in the first heat exchanger and the second heat exchanger.

[0010] The heat pump system of the third aspect of the present invention is the heat pump system of the second aspect, wherein the first heat load circuit further comprises a first heat load bypass circuit for connecting the portion between the first heat-load-processing section and the first branching portion, and the portion between the first heat-load-processing section and the second branching portion; and a first heat-load-bypass flow-rate-adjustment mechanism capable of adjusting the flow rate of the first fluid that passes through the first heat load bypass circuit. The controller performs a control in the flow rate adjustment control so that a target value of the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger and a target value of the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger become a temperature that exceeds the first heat-load-corresponding temperature. The controller operates the first heat-load-bypass flow-rate-adjustment mechanism and adjusts the flow rate of the first fluid flowing through the first heat load bypass circuit so that the temperature of the first fluid fed to the first heat-load-processing section becomes the first heat-load-corresponding temperature.

[0011] According to the heat pump system of the aspect described above, the temperature of the first fluid fed to the first heat-load-processing section can be adjusted by the first heat load bypass flow rate adjustment mechanism by adjusting the flow rate of the first fluid that passes through the first heat load bypass circuit, even in an operating condition in which the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger as well as the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger have become a temperature that exceeds the first heat load-corresponding temperature required in the first heat-load-processing section. It is thereby possible to bring the temperature of the first fluid fed to the first heat-load-corresponding section close to the first heat load-processing temperature in order to increase the efficiency of the heat pump circuit, even when the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger have exceeded the first heat load-corresponding temperature.

[0012] The heat pump system of the fourth aspect of the present invention is the heat pump system of the second aspect, wherein the controller performs a control in the flow rate adjustment control so that a target value of the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger and a target value of the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger become the first heat-load-corresponding temperature.

[0013] According to the heat pump system of the aspect described above, the temperature of the first fluid flowing through the portion of the first branching channel that has

passed through the first heat exchanger and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger are brought close to the first heat-load-corresponding temperature required in the first heat-load-processing section. It is thereby possible to avoid a state in which the temperature of the first fluid flowing through the first heat load circuit has considerably exceeded the first heat-load-corresponding temperature, and to effectively reduce radiation loss.

[0014] In the case that the first flow rate adjustment mechanism is controlled so as to obtain the first heat-load-corresponding temperature, it is possible to eliminate the need to provide to the first heat load circuit a function for adjusting the temperature of the first fluid moving toward the first heat-load-processing section.

[0015] The heat pump system of the fifth aspect of the present invention is the heat pump system of any of the second to fourth aspects, wherein the controller controls at least one of the low-stage-side compression mechanism, the high-stage-side compression mechanism, and the expansion mechanism in the flow rate adjustment control so as to: maintain a state in which predetermined compression ratio conditions are satisfied, including a case in which the ratio between the compression ratio in the low-stage-side compression mechanism and the compression ratio in the high-stage-side compression mechanism is 1, or reduce the difference between the compression ratio in the low-stage-side compression mechanism and the compression ratio in the high-stage-side compression mechanism.

[0016] According to the heat pump system of the aspect described above, the compressor drive force required in the high-stage-side compression mechanism and the low-stage-side compression mechanism can be minimized in the case that flow rate adjustment control is performed so that the temperature of the primary refrigerant flowing to the first heat exchanger and the temperature of the primary refrigerant flowing to the second heat exchanger both become a temperature equal to or greater than the first heat-load-corresponding temperature, while ensuring that the temperature of the primary refrigerant flowing to the first heat exchanger becomes a temperature equal to or greater than the temperature of the first fluid flowing to the first heat exchanger, and while ensuring that the temperature of the primary refrigerant flowing to the second heat exchanger becomes a temperature equal to or greater than the temperature of the first fluid flowing to the second heat exchanger. Not only is it thereby possible to reduce radiation loss by the first fluid, but it is also possible to handle the heat load in the first heat-load-processing section using a low drive force and to further improve efficiency.

[0017] The heat pump system of the sixth aspect of the present invention is the heat pump system of the fifth aspect, wherein the controller performs low-stage intake degree-of-superheat control for increasing the degree of superheat of the primary refrigerant taken in by the low-stage-side compression mechanism in the case that the discharge temperature of the primary refrigerant of the low-stage-side compression mechanism increases when the flow rate adjustment control is performed.

[0018] Generally, the compression ratio of the low-stage-side compression mechanism tends to increase in the case that the target value of the discharge temperature of the primary refrigerant in the low-stage-side compression mechanism is high. The compression ratio of the high-stage-side compression

sion mechanism also increases as a result. Therefore, the drive force required by the compression mechanism increases and energy consumption increases.

[0019] In contrast, with this heat pump system, low-stage intake degree-of-superheat control for increasing the target value of the degree of superheat of the primary refrigerant taken in by the low-stage-side compression mechanism is performed in the case that the target value of the discharge temperature of the primary refrigerant in the low-stage-side compression mechanism is to be increased. It is therefore possible to minimize the compression ratio of the low-stage-side compression mechanism required for discharge temperature of the primary refrigerant in the low-stage-side compression mechanism to reach the target value. In addition, the compression ratio of the high-stage-side compression mechanism can also be minimized. The required drive force of the compression mechanism can thereby be further minimized. In the converse case that the target value of the discharge temperature of the primary refrigerant in the low-stage-side compression mechanism becomes low, the degree of superheat of the primary refrigerant taken in by the low-stage-side compression mechanism is reduced to thereby enable to reduce the specific volume of the primary refrigerant taken in by the low-stage-side compression mechanism, while an increase in the compression ratio of the high-stage-side compression mechanism is also suppressed by reducing the increase in the compression ratio of the low-stage-side compression mechanism. A circulation amount can thereby be ensured and capacity can be increased while suppressing an increase in the compression ratio.

[0020] The heat pump system of the seventh aspect of the present invention is the heat pump system of the sixth aspect, wherein the heat pump circuit furthermore has a primary-refrigerant-to-primary-refrigerant heat exchanger for causing heat exchange to be performed between the primary refrigerant taken in by the low-stage-side compression mechanism and the primary refrigerant that has passed through the second heat exchanger and then flows toward the expansion mechanism. The controller performs low-stage intake degree-of-superheat control using the primary-refrigerant-to-primary-refrigerant heat exchanger.

[0021] According to the heat pump system of the aspect of the present invention described above, heat for cooling the primary refrigerant prior to being taken into the expansion mechanism can be recovered as heat for increasing the degree of superheat of the primary refrigerant taken in by the low-stage-side compression mechanism. It is thereby possible not only to increase the degree of superheat of the primary refrigerant taken in by the low-stage-side compression mechanism, but it is also possible suppress a reduction in the through-rate of the primary refrigerant in the expansion mechanism and to improve capacity.

[0022] The heat pump system of the eighth aspect of the present invention is the heat pump system of any of the fifth to seventh aspects, wherein the controller performs a control during load reduction for reducing the degree of superheat of the primary refrigerant taken in by the low-stage-side compression mechanism while reducing the target value of the discharge temperature of the primary refrigerant of the low-stage-side compression mechanism in the case that the temperature of the first fluid flowing from the first heat-load-processing section toward the first heat exchanger and the second heat exchanger has increased when flow rate adjustment control is performed.

[0023] According to the heat pump system of the aspect described above, the heat load in the first heat-load-processing section is low in the case that temperature of the first fluid flowing from the first heat load circuit toward the first heat exchanger and the second heat exchanger has increased, and it is therefore possible to handle the load even in the case that a change has been made to the efficient operating state described above. The density of the primary refrigerant taken in by the low-stage-side compression mechanism can also be increased, and the circulation amount of the primary refrigerant can be increased. It is thereby possible to increase the capacity of the heat pump circuit while adapting to load fluctuations.

[0024] The heat pump system of the ninth aspect of the present invention is the heat pump system of the eighth aspect, further comprising a second heat load circuit through which a second fluid circulates, the second heat load circuit having a second heat load section; and a third heat exchanger for causing heat exchange to be performed between the second fluid circulating through the second heat load circuit and the primary refrigerant flowing from the high-stage-side compression mechanism toward the second heat exchanger.

[0025] According to the heat pump system of the aspect described above, not only can the heat of the primary refrigerant discharged by the high-stage-side compression mechanism be used for both heat load processing in the first heat load circuit and heat load processing in the second heat load circuit, but a temperature range beyond what is required in the first heat load circuit can be used in the second heat load circuit.

[0026] The heat pump system of the tenth aspect of the present invention is the heat pump system of the ninth aspect, further comprising a fourth heat exchanger for causing heat exchange to be performed between the second fluid flowing from the second heat-load-processing section toward the third heat exchanger, among the second fluid that passes through the second heat load circuit, and the primary refrigerant which has passed through the second heat exchanger and is thereafter flowing toward the expansion mechanism.

[0027] According to the heat pump system of the aspect described above, in the case that the temperature variation range of the first fluid in the first heat-load-processing section is included in the temperature variation range of the second fluid in the second heat-load-processing section, heat exchange with the primary refrigerant in a low-temperature state and heat exchange with the primary refrigerant in a high-temperature state among the primary refrigerant discharged by the high-stage-side compression mechanism can be used for heat exchange with the second fluid, and the primary refrigerant in an intermediate-temperature state can be used for heat exchange with the first fluid. It is thereby possible to improve heat exchange efficiency because heat exchange can be performed in the second heat exchanger, the third heat exchanger, and the fourth heat exchanger while the temperature difference between the primary refrigerant and the first and second fluids is kept minimized.

[0028] The heat pump system of the eleventh aspect of the present invention is the heat pump system of the ninth and tenth aspects, wherein the controller adjusts the circulation amount of the second fluid circulating through the second heat load circuit so that the temperature of the primary refrigerant that passes through the third heat exchanger approximates a target value of the temperature of the primary refrigerant discharged by the low-stage-side compression

mechanism in the case that the target value of the temperature of the primary refrigerant discharged by the low-stage-side compression mechanism is less than the target value of the temperature of the primary refrigerant discharged by the high-stage-side compression mechanism.

[0029] According to the heat pump system of the aspect described above, the maximum temperature of the primary refrigerant flowing through the first heat exchanger and the maximum temperature of the primary refrigerant flowing through the second heat exchanger are brought close together, whereby the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger and the temperature of the first fluid flowing through portion of the second branching channel that has passed through the second heat exchanger are more readily brought close together.

[0030] For example, if it is desired to keep the flow rate of the first fluid fed to the first heat-load-processing section low, the temperature of the primary refrigerant flowing through the first heat exchanger and the temperature of the primary refrigerant flowing through second heat exchanger come close to each other even if the time for the first fluid to pass through the first heat exchanger and/or pass through the second heat exchanger is increased. Accordingly, the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger can be made to converge at a value near the temperature of the primary refrigerant flowing through the first heat exchanger (the temperature of the primary refrigerant flowing through the second heat exchanger).

[0031] The heat pump system of the twelfth aspect of the present invention is the heat pump system of any of the ninth to eleventh aspects, wherein the second heat load processing unit is a hot-water-supply tank. The second fluid is water for hot-water supply.

[0032] According to the heat pump system of the aspect described above, hot water can be made using the temperature of the primary refrigerant discharged from the high-stage-side compression mechanism.

[0033] The heat pump system of the thirteenth aspect of the present invention is the heat pump system of any of the second to twelfth aspects, wherein the controller operates the first flow rate adjustment mechanism in the flow rate adjustment control to thereby reduce the flow rate of the first fluid having a lower temperature among: the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger; and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger.

[0034] According to the heat pump system of the aspect described above, the flow rate of the first fluid having a lower temperature among the temperature of the first fluid flowing through the section of the first branching channel that has passed through the first heat exchanger and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger is reduced, whereby flow speed of the first fluid having a lower temperature can be reduced and the heating time can be increased. It is possible to increase the amount of heat recovered from the primary refrigerant in the heat

exchanger having the reduced flow rate among the first heat exchanger and the second heat exchanger.

[0035] For example, the amount of recovered heat can be increased by reducing the through speed to extend the time available for heat exchange in the case that the first fluid has passed through the first heat exchanger or the second heat exchanger at a high flow speed without being heated to the inlet temperature of the primary refrigerant.

[0036] The heat pump system of the fourteenth aspect of the present invention is the heat pump system of the thirteenth aspect, wherein the first flow rate adjustment mechanism is capable of adjusting the ratio between the flow rate of the first fluid flowing through the first branching channel and the flow rate of the first fluid flowing through the second branching channel. The controller operates the first flow rate adjustment mechanism in the flow rate adjustment control to thereby reduce the flow rate ratio of the first fluid having a lower temperature among: the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger; and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger, while keeping constant the flow rate of the first fluid fed to the first heat-load-processing section.

[0037] According to the heat pump system of the aspect described above, using the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger, the flow speed of the first fluid having the higher temperature is increased and the heating time is reduced, and the flow speed of the first fluid having the lower temperature is reduced and the heating time is extended, by adjusting the flow rate ratio. The temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger can be varied so as to reduce temperature difference. In the case that the heat load has not changed in the first heat-load-processing section, not only can the temperature difference be reduced, but it is also possible to adapt to the heat load of the first heat-load-processing section by keeping the flow rate of the first fluid fed to the first heat-load-processing section.

[0038] The heat pump system of the fifteenth aspect of the present invention is the heat pump system of the thirteenth aspect, wherein the first flow rate adjustment mechanism is capable of adjusting the flow rate of the first fluid fed to the first heat-load-processing section. In the flow rate adjustment control, the controller reduces the flow rate of the first fluid fed to the first heat-load-processing section by operating the first flow rate adjustment mechanism in the case that the flow rate ratio is low for the first fluid having a lower temperature among: the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger; and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger.

[0039] According to the heat pump system of the aspect described above, when the flow rate of the first fluid fed to the first heat-load-processing section is reduced, the temperature increase of the first fluid having a lower temperature is made

greater than the temperature increase of the first fluid having a higher temperature, in the case that the flow rate ratio is low for the first fluid having a lower temperature among: the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger; and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger. It is thereby possible to vary the flow rate so that the temperature difference is reduced. Also, in the case that the heat load is reduced in the first heat-load-processing section, not only can the temperature difference be reduced, but it is also possible to adapt to the heat load in the first heat-load-processing section.

[0040] The heat pump system of the sixteenth aspect of the present invention is the heat pump system of the thirteenth aspect, wherein the first flow rate adjustment mechanism includes a ratio adjustment section for adjusting the ratio between the flow rate of the first fluid flowing through the first branching channel and the flow rate of the first fluid flowing through the second branching channel, and a flow rate adjustment section for adjusting the flow rate of the first fluid fed to the first heat-load-processing section. The controller operates the first flow rate adjustment mechanism in the flow rate adjustment control to thereby increase the flow rate of the first fluid having a temperature that exceeds the first heat-load-corresponding temperature and/or reduce the flow rate of the first fluid having a temperature that is less than the first heat-load-corresponding temperature, as determined from among the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger; and the controller reduces the flow rate of the first fluid fed to the first heat-load-processing section in proportion to the increase of the temperature of the first fluid fed to the first heat-load-processing section in the case that the temperature of the first fluid fed to the first heat-load-processing section has exceeded the first heat-load-corresponding temperature.

[0041] According to the heat pump system of the aspect described above, the flow rate of the first fluid flowing through the first heat load circuit can be set to a rate adapted to the heat load in the first heat-load-processing section while the difference between the temperature of the first fluid flowing through section of the first branching channel that has passed through the first heat exchanger and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger is reduced.

[0042] The heat pump system of the seventeenth aspect of the present invention is the heat pump system of any of the first to sixteenth aspects, further comprising: first branching channel temperature detector for ascertaining the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger; and second branching channel temperature detector for ascertaining the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger.

[0043] According to the heat pump system of the aspect described above, the precision of flow rate adjustment control can be improved because it is possible to directly ascertain the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first

heat exchanger, and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger.

[0044] The heat pump system of the eighteenth aspect of the present invention is the heat pump system of any of the first to sixteenth aspects, further comprising: branching portion temperature detector and merging portion temperature detector. The branching portion temperature detector ascertains at least one of the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger. The merging portion temperature detector for ascertains the temperature of the first fluid flowing toward the first heat-load-processing section after the first fluid which has passed through the first branching channel has merged with the first fluid which has passed through the second branching channel.

[0045] According to the heat pump system of the aspect described above, it is possible to directly ascertain at least either of the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger by using the branching portion temperature detector and to directly ascertain the temperature of the first fluid after merging by using the merging portion temperature detector. The precision of flow rate adjustment control can thereby be improved by performing control so as to reduce the difference between the temperature ascertained by the branching portion temperature detector and the temperature ascertained by the merging portion temperature detector.

[0046] The heat pump system of the nineteenth aspect of the present invention is the heat pump system of any of the first to sixteenth aspects, further comprising: first branching channel flow rate detector for ascertaining the flow rate of the first fluid flowing through the first branching channel; and second branching channel flow rate detector for ascertaining the flow rate of the first fluid flowing through the second branching channel.

[0047] According to the heat pump system of the aspect described above, the precision of flow rate adjustment control can be improved because the flow rate of the first fluid flowing through the first branching channel and the flow rate of the first fluid flowing through the second branching channel can be directly ascertained.

[0048] The heat pump system of the twentieth aspect of the present invention is the heat pump system of any of the first to sixteenth aspects, further comprising: branching portion flow rate detector for ascertaining at least one among the flow rate of the first fluid flowing through the first branching channel and the flow rate of the first fluid flowing through the second branching channel; and merging portion flow rate detector for ascertaining the flow rate of the first fluid flowing toward the first heat-load-processing section after the first fluid flowing through the first branching channel and the first fluid flowing through the second branching channel have merged.

[0049] According to the heat pump system of the aspect described above, it is possible to directly ascertain at least either of the flow rate of the first fluid flowing through the first branching channel and the flow rate of the first fluid flowing through the second branching channel by using the branching

portion flow rate detector and to directly ascertain the flow rate of the first fluid after merging by using the merging portion flow rate detector. It is thereby possible to ascertain the flow rate of the first branching channel or the second branching channel, whichever is not provided with the branching portion flow rate detector, as the difference between the flow rate ascertained by the branching portion flow rate detector and the flow rate ascertained by the merging portion flow rate detector. The precision of flow rate adjustment control can thereby be improved.

[0050] The heat pump system of the twenty-first aspect of the present invention is the heat pump system of any of the first to twentieth aspects, wherein the primary refrigerant flowing from the discharge side of the low-stage-side compression mechanism toward the intake side of the high-stage-side compression mechanism and the first fluid flowing through the first branching channel are in an opposing-flow relationship in the first heat exchanger. The primary refrigerant flowing from the high-stage-side compression mechanism toward the expansion mechanism and the first fluid flowing through the second branching channel are in an opposing-flow relationship in the second heat exchanger.

[0051] According to the heat pump system of the aspect described above, it is possible to minimize the temperature required as the temperature of the primary refrigerant discharged from the low-stage-side compression mechanism and the temperature of the primary refrigerant discharged from the high-stage-side compression mechanism. The drive force of the compression mechanism can thereby be minimized.

[0052] The heat pump system of the twenty-second aspect of the present invention is the heat pump system of any of the first to twenty-first aspects, wherein the first heat-load-processing section is an air-warming heat exchanger for warming air in a disposed target space. The first fluid is a secondary refrigerant.

[0053] According to the heat pump system of the aspect described above, it is possible to warm the space in which the first heat-load-processing section is disposed.

[0054] The heat pump system of the twenty-third aspect of the present invention is the heat pump system of any of the first to twenty-second aspects, wherein the low-stage-side compression mechanism and the high-stage-side compression mechanism have a shared rotating shaft that is rotatably driven, whereby compression work is performed.

[0055] According to the heat pump system of the aspect described above, a rotating shaft is shared in an arrangement having a 180° phase difference, whereby drive efficiency can be improved.

[0056] The heat pump system of the twenty-fourth aspect of the present invention is the heat pump system of any of the first to twenty-third aspects, wherein the controller keeps the discharge pressure of the high-stage-side compression mechanism at a pressure that is equal to or greater than a critical pressure of the primary refrigerant in the flow rate adjustment control. The heat pump is used in an environment in which the ambient temperature of the first heat-load-processing section is a temperature equal to or less than the critical temperature of the primary refrigerant.

[0057] According to the heat pump system of the aspect described above, primary refrigerant in a state exceeding critical pressure is fed to a heat load having a temperature that is lower than the critical temperature of the primary refrigerant, whereby heat release can be carried out in an area in

which the slope of the isotherm of the primary refrigerant is smooth on a Mollier graph. It is therefore possible to perform operation that increases the enthalpy difference between the start and end of the primary refrigerant heat release step.

[0058] The heat pump system of the twenty-fifth aspect of the present invention is the heat pump system of any of the first to twenty-fourth aspects, wherein the primary refrigerant is carbon dioxide.

[0059] According to the heat pump system of the aspect described above, it is possible to implement a refrigeration cycle of a heat pump circuit using a natural refrigerant.

Effects of Invention

[0060] As noted in the description above, the following effects are obtained in accordance with the present invention.

[0061] In the first aspect, it is possible to improve the efficiency of the heat pump system in terms of processing the heat load in the first heat load heat exchanger.

[0062] In the second aspect, it is possible to adapt to the heat load in the first heat load heat exchanger by using only the heat amount obtained by the first fluid in the first heat exchanger and the second heat exchanger.

[0063] In the third aspect, it is possible to bring the temperature of the first fluid fed to the first heat-load-processing section close to the first heat load-corresponding temperature even when in order to increase the efficiency of the heat pump circuit, the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger have exceeded the first heat load-corresponding temperature.

[0064] In the fourth aspect, it is possible to avoid a state in which the temperature of the first fluid flowing through the first heat load circuit has considerably exceeded the first heat-load-corresponding temperature, and to effectively reduce radiation loss.

[0065] In the fifth aspect, not only is it possible to reduce radiation loss, but it is also possible to handle the heat load in the first heat-load-processing section using a low drive force and to further improve efficiency.

[0066] In the sixth aspect, a circulation amount can be ensured and capacity can be increased while suppressing an increase in the compression ratio.

[0067] In the seventh aspect, it is possible to not only increase the degree of superheat of the primary refrigerant taken in by the low-stage-side compression mechanism, but it is also possible to suppress a reduction in the through-rate of the primary refrigerant in the expansion mechanism and to improve capacity.

[0068] In the eighth aspect, it is possible to increase the capacity of the heat pump circuit while adapting to load fluctuations.

[0069] In the ninth aspect, not only can the heat of the primary refrigerant discharged by the high-stage-side compression mechanism be used for both heat load processing in the first heat load circuit and heat load processing in the second heat load circuit, but also a temperature range beyond what is required in the first heat load circuit can be used in the second heat load circuit.

[0070] In the tenth aspect, it is possible to improve heat exchange efficiency because heat exchange can be performed in the second heat exchanger, the third heat exchanger, and the

fourth heat exchanger while the temperature difference between the primary refrigerant and the first and second fluids is kept minimized.

[0071] In the eleventh aspect, the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger and the temperature of the first fluid flowing through portion of the second branching channel that has passed through the second heat exchanger are more readily brought close together.

[0072] In the twelfth aspect, hot water can be made using the temperature of the primary refrigerant discharged from the high-stage-side compression mechanism

[0073] In the thirteenth aspect, it is possible to increase the amount of heat recovered from the primary refrigerant in the heat exchanger having the reduced flow rate among the first heat exchanger and the second heat exchanger.

[0074] In the fourteenth aspect, the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger can be varied so as to reduce the temperature difference, and in the case that the heat load has not changed in the first heat-load-processing section, not only can the temperature difference be reduced, but it is also possible to adapt to the heat load of the first heat-load-processing section by keeping the flow rate of the first fluid fed to the first heat-load-processing section.

[0075] In the fifteenth aspect, it is possible to vary the flow rate so that the temperature difference is reduced. Also, in the case that the heat load is reduced in the first heat-load-processing section, not only can the temperature difference be reduced, but it is also possible to adapt to the heat load in the first heat-load-processing section.

[0076] In the sixteenth aspect, the flow rate of the first fluid flowing through the first heat load circuit can be set to a rate adapted to the heat load in the first heat-load-processing section while the difference between the temperature of the first fluid flowing through section of the first branching channel that has passed through the first heat exchanger and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger is reduced.

[0077] In the seventeenth aspect, the precision of flow rate adjustment control can be improved.

[0078] In the eighteenth aspect; the precision of flow rate adjustment control can be improved by performing control so as to reduce the difference between the temperature ascertained by the branching portion temperature detector and the temperature ascertained by the merging portion temperature detector.

[0079] In the nineteenth aspect, the precision of flow rate adjustment control can be improved.

[0080] In the twentieth aspect, the precision of flow rate adjustment control can be improved.

[0081] In the twenty-first aspect, the drive force of the compression mechanism can be minimized.

[0082] In the twenty-second aspect, it is possible to warm the space in which the first heat-load-processing section is disposed.

[0083] In the twenty-third aspect, a rotating shaft is shared and a 180° phase difference is provided, whereby drive efficiency can be improved.

[0084] In the twenty-fourth aspect, it is possible to perform operation that increases the enthalpy difference between the start and end of the primary refrigerant heat release step.

[0085] In the twenty-fifth aspect, it is possible to implement a refrigeration cycle of a heat pump circuit using a natural refrigerant.

BRIEF DESCRIPTION OF THE DRAWINGS

[0086] FIG. 1 is a schematic structural diagram of a heat pump system according to the first embodiment of the present invention.

[0087] FIG. 2 is a pressure-enthalpy graph of a heat pump circuit according to the first embodiment.

[0088] FIG. 3 is a temperature-entropy graph of the heat pump circuit according to the first embodiment.

[0089] FIG. 4 is a schematic structural diagram of a heat pump system according to the second embodiment.

[0090] FIG. 5 is a schematic structural diagram of a heat pump system according to the third embodiment.

[0091] FIG. 6 is a schematic structural diagram of a heat pump system according to the fourth embodiment.

[0092] FIG. 7 is a schematic structural diagram of a heat pump system according to the fifth embodiment.

[0093] FIG. 8 is a schematic structural diagram of a heat pump system according to Modification A of the fifth embodiment.

[0094] FIG. 9 is a schematic structural diagram of a heat pump system according to Modification B of the fifth embodiment.

[0095] FIG. 10 is a schematic structural diagram of a heat pump system according to Modification C of the fifth embodiment.

[0096] FIG. 11 is a schematic structural diagram of a heat pump system according to the sixth embodiment.

[0097] FIG. 12 is a schematic structural diagram of a heat pump system according to Modification A of the sixth embodiment.

[0098] FIG. 13 is a schematic structural diagram of a heat pump system according to the seventh embodiment.

[0099] FIG. 14 is a schematic structural diagram of a heat pump system according to the eighth embodiment.

[0100] FIG. 15 is a schematic structural diagram of a heat pump system according to the ninth embodiment.

[0101] FIG. 16 is a schematic structural diagram of a heat pump system according to the tenth embodiment.

[0102] FIG. 17 is a schematic structural diagram of a heat pump system according to the eleventh embodiment.

[0103] FIG. 18 is a schematic structural diagram of a heat pump system according to the twelfth embodiment.

[0104] FIG. 19 is a schematic structural diagram of a heat pump system according to the thirteenth embodiment.

[0105] FIG. 20 is a schematic structural diagram of a heat pump system according to Modification <14-5> of the embodiments.

[0106] FIG. 21 is a schematic structural diagram of a heat pump system according to Modification <14-5> of each of the embodiments.

[0107] FIG. 22 is a Mollier graph of Modification <14-8> of the embodiments.

[0108] FIG. 23 is a Mollier graph of Modification <14-9> of the embodiments.

[0109] FIG. 24 is a schematic structural diagram of a heat pump system according to Modification <14-11> of the embodiments.

[0110] FIG. 25 is a schematic structural diagram of a heat pump system according to Modification <14-12> of the embodiments.

[0111] FIG. 26 is a schematic structural diagram of a heat pump system according to Modification <14-13> of the embodiments.

[0112] FIG. 27 shows a comparative example of the Mollier graph of Modification <14-17> of the embodiments.

[0113] FIG. 28 is a Mollier graph of Modification <14-17> of the embodiments.

[0114] FIG. 29 is a Mollier graph of Modification <14-18> of the embodiments.

DESCRIPTION OF EMBODIMENTS

<1> First Embodiment

<1-1> Configuration of the Heat Pump System 1

[0115] FIG. 1 is a schematic structural diagram of a heat pump system 1 according to the first embodiment, which is an embodiment of the present invention.

[0116] The heat pump system 1 is provided with a heat pump circuit 10, an air-warming circuit 60, a hot-water supply circuit 90, an intermediate-pressure water heat exchanger 40, and a high-pressure water heat exchanger 50. The heat pump system 1 is a system that uses the heat obtained by the heat pump circuit 10 not only as heat for air warming via the air-warming circuit 60, but also as heat for hot-water supply via the hot-water supply circuit 90. (Intermediate-pressure water heat exchanger 40)

[0117] The intermediate-pressure water heat exchanger 40 causes heat exchange to be performed between carbon dioxide as the primary refrigerant circulating through the heat pump circuit 10 and water as the secondary refrigerant circulating through the air-warming circuit 60.

(High-Pressure Water Heat Exchanger 50)

[0118] The high-pressure water heat exchanger 50 has a first high-pressure water heat exchanger 51, a second high-pressure water heat exchanger 52, and a third high-pressure water heat exchanger 53. The first high-pressure water heat exchanger 51 causes heat exchange to be performed between carbon dioxide as the primary refrigerant circulating through the heat pump circuit 10 and water for hot-water supply that circulates through the hot-water supply circuit 90. The second high-pressure water heat exchanger 52 causes heat exchange to be performed between carbon dioxide as the primary refrigerant circulating through the heat pump circuit 10 and water as the secondary refrigerant circulating through the air-warming circuit 60. The third high-pressure water heat exchanger 53 causes heat exchange to be performed between carbon dioxide as the primary refrigerant circulating through the heat pump circuit 10 and water for hot-water supply that circulates through the hot-water supply circuit 90.

(Heat Pump Circuit 10)

[0119] The heat pump circuit 10 is a circuit that uses a natural refrigerant and is a circuit through which carbon dioxide is circulated as a primary refrigerant. The heat pump circuit 10 is provided with a low-stage-side compressor 21, a high-stage-side compressor 25, an economizer heat exchanger 7, an injection channel 70, a primary-refrigerant-to-primary-refrigerant heat exchanger 8, a primary bypass 80, an expansion valve 5a, an evaporator 4, an intermediate-

pressure tube 23, a high-pressure tube 27, a low-pressure tube 20, a fan 4f, and a controller 11. The evaporator 4 is disposed, e.g., outdoors.

[0120] The intermediate-pressure tube 23 connects the discharge side of the low-stage-side compressor 21 and the intake side of the high-stage-side compressor 25. The intermediate-pressure tube 23 has a first intermediate-pressure tube 23a, a second intermediate-pressure tube 23b, a third intermediate-pressure tube 23c, and a fourth intermediate-pressure tube 23d.

[0121] The first intermediate-pressure tube 23a connects the discharge side of the low-stage-side compressor 21 and the upstream-side end section of the intermediate-pressure water heat exchanger 40 via a low-stage-discharge point B. An intermediate-pressure temperature sensor 23T for detecting the temperature of the passing primary refrigerant is mounted in the first intermediate-pressure tube 23a. The second intermediate-pressure tube 23b passes through the interior of the intermediate-pressure water heat exchanger 40 while carbon dioxide as the primary refrigerant is allowed to flow therein so that there is no mixing with the water for air warming as the secondary refrigerant. The third intermediate-pressure tube 23c connects the downstream-side end section of the intermediate-pressure water heat exchanger 40 and an injection merging point D via an intermediate-pressure water heat exchanger passage point C. The fourth intermediate-pressure tube 23d connects the injection merging point D and the intake side of the high-stage-side compressor 25. A high-stage intake pressure sensor 24P for detecting the pressure of the passing primary refrigerant and a high-stage intake temperature sensor 24T for detecting the temperature of the passing primary refrigerant are mounted in the fourth intermediate-pressure tube 23d.

[0122] The high-pressure tube 27 connects the discharge side of the high-stage-side compressor 25 and the expansion valve 5 or a primary bypass expansion valve 5b. The high-pressure tube 27 has a first high-pressure tube 27a, a second high-pressure tube 27b, a third high-pressure tube 27c, a fourth high-pressure tube 27d, a fifth high-pressure tube 27e, a sixth high-pressure tube 27f, a seventh high-pressure tube 27g, an eighth high-pressure tube 27h, a ninth high-pressure tube 27i, a tenth high-pressure tube 27j, an eleventh high-pressure tube 27k, a twelfth high-pressure tube 27l, and a thirteenth high-pressure tube 27m.

[0123] The first high-pressure tube 27a connects the discharge side of the high-stage-side compressor 25 and a first high-pressure water heat exchanger 51 via a high-stage discharge point E. A high-pressure pressure sensor 27P for detecting the pressure of the passing primary refrigerant, and a high-pressure temperature sensor 27T for detecting the temperature of the passing primary refrigerant are mounted in the first high-pressure tube 27a. The second high-pressure tube 27b passes through the interior of the first high-pressure water heat exchanger 51 while carbon dioxide as the primary refrigerant is allowed to flow therein so that there is no mixing with the water for hot-water supply. The third high-pressure tube 27c connects the downstream-side end section of the first high-pressure water heat exchanger 51 and the upstream-side end section of the second high-pressure water heat exchanger 52 via a first high-pressure point F. The fourth high-pressure tube 27d passes through the interior of the second high-pressure-water heat exchanger 52 while carbon dioxide as the primary refrigerant is allowed to flow therein so that there is no mixing with the water as secondary refrigerant for air

warming. The fifth high-pressure tube 27e connects the downstream-side end section of the second high-pressure water heat exchanger 52 and the upstream-side end section of the third high-pressure-water heat exchanger 53 via a second high-pressure point G. The sixth high-pressure tube 27f passes through the interior of the third high-pressure-water heat exchanger 53 while carbon dioxide as the primary refrigerant is allowed to flow therein so that there is no mixing with the water as secondary refrigerant for air warming. The seventh high-pressure tube 27g connects the downstream-side end section of the third high-pressure-water heat exchanger 53 and a third high-pressure point H. The eighth high-pressure tube 27h connects the third high-pressure point H and the upstream-side end section in the flow direction of the primary refrigerant toward the expansion valve 5a side in the economizer heat exchanger 7. The ninth high-pressure tube 27i passes through the economizer heat exchanger 7 while primary refrigerant is allowed to flow therein so that there is no mixing with the primary refrigerant flowing through the injection channel 70. The tenth high-pressure tube 27j connects a fourth high-pressure point I and the downstream-side end section in the flow direction of the primary refrigerant toward the expansion valve 5a side in the economizer heat exchanger 7. The eleventh high-pressure tube 27k connects the fourth high-pressure point I and the upstream-side end section in the flow direction of the primary refrigerant toward the expansion valve 5a side in the primary-refrigerant-to-primary-refrigerant heat exchanger 8. The twelfth high-pressure tube 27l passes through the primary-refrigerant-to-primary-refrigerant heat exchanger 8 while primary refrigerant is allowed to flow therein so that there is no mixing between the primary refrigerant flowing through the low-pressure tube 20. The thirteenth high-pressure tube 27m connects the expansion valve 5a and the downstream-side end section in the flow direction of the primary refrigerant toward the expansion valve 5a side in the primary-refrigerant-to-primary-refrigerant heat exchanger 8 via a fifth high-pressure point J.

[0124] The low-pressure tube 20 has a first low-pressure tube 20a, a second low-pressure tube 20b, a third low-pressure tube 20c, a fourth low-pressure tube 20d, and a fifth low-pressure tube 20e. The first low-pressure tube 20a connects the expansion valve 5a and a third low-pressure point M via a first low-pressure point K. The second low-pressure tube 20b connects the third low-pressure point M and the upstream-side end section of the evaporator 4. The third low-pressure tube 20c connects the downstream-side end section of the evaporator 4 and the upstream-side end section of the primary-refrigerant-to-primary-refrigerant heat exchanger 8 in terms of the flow direction of the primary refrigerant in the low-pressure tube 20 via a fourth low-pressure point N. The fourth low-pressure tube 20d passes through the primary-refrigerant-to-primary-refrigerant heat exchanger 8 while the primary refrigerant is allowed to flow therein so that there is no mixing with the primary refrigerant flowing through the twelfth high-pressure tube 27l. The fifth low-pressure tube 20e connects the downstream-side end section of the primary-refrigerant-to-primary-refrigerant heat exchanger 8 in terms of the flow direction of the primary refrigerant in the low-pressure tube 20 and an intake point A, which is the intake side of the low-stage-side compressor 21. A low-pressure pressure sensor 20P for detecting the pressure of the passing primary refrigerant and a low-pressure temperature sensor 20T for detecting the temperature of the passing primary refrigerant are mounted in the fifth low-pressure tube 20e.

[0125] The injection channel 70 has an injection expansion valve 73, a first injection tube 72, a second injection tube 74, a third injection tube 75, and a fourth injection tube 76.

[0126] The first injection tube 72 connects the third high-pressure point H and the injection expansion valve 73. The second injection tube 74 connects the injection expansion valve 73 and the upstream-side end section in terms of the flow direction of the primary refrigerant flowing through the injection channel 70 in the economizer heat exchanger 7 via an injection intermediate-pressure point Q. The third injection tube 75 passes through the economizer heat exchanger 7 while primary refrigerant is allowed to flow therein so that there is no mixing with the primary refrigerant flowing through the ninth high-pressure tube 27i. The fourth injection tube 76 connects the injection merging point D and the downstream-side end section in the flow direction of the primary refrigerant flowing through the injection channel 70 in the economizer heat exchanger 7 via an economizer post-heat-exchange point R.

[0127] In the heat pump circuit 10, the coefficient of performance the heat pump circuit is thus improved because the injection channel 70 is used. For example, in the case that, among other things, the air-warming load is low, operating efficiency can be improved by increasing the injection amount that passes through the injection channel 70, even when the cooling effect of the primary refrigerant cannot be sufficiently obtained in the intermediate-pressure water heat exchanger 40, which is used for improving the efficiency of the heat pump circuit 10. In the heat pump circuit 10, the injection merging point D is provided between the intermediate-pressure water heat exchanger 40 and the high-stage-side compressor 25. Accordingly, high-temperature primary refrigerant discharged from the low-stage-side compressor 21 can be fed to the intermediate-pressure water heat exchanger 40 while being kept in a high-temperature state without being cooled prior to arriving in the intermediate-pressure water heat exchanger 40. For this reason, the water for air warming that passes through the intermediate-pressure water heat exchanger 40 can be brought to a sufficiently high temperature. Furthermore, the third high-pressure point H is provided in a position that allows a portion of the primary refrigerant in the upstream side of the economizer heat exchanger 7 to be branched to the injection channel 70. Therefore, it is possible to avoid a reduction in capacity due to overcooling of the primary refrigerant moving from the low-stage-side compressor 21 toward the high-stage-side compressor 25. The primary bypass 80 has a fourteenth high-pressure tube 27n, a sixth low-pressure tube 20f, and the primary bypass expansion valve 5b. The fourteenth high-pressure tube 27n connects the fourth high-pressure point I and the primary bypass expansion valve 5b. The sixth low-pressure tube 20f connects primary bypass expansion valve 5b and the third low-pressure point M via the second low-pressure point L. Since the primary bypass expansion valve 5b is provided to the primary bypass 80, the controller 11 can adjust the amount of primary refrigerant that passes through the primary-refrigerant-to-primary-refrigerant heat exchanger 8 side. It is therefore possible to make adjustment so that the primary refrigerant taken in by the low-stage-side compressor 21 is at a suitable degree of superheat. Specifically, the controller 11 can increase the flow rate of the primary refrigerant that passes through the primary-refrigerant-to-primary-refrigerant heat exchanger 8 and increase the degree of superheat of the primary refrigerant taken in by the

low-stage-side compressor **21** in the case that the valve opening degree of the primary bypass expansion valve **5b** is reduced, whereby the compression ratio required for the discharge refrigerant temperature of the low-stage-side compressor **21** to reach a target temperature can be minimized. Also, the controller **11** can reduce the flow rate of the primary refrigerant that passes through the primary-refrigerant-to-primary-refrigerant heat exchanger **8** and reduce the degree of superheat of the primary refrigerant taken in by the low-stage-side compressor **21** in the case that the opening degree of the primary bypass expansion valve **5b** is increased, thereby making it possible to avoid a situation in which the intake refrigerant density of the low-stage-side compressor **21** is dramatically reduced and a circulation amount cannot be obtained.

[0128] The controller **11** controls the low-stage-side compressor **21**, the high-stage-side compressor **25**, the injection expansion valve **73**, the expansion valve **5a**, the primary bypass expansion valve **5b**, the fan **4f**, and other components on the basis of values detected by the above-described intermediate-pressure temperature sensor **23T**, the high-stage intake pressure sensor **24P**, the high-stage intake temperature sensor **24T**, the high-pressure pressure sensor **27P**, the high-pressure temperature sensor **27T**, the low-pressure pressure sensor **20P**, the low-pressure temperature sensor **20T**, and the like.

(Air-Warming Circuit **60**)

[0129] The air-warming circuit **60** circulates water as a secondary refrigerant. The air-warming circuit **60** has a radiator **61**, a branch flow mechanism **62**, an air-warming feed tube **65**, an air-warming return tube **66**, an intermediate-pressure-side branching channel **67**, and a high-pressure-side branching channel **68**. The branch flow mechanism **62** includes an air-warming mixing valve **64** and an air-warming pump **63**. The radiator **61** is disposed in a space where air warming will be performed, and warm water as the secondary refrigerant flows therein, whereby the air of the target space is warmed to perform air warming. A radiator temperature sensor **61T** is provided to the radiator **61** in order to detect the temperature of the water for air warming flowing inside the radiator. Although not shown in the drawings, the radiator **61** has a feed port for receiving warm water sent from the air-warming pump **63** and a return port for feeding out water releasing heat in the radiator **61** to the intermediate-pressure-water heat exchanger **40** and the second high-pressure-water heat exchanger **52**. The air-warming return tube **66** connects the return port of the radiator **61** and an air-warming branching point **X**. In the air-warming branching point **X**, the water after releasing heat in the radiator **61** is branched to the intermediate-pressure-side branching channel **67**, which sends the water to the intermediate-pressure-water heat exchanger **40** side, and the high-pressure-side branching channel **68**, which sends the heated water to the second high-pressure-water heat exchanger **52**. An air-warming-return temperature sensor **66T** is provided to the air-warming-return tube **66** in order to detect the temperature of the passing secondary refrigerant for air warming.

[0130] The intermediate-pressure-side branching channel **67** has a first intermediate-pressure-side branching channel **67a**, a second intermediate-pressure-side branching channel **67b**, and a third intermediate-pressure-side branching channel **67c**. The first intermediate-pressure-side branching channel **67a** connects the branching point **X** and the upstream-side end section of the intermediate-pressure-water heat

exchanger **40** in the flow direction of the water in the intermediate-pressure-side branching channel **67**. The second intermediate-pressure-side branching channel **67b** passes through the interior of the intermediate-pressure-water heat exchanger **40** while water for air warming as the secondary refrigerant is allowed to flow therein so that there is no mixing with the carbon dioxide as the primary refrigerant flowing through the second intermediate-pressure tube **23b**. Here, an opposing-flow arrangement is used in the intermediate-pressure-water heat exchanger **40** in which the carbon dioxide as the primary refrigerant flowing through the second intermediate-pressure tube **23b** and the water for air warming as the secondary refrigerant flowing through the second intermediate-pressure-side branching channel **67b** flow in mutually opposite directions. The third intermediate-pressure-side branching channel **67c** connects an air-warming merging point **Y** and the downstream-side end section of the intermediate-pressure-water heat exchanger **40** in the flow direction of the water in the intermediate-pressure-side branching channel **67**. An intermediate-pressure-side branching channel temperature sensor **67T** is provided to the third intermediate-pressure-side branching channel **67c** in order to detect the temperature of the passing water for air warming.

[0131] The high-pressure-side branching channel **68** has a first high-pressure-side branching channel **68a**, a second high-pressure-side branching channel **68b**, and a third high-pressure-side branching channel **68c**. The first high-pressure-side branching channel **68a** connects the air-warming branching point **X** and the upstream-side end section of the second high-pressure-water heat exchanger **52** in the flow direction of the water in the high-pressure-side branching channel **68**. The second high-pressure-side branching channel **68b** passes through the interior of the second high-pressure-water heat exchanger **52** while water for air warming as the secondary refrigerant is allowed to flow therein so that there is no mixing with the carbon dioxide as the primary refrigerant flowing through the fourth high-pressure tube **27d**. In the second high-pressure-water heat exchanger **52**, an opposing-flow arrangement is used in which the carbon dioxide as the primary refrigerant flowing through the fourth high-pressure tube **27d** and the water for air warming as the secondary refrigerant flowing through the second high-pressure-side branching channel **68b** flow in mutually opposite directions. The third high-pressure-side branching channel **68c** connects the air-warming merging point **Y** and the downstream-side end section of the second high-pressure-water heat exchanger **52** in the flow direction of the water in the high-pressure-side branching channel **68**. A high-pressure-side branching channel temperature sensor **68T** is provided to the third high-pressure-side branching channel **68c** in order to detect the temperature of the passing water for air warming.

[0132] The temperature of the water for air warming that is flowing through the first intermediate-pressure-side branching channel **67a** and the temperature of the water for air warming that is flowing through the first high-pressure-side branching channel **68a** have the same temperature distribution because these waters for air warming remain branched by the air-warming branching point **X** and there is no interchange of heat with the exterior. In contrast, the temperature of the water for air warming that flows through the third intermediate-pressure-side branching channel **67c** becomes a temperature that corresponds to the heat amount obtained by heat exchange with the primary refrigerant flowing through the second intermediate-pressure tube **23b** in the intermedi-

ate-pressure-water heat exchanger 40. The temperature of the water for air warming flowing through the third high-pressure-side branching channel 68c becomes a temperature that corresponds to the heat amount obtained by heat exchange with the primary refrigerant flowing through the fourth high-pressure tube 27d in the second high-pressure-water heat exchanger 52. Therefore, there are cases in which the temperature of the water for air warming flowing through the third intermediate-pressure-side branching channel 67c and the temperature of the water for air warming flowing through the third high-pressure-side branching channel 68c are different.

[0133] The air-warming feed tube 65 connects the air-warming merging point Y and the feed port of the radiator 61. The air-warming pump 63 for adjusting the flow rate of the water for air warming that passes through the air-warming feed tube 65 is provided at a midway point of the air-warming feed tube 65. The air-warming mixing valve 64 is provided at the air-warming merging point Y for merging the water for air warming that has passed through the third intermediate-pressure-side branching channel 67c and the water for air warming that has passed through the third high-pressure-side branching channel 68c. The air-warming mixing valve 64 adjusts the opening degree of the portion connected to the third intermediate-pressure-side branching channel 67c side and the opening degree of the portion connected to the third high-pressure-side branching channel 68c side to thereby adjust the ratio of the flow rate of the water for air warming flowing to the intermediate-pressure-side branching channel 67 and the flow rate of the water for air warming flowing to the third high-pressure-side branching channel 68c.

[0134] The controller 11 controls the branch flow ratio in the air-warming mixing valve 64 and the flow rate through the air-warming pump 63 so that the secondary refrigerant having a required temperature in the radiator 61 can be fed on the basis of, among other things, the temperature detected by the above-described radiator temperature sensor 61T, the intermediate-pressure-side branching temperature sensor 67T, and the high-pressure-side branching temperature sensor 68T.

(Hot-Water Supply Circuit 90)

[0135] The hot-water supply circuit 90 circulates water for hot-water supply. The hot-water supply circuit 90 has a hot-water storage tank 91, a water supply tube 94, a hot-water supply tube 98, a hot-water supply bypass tube 99, a hot-water supply mixing valve 93, a hot-water supply heat pump tube 95, and a hot-water supply pump 92.

[0136] Although not shown in the drawings, a circulation feed port and a circulation return port are provided to the hot-water storage tank 91. Normal-temperature water is fed into the hot-water storage tank 91 from near the lower-end section of the hot-water storage tank 91 via the water supply tube 94 after having passed through external city water (not shown). The hot-water supply heat pump tube 95 has a first hot-water supply heat pump tube 95a, a second hot-water supply heat pump tube 95b, a third hot-water supply heat pump tube 95c, a fourth hot-water supply heat pump tube 95d, a fifth hot-water supply heat pump tube 95e, and a sixth hot-water supply heat pump tube 95f.

[0137] The first hot-water supply heat pump tube 95a connects the circulation feed port of the hot-water storage tank 91 and the hot-water supply pump 92. A hot-water supply water-intake temperature sensor 94T is provided to the first hot-

water supply heat pump tube 95a in order to detect the temperature of the passing water for hot-water supply. The second hot-water supply heat pump tube 95b connects the hot-water supply pump 92 and the upstream-side end section of the third high-pressure-water heat exchanger 53 in the flow direction of the water in the hot-water supply heat pump tube 95. The third hot-water supply heat pump tube 95c passes through the interior of the third high-pressure-water heat exchanger 53 while water for hot-water supply is allowed to flow therein so that there is no mixing with the carbon dioxide as the primary refrigerant flowing through the sixth high-pressure tube 27f. Here, an opposing-flow arrangement is used in the third high-pressure-water heat exchanger 53 in which the carbon dioxide as the primary refrigerant flowing through the sixth high-pressure tube 27f and the water for hot-water supply flowing through the third hot-water supply heat pump tube 95c flow in mutually opposite directions. The fourth hot-water supply heat pump tube 95d connects the downstream-side end section of the third high-pressure-water heat exchanger 53 in the flow direction of the water in the hot-water supply heat pump tube 95, and the upstream-side end section of the first high-pressure water heat exchanger 51 in the flow direction of the water in the hot-water supply heat pump tube 95. A hot-water supply intermediate temperature sensor 95T for detecting the temperature of the passing water for hot-water supply is provided in the fourth hot-water supply heat pump tube 95d. Heat exchange is not performed in the second high-pressure-water heat exchanger 52 between the water for hot-water supply and the carbon dioxide as the primary refrigerant. The fifth hot-water supply heat pump tube 95e passes through the interior of the first high-pressure water heat exchanger 51 while water for hot-water supply is allowed to flow therein so that there is no mixing with the carbon dioxide as the primary refrigerant flowing through the second high-pressure tube 27b. Here, an opposing-flow arrangement is used in the first high-pressure water heat exchanger 51 in which the carbon dioxide as the primary refrigerant flowing through the second high-pressure tube 27b and the water for hot-water supply flowing through the fifth hot-water supply heat pump tube 95e flow in mutually opposite directions. The sixth hot-water supply heat pump tube 95f connects the circulation return port of the hot-water storage tank 91 and the downstream-side end section of the first high-pressure water heat exchanger 51 in the flow direction of the water in the hot-water supply heat pump tube 95. A hot-water supply hot-water outlet temperature sensor 98T is provided to the sixth hot-water supply heat pump tube 95f in order to detect the temperature of the passing water for hot-water supply.

[0138] The hot-water supply tube 98 directs hot water accumulated in the hot-water storage tank 91 from the vicinity of the upstream-side end section of the hot-water storage tank 91 to a location (not shown) in which the hot water is to be used. The water supply tube 94 is provided with a water supply branching point W, which is a branching portion for branching from the flow that moves toward the hot-water storage tank 91 side. The hot-water supply tube 98 is provided with a hot-water supply merging point Z for merging with the flow that moves from the hot-water storage tank 91 toward a location in which the hot water is to be used. The hot-water supply bypass tube 99 connects the water supply branching point W and the hot-water supply merging point Z. The hot-water supply mixing valve 93 is provided at the hot-water supply merging point Z and is capable of adjusting the mixing ratio

of the hot water sent from the hot-water storage tank **91** via the hot-water supply tube **98** and the normal-temperature water fed from city water via the hot-water supply bypass tube **99**. The mixing ratio in the hot-water supply mixing valve **93** is adjusted to thereby adjust the temperature of the water sent to the location in which it is to be used.

[0139] The controller **11** controls the flow rate through the hot-water supply pump **92** on the basis of the temperatures and the like detected by the above-described hot-water supply water-intake temperature sensor **94T**, the hot-water supply intermediate temperature sensor **95T**, the hot-water supply hot-water outlet temperature sensor **98T**, and the like.

<1-2> Operation of the Heat Pump Circuit 10

[0140] FIG. 2 is a pressure-enthalpy graph of the case in which the heat pump system **1** is operated. FIG. 3 is a temperature-entropy graph of the case in which the heat pump system **1** is operated.

[0141] The state of temperature distribution of the primary refrigerant is described below using a specific example.

[0142] The low-stage-side compressor **21** compresses (point B) the primary refrigerant (point A) at about 22° C. flowing from the low-pressure tube **20**, so that the target discharge temperature reaches about 90° C. The pressure of the primary refrigerant flowing through the low-pressure tube **20** is adjusted by the controller **11** so as to become a reduced pressure (evaporative pressure) capable of causing the carbon dioxide as the primary refrigerant to evaporate using the ambient temperature in the location where the evaporator **4** is disposed.

[0143] The primary refrigerant discharged from the low-stage-side compressor **21** flows into the second intermediate-pressure tube **23b** inside the intermediate-pressure-water heat exchanger **40** via the first intermediate-pressure tube **23a**. The primary refrigerant which has flowed into the intermediate-pressure-water heat exchanger **40** is cooled (point C) to about 35° C. by heat exchange with the water as secondary refrigerant for air warming which is passing through the second intermediate-pressure-side branching channel **67b**. Here, the primary refrigerant and the secondary refrigerant in the intermediate-pressure-water heat exchanger **40** are flowing in an opposing-flow arrangement, and the primary refrigerant is therefore effectively cooled by the secondary refrigerant, which is in a state cooled to about 30° C. by heat release in the radiator **61**, in the vicinity of the outlet of the second intermediate-pressure tube **23b** inside the intermediate-pressure-water heat exchanger **40**.

[0144] The primary refrigerant which has passed through the intermediate-pressure-water heat exchanger **40** is further cooled to about 30° C. at the injection merging point D of the third intermediate-pressure tube **23c** by merging with the primary refrigerant at about 27° C. flowing in via the injection channel **70**. Here, the controller **11** performs a control such that to achieves the primary refrigerant merged at the injection merging point D has a degree of superheat or a supercritical state. Furthermore, at this point, the controller **11** performs a control so that the target temperature of the primary refrigerant discharged from the high-stage-side compressor **25** is brought to 90° C., which is the same as the target temperature of the primary refrigerant discharged from the low-stage-side compressor **21**, while the high-stage-side compressor **25** is driven with the primary refrigerant merged in the injection merging point D being at the same compression ratio as the compression ratio in the low-stage-side com-

pressor **21**. The controller **11** performs a control so as to adjust the heat balance in the intermediate-pressure-water heat exchanger **40** and the injection channel **70** in relation to the primary refrigerant to be taken into the high-stage-side compressor **25**.

[0145] The primary refrigerant merged in the injection merging point D is taken into the high-stage-side compressor **25**, and the primary refrigerant is further compressed so that the target discharge temperature reaches about 90° C., which is the same temperature as the target temperature of the discharge refrigerant of the low-stage-side compressor **21**. At this point, the high-stage-side compressor **25** is controlled by the controller **11** so as to compress the primary refrigerant and bring the discharge refrigerant pressure to a pressure that exceeds the critical pressure of the primary refrigerant (point E).

[0146] The primary refrigerant discharged by the high-stage-side compressor **25** flows into the second high-pressure tube **27b** inside the first high-pressure water heat exchanger **51** via the first high-pressure tube **27a**. The primary refrigerant that has flowed into the first high-pressure water heat exchanger **51** undergoes heat exchange with the water for hot-water supply passing through the fifth hot-water supply heat pump tube **95e** and is thereby cooled to about 85° C. (point F). The temperature continuously changes because the primary refrigerant releases heat while being kept in a state of having exceeded critical pressure. Here, the primary refrigerant and the secondary refrigerant in the first high-pressure water heat exchanger **51** are flowing in an opposing-flow arrangement, and the primary refrigerant is therefore effectively cooled by the water for hot-water supply, which is at about 30° C. and not yet sufficiently heated, in the vicinity of the outlet of the second high-pressure tube **27b** inside the first high-pressure water heat exchanger **51**.

[0147] The primary refrigerant which has passed through the first high-pressure water heat exchanger **51** flows into the fourth high-pressure tube **27d** inside the second high-pressure-water heat exchanger **52** via the third high-pressure tube **27c**. The primary refrigerant which has flowed into the second high-pressure-water heat exchanger **52** is cooled to about 35° C. (point G) by undergoing heat exchange with the water as the secondary refrigerant for air warming that is passed through the second high-pressure-side branching channel **68b**. Here, the primary refrigerant and the secondary refrigerant in the second high-pressure-water heat exchanger **52** are flowing in an opposing-flow arrangement, and the primary refrigerant is therefore effectively cooled by the secondary refrigerant, which is in a cooled state of about 30° C. having released heat in the radiator **61**, in the vicinity of the outlet of the fourth high-pressure tube **27d** inside the second high-pressure-water heat exchanger **52**.

[0148] The primary refrigerant which has passed through the second high-pressure-water heat exchanger **52** flows into the sixth high-pressure tube **27f** inside the third high-pressure-water heat exchanger **53** via the fifth high-pressure tube **27e**. The primary refrigerant which has flowed into the third high-pressure-water heat exchanger **53** undergoes heat exchange with the water for hot-water supply that is passing through the third hot-water supply heat pump tube **95c** and is further cooled to about 30° C. (point H). Here, the primary refrigerant and the secondary refrigerant in the third high-pressure-water heat exchanger **53** are flowing in an opposing flow arrangement, and the primary refrigerant is therefore effectively cooled by the water for hot-water supply at about

20° C., which is slightly increased from the temperature of the municipal water due to mixing in the hot-water storage tank 91, in the vicinity of the outlet of the sixth high-pressure tube 27f inside the third high-pressure-water heat exchanger 53. The primary refrigerant which has passed through the third high-pressure-water heat exchanger 53 reaches the third high-pressure point H via the seventh high-pressure tube 27g. [0149] Here, the high-pressure water heat exchanger 50 is divided into three heat exchangers, and temperature changes occur in the heat release process because the primary refrigerant flowing through the high-pressure water heat exchanger 50 is in a supercritical state, and the range of temperature variation (30° C. to 65° C.) of the water as the secondary refrigerant circulating through the air-warming circuit 60 is included in the range of temperature variation (20° C. to 90° C.) of the water for hot-water supply in the hot-water supply circuit 90. Heat exchange with the primary refrigerant in a relatively low-temperature state and heat exchange with the primary refrigerant in a relatively high-temperature state among the primary refrigerant discharged from the high-stage-side compressor 25 is used for heat exchange for hot-water supply, and the heat exchange with the primary refrigerant in an intermediate-temperature state is used for heat exchange with the secondary refrigerant for air warming so as to adapt to this temperature distribution. It is thereby possible to improve heat exchange efficiency because the temperature difference between the fluids for carrying out heat exchange can be minimized in not only heat exchange between the primary refrigerant and the water for hot-water supply, but also in heat exchange between the primary refrigerant and the water for air warming.

[0150] The primary refrigerant which has reached the third high-pressure point H is branched into a flow that moves toward the expansion valve 5a side via the eighth high-pressure tube 27h and a flow that moves toward the injection channel 70 side. The amount of branching at this point is controlled by the controller 11 by adjusting the opening degree of the injection expansion valve 73. The primary refrigerant branched to the injection channel 70 side passes through the first injection tube 72 and is depressurized in the injection expansion valve 73; and the temperature of the primary refrigerant is reduced to about 23° C. (point Q).

[0151] The primary refrigerant depressurized in the injection expansion valve 73 flows into the third injection tube 75 inside the economizer heat exchanger 7 via the second injection tube 74. The primary refrigerant which has flowed into the economizer heat exchanger 7 undergoes heat exchange with the primary refrigerant flowing through the ninth high-pressure tube 27i at about 30° C. and is heated to about 27° C. (point R).

[0152] The primary refrigerant at about 27° C. that has passed through the third injection tube 75 inside the economizer heat exchanger 7 merges with the primary refrigerant flowing through the intermediate-pressure tube 23 at the above-described injection merging point D via the fourth injection tube 76.

[0153] Of the primary refrigerant that has arrived at the third high-pressure point H, the primary refrigerant at about 30° C. which has not flowed to the injection channel 70 side flows into the ninth high-pressure tube 27i inside the economizer heat exchanger 7 via the eighth high-pressure tube 27h. The primary refrigerant at about 30° C. which has flowed into the ninth high-pressure tube 27i inside the economizer heat exchanger 7 undergoes heat exchange with the primary

refrigerant at about 27° C. that is flowing through the third injection tube 75, as described above, and is thereby further cooled to about 25° C. (point I). The primary refrigerant which has passed through the ninth high-pressure tube 27i inside the economizer heat exchanger 7 arrives at the fourth high-pressure point I via the tenth high-pressure tube 27j.

[0154] The primary refrigerant which has reached the fourth high-pressure point I is branched into a flow that moves toward the primary bypass 80 side and a flow that moves toward the eleventh high-pressure tube 27k side. The amount of branching at this point is adjusted by the controller 11 which controls the opening degree of the primary bypass expansion valve 5b. The primary refrigerant that has flowed through the eleventh high-pressure tube 27k flows into the twelfth high-pressure tube 27l inside the primary-refrigerant-to-primary-refrigerant heat exchanger 8. The primary refrigerant at about 25° C. having flowed into the twelfth high-pressure tube 27l inside the primary-refrigerant-to-primary-refrigerant heat exchanger 8 undergoes heat exchange with the primary refrigerant at about -3° C. that flows through the fourth low-pressure tube 20d and is cooled to about 20° C. (point J).

[0155] The primary refrigerant that has passed through the twelfth high-pressure tube 27l inside the primary-refrigerant-to-primary-refrigerant heat exchanger 8 flows to the expansion valve 5a via the thirteenth high-pressure tube 27m. In the expansion valve 5a, the opening degree is adjusted by the controller 11, whereby the amount of depressurization of the passing primary refrigerant is adjusted, the refrigerant pressure of the primary refrigerant that has passed by is reduced, and the refrigerant temperature is also reduced to about -3° C. (point K). Here, the amount of depressurization of the primary refrigerant is adjusted by the controller 11, whereby the pressure is reduced to a pressure that is equal to or less than the critical pressure to achieve a gas-liquid two-phase state.

[0156] In the heat pump circuit 10, not only can the primary refrigerant be cooled by the economizer heat exchanger 7; it can also be further cooled by the primary-refrigerant-to-primary-refrigerant heat exchanger 8. In the heat pump circuit 10 it is possible to use the primary refrigerant of the intake side of the low-stage-side compressor 21 through which primary refrigerant at the lowest temperature flows in the cooling of the primary refrigerant flowing through the primary-refrigerant-to-primary-refrigerant heat exchanger 8. The density of the primary refrigerant passing through the expansion valve 5a can thereby be increased, and the circulation amount of the primary refrigerant in the heat pump circuit 10 can be increased.

[0157] The primary refrigerant that has passed through the expansion valve 5a flows to the third low-pressure point M via the first low-pressure tube 20a and merges with the primary refrigerant flowing in through the sixth low-pressure tube 20f (point M).

[0158] Of the primary refrigerant that has arrived at the fourth high-pressure point I, the primary refrigerant at about 25° C. which has not flowed to the eleventh high-pressure tube 27k side flows to the primary bypass 80 side and flows to the primary bypass expansion valve 5b via the fourteenth high-pressure tube 27n. The opening degree of the primary bypass expansion valve 5b is adjusted by the controller 11, whereby the amount of depressurization of the primary refrigerant passing through is adjusted, the refrigerant pressure of the primary refrigerant that has passed through is reduced, and the refrigerant temperature is also reduced to

about -3°C . (point L). In this case as well, the amount of depressurization of the primary refrigerant is adjusted by the controller 11 in similar fashion to point K, whereby the pressure is reduced to a pressure that is equal to or less than the critical pressure to achieve a gas-liquid two-phase state.

[0159] The primary refrigerant that has passed through the primary bypass expansion valve 5b flows to the third low-pressure point M via the sixth low-pressure tube 20f and merges with the primary refrigerant that has flowed in via the first low-pressure tube 20a described above (point M).

[0160] The primary refrigerant at about -3°C . which has merged at the third low-pressure point M flows into the evaporator 4 via the second low-pressure tube 20b. The primary refrigerant which has flowed into the evaporator 4 undergoes heat exchange with air actively fed by the fan 4f to the evaporator 4. The primary refrigerant at about -3°C . in a gas-liquid two-phase state evaporates while the temperature is kept constant by heat exchange in the evaporator 4 (latent heat variation) to increase dryness and to achieve a nearly saturated state (point N).

[0161] The primary refrigerant that has passed through the evaporator 4 flows into the fourth low-pressure tube 20d inside the primary-refrigerant-to-primary-refrigerant heat exchanger 8 via the third low-pressure tube 20c. The primary refrigerant at about -3°C . which flows through the fourth low-pressure tube 20d of the primary-refrigerant-to-primary-refrigerant heat exchanger 8 undergoes heat exchange with the primary refrigerant at about 25°C . that flows through the twelfth high-pressure tube 27l, as described above, and is thereby heated to about 22°C . to achieve a state with a degree of superheat (point A).

[0162] The primary refrigerant which has passed through the fourth low-pressure tube 20d inside the primary-refrigerant-to-primary-refrigerant heat exchanger 8 becomes superheated state and is taken into the low-stage-side compressor 21.

[0163] The primary refrigerant circulates in the heat pump circuit 10 in the manner described above.

<1-3> Operation of the Air-Warming Circuit 60

[0164] The controller 11 performs a control so that water as the secondary refrigerant at about 65°C . is fed to the radiator 61 in order to warm the space in which the radiator 61 is disposed.

[0165] The state of temperature distribution of the secondary refrigerant for air warming is described below using a specific example.

[0166] The water as the secondary refrigerant for air warming which released heat while passing through the interior of the radiator 61 falls to a temperature of about 35°C . (although this depends on the performance of the radiator 61 and level of the air-warming load) and flows to the air-warming branching point X via the air-warming-return tube 66.

[0167] The flow toward the intermediate-pressure-side branching channel 67 and the flow toward the high-pressure-side branching channel 68 side are branching at the air-warming branching point X.

[0168] The secondary refrigerant which has flowed from the air-warming branching point X toward the intermediate-pressure-side branching channel 67 side flows into the second intermediate-pressure-side branching channel 67b inside the intermediate-pressure-water heat exchanger 40 via the first intermediate-pressure-side branching channel 67a. The secondary refrigerant flowing through the second intermediate-

pressure-side branching channel 67b inside the intermediate-pressure-water heat exchanger 40 is heated by the primary refrigerant passing through the second intermediate-pressure tube 23b, as described above, whereby the temperature of the secondary refrigerant at about 30°C . is increased to about 65°C . As described above, the primary refrigerant and the secondary refrigerant in the intermediate-pressure-water heat exchanger 40 are flowing in an opposing-flow arrangement, and the secondary refrigerant is therefore effectively heated by the primary refrigerant at about 90°C ., which is a relatively high temperature, in the vicinity of the outlet of the second intermediate-pressure-side branching channel 67b inside the intermediate-pressure-water heat exchanger 40. The secondary refrigerant which passed through the second intermediate-pressure-side branching channel 67b inside the intermediate-pressure-water heat exchanger 40 and was warmed to about 65°C . passes through the third intermediate-pressure-side branching channel 67c and flows to the air-warming merging point Y.

[0169] The secondary refrigerant that flows from the air-warming branching point X toward the high-pressure-side branching channel 68 side flows into the second high-pressure-side branching channel 68b inside the second high-pressure-water heat exchanger 52 by way of the first high-pressure-side branching channel 68a. The secondary refrigerant flowing through the second high-pressure-side branching channel 68b inside the second high-pressure-water heat exchanger 52 is heated by the primary refrigerant passing through the fourth high-pressure tube 27d, as described above, whereby the temperature of the secondary refrigerant at about 30°C . is increased to about 65°C . As described above, the primary refrigerant and the secondary refrigerant in the second high-pressure-water heat exchanger 52 are flowing in an opposing-flow arrangement, and the secondary refrigerant is therefore effectively heated by the primary refrigerant at about 85°C ., which is a relatively high temperature, in the vicinity of the outlet of the second high-pressure-side branching channel 68b inside the second high-pressure-water heat exchanger 52. The secondary refrigerant which passed through the second high-pressure-side branching channel 68b inside the second high-pressure-water heat exchanger 52 and was warmed to about 65°C . passes through the third high-pressure-side branching channel 68c and flows to the air-warming merging point Y.

[0170] The secondary refrigerant which has passed through the third intermediate-pressure-side branching channel 67c and the secondary refrigerant which has passed through the third high-pressure-side branching channel 68c are merged at the air-warming merging point Y. The controller 11 adjusts the opening degree of the intermediate-pressure-side branching channel 67 side of the air-warming mixing valve 64t and the opening degree of the high-pressure-side branching channel 68 side of the air-warming mixing valve 64 to thereby adjust the flow rate of the secondary refrigerant flowing through the intermediate-pressure-side branching channel 67 side and the flow rate of the secondary refrigerant flowing through the high-pressure-side branching channel 68 side. The controller 11 can thereby perform a control so that the temperature of the secondary refrigerant merged in the air-warming merging point Y becomes the temperature requested in the radiator 61, by adjusting the flow rate of the secondary refrigerant that passes through the air-warming pump 63 while adjusting the ratio of the amount by which the secondary refrigerant circulating through the air-warming circuit 60

is heated in the intermediate-pressure-water heat exchanger **40** side and the amount by which the secondary refrigerant is heated in the second high-pressure-water heat exchanger **52** side.

[0171] In this manner, the secondary refrigerant merged at the air-warming merging point Y and heated to about 65° C. is fed to the radiator **61** via the air-warming feed tube **65**. The secondary refrigerant circulates in the air-warming circuit **60** in the manner described above.

<1-4> Operation of the Hot-Water Supply Circuit **90**

[0172] The controller **11** controls the flow rate of the hot-water supply pump **92** so that hot water at about 90° C. can be stored in the hot-water storage tank **91**.

[0173] The state of temperature distribution of the water for hot-water supply is described below using a specific example.

[0174] Water including inflowing city water at a relatively low temperature in the lower part of the hot-water storage tank **91** flows toward the hot-water supply heat pump tube **95** at a temperature of about 20° C.

[0175] Water for hot-water supply at about 20° C. which has passed through the first hot-water supply heat pump tube **95a** and the second hot-water supply heat pump tube **95b** flows into the third hot-water supply heat pump tube **95c** inside the third high-pressure-water heat exchanger **53**. The water for hot-water supply that flows through the third hot-water supply heat pump tube **95c** inside the third high-pressure-water heat exchanger **53** is heated by the primary refrigerant at about 35° C. that passes through the sixth high-pressure tube **27f** inside the third high-pressure-water heat exchanger **53**, as described above, whereby the temperature of the water for hot-water supply at about 20° C. is increased to about 30° C. As described above, an opposing-flow arrangement is used in the third high-pressure-water heat exchanger **53** in which the primary refrigerant and the secondary refrigerant flow in mutually opposite directions, and the secondary refrigerant is thereby effectively heated by the primary refrigerant at about 35° C., which is a relatively high temperature, in the vicinity of the outlet of the third hot-water supply heat pump tube **95c** inside the third high-pressure-water heat exchanger **53**.

[0176] The water for hot-water supply warmed to about 30° C. in the third high-pressure-water heat exchanger **53** flows through the fourth hot-water supply heat pump tube **95d** into the fifth hot-water supply heat pump tube **95e** inside the first high-pressure water heat exchanger **51**. The water for hot-water supply flowing through the fifth hot-water supply heat pump tube **95e** inside the first high-pressure water heat exchanger **51** is heated by the primary refrigerant at about 90° C. passing through the second high-pressure tube **27b** inside the first high-pressure water heat exchanger **51**, as described above, whereby the temperature of the water for hot-water supply at about 30° C. is increased to about 90° C. As described above, the primary refrigerant and the secondary refrigerant in the first high-pressure water heat exchanger **51** are flowing in an opposing-flow arrangement, and the secondary refrigerant is therefore effectively heated by the primary refrigerant at about 90° C., which is a relatively high temperature, in the vicinity of the outlet of the fifth hot-water supply heat pump tube **95e** inside the first high-pressure water heat exchanger **51**.

[0177] The water for hot-water supply heated to about 90° C. in the first high-pressure water heat exchanger **51** flows

through the sixth hot-water supply heat pump tube **95f** to the upper part of the hot-water storage tank **91**.

[0178] In this manner, water for hot-water supply circulates through the hot-water supply circuit **90**, whereby the temperature of the water for hot-water supply stored inside the hot-water storage tank **91** can be increased.

<1-5> Secondary Refrigerant-Temperature Equalization Control

[0179] As described above, the controller **11** operates the heat pump circuit **10** so that cycle efficiency can be kept as optimal as possible while making it possible to feed to each of the circuits a heat amount that can adapt to not only the air warming load of the air-warming circuit **60**, but also to the hot-water supply load of the hot-water supply circuit **90**. Specifically, in relation to the air-warming circuit **60**, the controller **11** controls the low-stage-side compressor **21**, the high-stage-side compressor **25**, the expansion valve **5a**, and the like so that the temperature of the primary refrigerant flowing into the intermediate-pressure-water heat exchanger **40** and the temperature of the primary refrigerant flowing into the second high-pressure-water heat exchanger **52** are both higher temperatures than the temperature required in the radiator **61**, while the temperature of the primary refrigerant flowing into the intermediate-pressure-water heat exchanger **40** is made to be a higher temperature than the temperature of the secondary refrigerant for air warming that flows into the intermediate-pressure-water heat exchanger **40**, and while the temperature of the primary refrigerant flowing into the second high-pressure-water heat exchanger **52** is made to be a higher temperature than the temperature of the secondary refrigerant for air warming that flows into the second high-pressure-water heat exchanger **52**.

[0180] The controller **11** performs a control so that the temperature obtained after the heat released during passage through the first high-pressure water heat exchanger **51** has been subtracted from the target discharge temperature of the high-stage-side compressor **25** becomes greater than the temperature requested by the radiator **61**, while the target discharge temperature of the low-stage-side compressor **21** is made to be greater than the temperature requested by the radiator **61**. Also, the controller **11** performs a control so that the compression ratio of the low-stage-side compressor **21** and the compression ratio of the high-stage-side compressor **25** are equal and are made as low as possible with the evaporation temperature having been established based on the installation environment of the evaporator **4**. In order to achieve these objects, the controller **11** specifically controls the low-stage-side compressor **21**, the high-stage-side compressor **25**, the expansion valve **5a**, the injection expansion valve **73**, the primary bypass expansion valve **5b**, and the fan **4f** of the heat pump circuit **10**. The temperature of the primary refrigerant is controlled by the controller **11** so as to be equal to or less than a predetermined high-temperature limit value because when the temperature of the primary refrigerant is excessively high, scale (scale or the like) is liable to form on the inner surface of the tubes through which flows the secondary refrigerant for air warming performing heat exchange and/or the inner surfaces of the tubes of the water for hot-water supply performing heat exchange.

[0181] The controller **11** performs secondary refrigerant-temperature equalization control so that the temperature of the secondary refrigerant flowing through the third intermediate-pressure-side branching channel **67c** of the air-warming

circuit 60 and the temperature of the secondary refrigerant flowing through third high-pressure-side branching channel 68c become the same temperature, while making it possible to maintain to the extent possible an operating state having optimal cycle efficiency in the heat pump circuit 10 side described above. The controller 11 not only merely performs a control so that the temperature of the secondary refrigerant flowing through the third intermediate-pressure-side branching channel 67c and the temperature of the secondary refrigerant flowing through the third high-pressure-side branching channel 68c are equalized, but also performs a control so that the equalized temperature matches the temperature requested by the radiator 61. The controller 11 specifically controls to thereby bring the temperature into conformity with the temperature requested by the radiator 61, by performing: mixing-ratio control for controlling the mixing ratio of the air warming mixing valve 64 to thereby adjust the ratio of the flow rate of the secondary refrigerant for air warming flowing through the intermediate-pressure-side branching channel 67 and the flow rate of the secondary refrigerant for air warming flowing through the high-pressure-side branching channel 68; and flow-rate control for controlling the flow rate through the air-warming pump 63 to adjust the flow rate of the secondary refrigerant for air warming fed to the radiator 61.

[0182] In order to set the temperature of the secondary refrigerant for air warming flowing through the third intermediate-pressure-side branching channel 67c and the temperature of the secondary refrigerant for air warming flowing through the third high-pressure-side branching channel 68c to the same temperature, the controller 11 controls the air-warming mixing valve 64 so that the flow rate of the secondary refrigerant having a lower temperature is reduced and the flow rate of the secondary refrigerant having a higher temperature is increased on the basis of the temperature detected by the intermediate-pressure-side branching temperature sensor 67T and the temperature detected by the high-pressure-side branching temperature sensor 68T. The flow speed of the secondary refrigerant having a lower temperature is thereby reduced when the flow rate is reduced, and the time for the secondary refrigerant to receive heat from the primary refrigerant in the heat exchange with the primary refrigerant can be extended, and the temperature is increased as a result. Conversely, the flow speed of the secondary refrigerant having a higher temperature is thereby increased when the flow rate is increased, and the time for the secondary refrigerant to receive heat from the primary refrigerant in the heat exchange with the primary refrigerant can be shortened, and the temperature is reduced as a result. In this manner, the difference between the temperature of the secondary refrigerant for air warming flowing through the third intermediate-pressure-side branching channel 67c and the temperature of the secondary refrigerant for air warming flowing through the third high-pressure-side branching channel 68c is reduced.

[0183] As used herein, the temperature requested by the radiator 61 is the value of a temperature having a constant width, which is described below.

[0184] In the air-warming circuit 60, the required heat release amount in the radiator 61 by the secondary refrigerant for air warming can be set by input by the user. The controller 11 controls the air-warming mixing valve 64 and the air-warming pump 63 so as to ensure the heat release amount in the radiator 61 requested by the user. Specifically, control for ensuring the heat release amount requested by the radiator 61 includes the case in which the temperature of the secondary

refrigerant for air warming is kept low while the flow rate of the secondary refrigerant for air warming passing through the air-warming pump 63 is increased, the case in which the temperature of the secondary refrigerant for air warming is set high while the flow rate of the secondary refrigerant for air warming passing through the air-warming pump 63 is reduced, and other cases. In other words, in the case that the same heat amount is to be ensured, the temperature required as the temperature of the secondary refrigerant for air warming in the case that the flow rate through the air-warming pump 63 has been increased to a designated value is a lower temperature than the temperature required as the temperature of the secondary refrigerant for air warming in the case that the flow rate of through the air-warming pump 63 has been made less than the designated value. Conversely, in the case that the same heat amount is to be ensured, the temperature required as the temperature of the secondary refrigerant for air warming in the case that the flow rate through the air-warming pump 63 has been reduced to another value is a higher temperature than the temperature required as the temperature of the secondary refrigerant for air warming in the case that the flow rate has been increased above the other value. Furthermore, the temperature of the secondary refrigerant fed to the radiator 61 must be a higher temperature than the ambient temperature of the radiator 61 (the temperature detected by the radiator temperature sensor 61T) because an object is to warm the air of the surrounding space in which the radiator 61 is disposed. The temperature requested by the radiator 61 is a higher temperature than that detected by the radiator temperature sensor 61T and has a temperature width that corresponds to a flow rate range capable of ensuring the heat release amount requested by the radiator 61. It is furthermore possible to cause the heat release performance of the radiator 61 itself to be reflected to limit the temperature width.

[0185] The temperature of the secondary refrigerant for air warming flowing through the air-warming feed tube 65 toward the radiator 61 is the temperature obtained after the merging of the secondary refrigerant for air warming that has flowed in through the intermediate-pressure-side branching channel 67 and the secondary refrigerant for air warming that has flowed in through the high-pressure-side branching channel 68 in the air-warming merging point Y.

[0186] Therefore, in the case that the temperature of the secondary refrigerant flowing through the third intermediate-pressure-side branching channel 67c and the temperature of the secondary refrigerant flowing through the third high-pressure-side branching channel 68c are the same temperature, the temperature of the secondary refrigerant after merging in the air-warming merging point Y is the same temperature as that prior to merging, and is the temperature of the secondary refrigerant for air warming fed toward the radiator 61.

(Processing for Increasing the Heat Amount)

[0187] In the case that the secondary refrigerant-temperature equalization control described above is performed and the temperature equalized by the secondary refrigerant-temperature equalization control are lower than the temperature requested by the radiator 61, the controller 11 performs a control for reducing the flow rate through air-warming pump 63 in order to increase the heat amount.

[0188] The flow speed of the secondary refrigerant flowing through the intermediate-pressure-side branching channel 67 and the flow speed of the secondary refrigerant flowing

through the high-pressure-side branching channel **68** can both be reduced thereby. As a result, the time available for the secondary refrigerant flowing through the intermediate-pressure-side branching channel **67** to receive heat from the primary refrigerant and the time for the secondary refrigerant flowing through the high-pressure-side branching channel **68** to receive heat from the primary refrigerant can both be extended. The temperature of the secondary refrigerant for air warming flowing through the third intermediate-pressure-side branching channel **67c** and the temperature of the secondary refrigerant for air warming flowing through the third high-pressure-side branching channel **68c** can be equalized thereby at the temperature requested by the radiator **61**, and it is possible to adapt to the heat load in the radiator **61**.

(Processing for Reducing the Heat Amount)

[0189] In the case that the secondary refrigerant-temperature equalization control described above is performed and the temperature equalized by the secondary refrigerant-temperature equalization control exceed the temperature requested by the radiator **61**, the controller **11** performs a control for increasing the flow rate through air-warming pump **63** in order to reduce the heat amount.

[0190] The flow speed of the secondary refrigerant flowing through the intermediate-pressure-side branching channel **67** and the flow speed of the secondary refrigerant flowing through the high-pressure-side branching channel **68** can both be increased thereby. As a result, the time available for the secondary refrigerant flowing through the intermediate-pressure-side branching channel **67** to receive heat from the primary refrigerant and the time for the secondary refrigerant flowing through the high-pressure-side branching channel **68** to receive heat from the primary refrigerant can both be shortened. The temperature of the secondary refrigerant for air warming flowing through the third intermediate-pressure-side branching channel **67c** and the temperature of the secondary refrigerant for air warming flowing through the third high-pressure-side branching channel **68c** can be equalized thereby at the temperature requested by the radiator **61**, and it is possible to adapt to the heat load in the radiator **61**.

<1-6> Characteristics of the First Embodiment

[0191] In the heat pump system **1** of the first embodiment, the controller **11** performs a control so that the temperature of the secondary refrigerant flowing through the third intermediate-pressure-side branching channel **67c** and the temperature of the secondary refrigerant flowing through the third high-pressure-side branching channel **68c** are equalized. Here, the secondary refrigerant flowing through the third intermediate-pressure-side branching channel **67c** and the secondary refrigerant flowing through the third high-pressure-side branching channel **68c** both release heat to the lower temperature surroundings and undergo heat loss before arriving at the radiator **61**. However, in the heat pump system **1** of the first embodiment, the temperature of the secondary refrigerant flowing through the third intermediate-pressure-side branching channel **67c** as well as the temperature of the secondary refrigerant flowing through the third high-pressure-side branching channel **68c** can be set to temperature that is not excessively high, and it is possible to minimize the difference from the ambient temperature. Therefore, it is possible to minimize heat loss to the surroundings in relation to the temperature of the secondary refrigerant flowing through

the third intermediate-pressure-side branching channel **67c** as well as the temperature of the secondary refrigerant flowing through the third high-pressure-side branching channel **68c**.

[0192] The controller **11** furthermore performs a control so that the temperature of the secondary refrigerant flowing through the third intermediate-pressure-side branching channel **67c** and the temperature of the secondary refrigerant flowing through the third high-pressure-side branching channel **68c** are equalized with the temperature requested by the radiator **61**. Therefore, the temperature does not need to be adjusted by heating or cooling so that the temperature of the secondary refrigerant for air warming that has merged at the air-warming merging point **Y** becomes the temperature requested by the radiator **61**. It is thereby possible to dispense with a temperature adjustment heater or cooler.

[0193] In the heat pump circuit **10** in the heat pump system **1** of the first embodiment, the primary refrigerant taken in by the high-stage-side compressor **25** is cooled by the secondary refrigerant for air warming during passage through the intermediate-pressure-water heat exchanger **40** and is further cooled by the primary refrigerant flowing in through the injection channel **70**. Accordingly, the density of the primary refrigerant taken in by the high-stage-side compressor **25** can be increased and the efficiency of the heat pump circuit **10** can be improved.

[0194] The heat obtained by the secondary refrigerant for air warming by cooling the primary refrigerant taken in by the high-stage-side compressor **25** can be used for an air-warming heat load in the radiator **61**.

[0195] The temperature of the primary refrigerant flowing through the high-pressure water heat exchanger **50** is in a temperature range capable of heating the secondary refrigerant for air warming, even when the heat required for increasing the temperature of water for hot-water supply to the requested water temperature is obtained from the primary refrigerant flowing through the high-pressure water heat exchanger **50**. Accordingly, the heat of the primary refrigerant flowing through the second high-pressure-water heat exchanger **52**, which is a part of the high-pressure water heat exchanger **50**, can be effectively used for heating the secondary refrigerant for air warming in a range that allows operating efficiency of the heat pump circuit **10** to be optimized. The heat of the primary refrigerant flowing through the high-pressure water heat exchanger **50** can be effectively used while the operating efficiency of the heat pump circuit **10** is optimized.

[0196] In the case that, e.g., the secondary refrigerant for air warming or the water for hot-water supply is to be further warmed in the high-pressure water heat exchanger **50** after having been warmed in the intermediate-pressure-water heat exchanger **40**, the heat in the primary refrigerant flowing through the high-pressure water heat exchanger **50** cannot be sufficiently used in an effective manner because the secondary refrigerant for air warming or the water for hot-water supply that is to flow into the high-pressure water heat exchanger **50** has already been warmed. In other words, enthalpy variation of the primary refrigerant in the heat release step cannot be sufficiently used in terms of a Mollier graph. In the similar case that an attempt is made to heat the secondary refrigerant for air warming or the water for hot-water supply in the intermediate-pressure-water heat exchanger **40** after having been heated in the high-pressure water heat exchanger **50**, the secondary refrigerant for air warming or the water for hot-water supply that is to flow into

the intermediate-pressure-water heat exchanger 40 has already been warmed. Accordingly, there are cases in which the heat of the primary refrigerant flowing through the intermediate-pressure-water heat exchanger 40 cannot be sufficiently used and it is difficult to improve the operating efficiency of a multistage compression-type heat pump circuit 10.

[0197] In contrast, in the heat pump system 1 of the first embodiment, the secondary refrigerant cooled in the radiator 61 is divided in the heat pump circuit 10 and used for being heated in the intermediate-pressure-water heat exchanger 40 while being made to pass through the intermediate-pressure-side branching channel 67 and for being heated in the second high-pressure-water heat exchanger 52 while being made to pass through the high-pressure-side branching channel 68. Secondary refrigerant cooled in the radiator 61 and not yet warmed can be fed to the intermediate-pressure-water heat exchanger 40 and the second high-pressure-water heat exchanger 52. It is thereby possible to sufficiently use the heat of the primary refrigerant flowing through the intermediate-pressure tube 23 in an effective manner while improving the cooling effect of the primary refrigerant taken in by the high-stage-side compressor 25.

<2> Second Embodiment

[0198] A heat pump system 201 of the second embodiment is not provided with the primary bypass 80 (the fourteenth high-pressure tube 27n, the primary bypass expansion valve 5b, and the sixth low-pressure tube 20f) in the heat pump system 1 of the first embodiment, and is a system in which all of the circulating primary refrigerant passes through the primary-refrigerant-to-primary-refrigerant heat exchanger 8, as shown in FIG. 4. The configuration is otherwise the same as the configuration in the first embodiment described above, and a description is therefore omitted.

[0199] It is not only possible to reduce the number of components, but it is also possible to dispense with control of the primary bypass expansion valve 5b, in the case of a service environment in which capacity and efficiency problems are less liable to occur when all of the primary refrigerant circulating through the heat pump circuit 10 is made to undergo heat exchange in the primary-refrigerant-to-primary-refrigerant heat exchanger 8.

<3> Third Embodiment

[0200] A heat pump system 301 of the third embodiment is a system in which the cooling of the primary refrigerant flowing through the intermediate-pressure tube 23 is entirely carried out in the intermediate-pressure-water heat exchanger 40 without injection of primary refrigerant into the intermediate-pressure tube 23, as shown in FIG. 5. In other words, the heat pump system 301 of the third embodiment is a system provided with a 33rd intermediate-pressure tube 323c and a 38th high-pressure tube 327g in place of the economizer heat exchanger 7, the injection channel 70 (injection expansion valve 73, first injection tube 72, second injection tube 74, third injection tube 75, and fourth injection tube 76), the eighth high-pressure tube 27h, the ninth high-pressure tube 27i, the tenth high-pressure tube 27j, the third intermediate-pressure tube 23c, and the fourth intermediate-pressure tube 23d, which are provided to the heat pump system 1 of the first embodiment. The 33rd intermediate-pressure tube 323c connects the second intermediate-pressure tube 23b inside the

intermediate-pressure-water heat exchanger 40 and the intake side of the high-stage-side compressor 25. The 38th high-pressure tube 327g connects the sixth high-pressure tube 27f inside the third high-pressure-water heat exchanger 53 and the fourth high-pressure point I. The configuration is otherwise the same as the configuration in the first embodiment described above, and a description is therefore omitted.

[0201] In the heat pump system 301, it is possible to avoid a state in which the refrigerant taken in by the high-stage-side compressor 25 is cooled more as the wet state increases, and it is possible to minimize the number of components and simplify the circuit configuration.

[0202] With the heat pump system 301, the injection channel 70 is not provided, and the amount of primary refrigerant moving toward the high-pressure water heat exchanger 50 can therefore be increased in a range in which the primary refrigerant taken in by the high-stage-side compressor 25 does not enter a wet state, even if the temperature of the primary refrigerant passing through the intermediate-pressure-water heat exchanger 40 has been excessively reduced by secondary refrigerant-temperature equalization control.

<4> Fourth Embodiment

[0203] The heat pump system 401 of the fourth embodiment is a system in which the branching to the injection channel 70 side is disposed in the downstream side of the economizer heat exchanger 7, as shown in FIG. 6. In other words, the heat pump system 401 of the fourth embodiment is a system, in the heat pump system 1 of the first embodiment, provided with a 43rd high-pressure point 4H in place of the third high-pressure point H, a 47th high-pressure tube 427g in place of the seventh high-pressure tube 27g, a 48th high-pressure tube 427h in place of the eighth high-pressure tube 27h, a 49th high-pressure tube 427i in place of the ninth high-pressure tube 27i, and a 410th high-pressure tube 427j in place of the tenth high-pressure tube 27j. The 43rd high-pressure point 4H is provided on the downstream side of the economizer heat exchanger 7 and in the upstream side of the fourth high-pressure point I, in the flow direction of the primary refrigerant in the heat pump circuit 10, and branches to the injection channel 470. The 47th high-pressure tube 427g connects the sixth high-pressure tube 27f inside the third high-pressure-water heat exchanger 53 and the 48th high-pressure tube 427h inside the economizer heat exchanger 7. The 49th high-pressure tube 427i connects the 43rd high-pressure point 4H and the 48th high-pressure tube 427h inside the economizer heat exchanger 7. The 410th high-pressure tube 427j connects the 43rd high-pressure point 4H and the fourth high-pressure point I. The configuration is otherwise the same as the configuration in the first embodiment described above, and a description is therefore omitted.

[0204] With this heat pump system 401, the cooling effect at the injection merging point D can be improved because the temperature of the primary refrigerant flowing through the injection channel 470 can be reduced in comparison with the primary refrigerant flowing through the injection channel 70 of the heat pump system 1 of the first embodiment.

<5-1> Fifth Embodiment

[0205] The heat pump system 501 of the fifth embodiment is a system that excludes the third high-pressure-water heat exchanger 53 in the heat pump system 1 of the first embodiment, as shown in FIG. 7. In other words, the heat pump

system **501** of the fifth embodiment is a system provided with a 52nd hot-water supply heat pump tube **595b** in place of the second hot-water supply heat pump tube **95b**, third hot-water supply heat pump tube **95c**, and fourth hot-water supply heat pump tube **95d**; and a 55th high-pressure tube **527e** in place of the fifth high-pressure tube **27e**, the sixth high-pressure tube **27f**, and the seventh high-pressure tube **27g** in the heat pump system **1** of the first embodiment. Here, the hot-water supply intermediate temperature sensor **95T** used in the heat pump system **1** of the first embodiment is not required. The 52nd hot-water supply heat pump tube **595b** connects the hot-water supply pump **92** and the upstream-side end section of the fifth hot-water supply heat pump tube **95e** inside the first high-pressure water heat exchanger **51** in the flow direction of the water for hot-water supply. The 55th high-pressure tube **527e** connects the third high-pressure point H and the downstream-side end section of the fourth high-pressure tube **27d** inside the second high-pressure-water heat exchanger **52** in the flow direction of the primary refrigerant. The configuration is otherwise the same as the configuration in the first embodiment described above, and a description is therefore omitted.

[0206] With this heat pump system **501**, the primary refrigerant moving toward the third high-pressure point H is not warmed and the water for hot-water supply is not cooled, even when, e.g., the temperature of the water for hot-water supply stored in the hot-water storage tank **91** has increased and the temperature of the water for hot-water supply detected by the hot-water supply water-intake temperature sensor **94T** is higher than the temperature of the primary refrigerant passing through the outlet of the fourth high-pressure tube **27d** inside the second high-pressure-water heat exchanger **52**. Therefore, operation with good efficiency can be obtained even in a state of low heat load for hot-water supply.

<5-2> Modification of the Fifth Embodiment

(A)

[0207] The heat pump system **501** of the fifth embodiment described above may be a heat pump system **501A** in which the 55th high-pressure tube **527e** described above is used in place of the 47th high-pressure tube **427g** while the injection channel **470** described in the fourth embodiment is used, as shown in FIG. 8.

[0208] In this case, an effect similar to that of the heat pump system **401** of the fourth embodiment can also be obtained.

(B)

[0209] The heat pump system **501** of the fifth embodiment may be a heat pump system **501B** in which the connection to the 55th high-pressure tube **527e** described above is the fourth high-pressure point I while the injection channel **70** described in the third embodiment is eliminated, as shown in FIG. 9.

[0210] In this case, an effect similar to that of the heat pump system **301** of the third embodiment can be further obtained.

(C)

[0211] The heat pump system **501B** of the modification (B) of the fifth embodiment may be a heat pump system **501C** in which the primary bypass **80** is eliminated as described in the second embodiment, as shown in FIG. 10.

[0212] In this case, an effect similar to that of the heat pump system **201** of the second embodiment can be further obtained.

<6-1> Sixth Embodiment

[0213] The heat pump system **601** of the sixth embodiment is a system provided with a gas-liquid separation injection channel **630**, as shown in FIG. 11, in the heat pump system **301** of the third embodiment, which does not have the injection channel **70**. The gas-liquid separation injection channel **630** has a pre-separation gas-liquid tube **631**, a gas-liquid separator **632**, a post-separation liquid tube **633**, a post-separation gas tube **634**, a post-separation gas tube on-off valve **635**, and a gas-liquid separation expansion valve **605**. The pre-separation gas-liquid tube **631** extends from the third low-pressure point M to the gas-phase space in the upper part of the gas-liquid separator **632**. The gas-liquid separator **632** separates the primary refrigerant flowing in from the pre-separation gas-liquid tube **631** into a gas-phase region in the upper space and a liquid-phase region in the lower space. The post-separation liquid tube **633** directs the primary refrigerant present in the liquid-phase region of the gas-liquid separator **632** to the gas-liquid separation expansion valve **605**. The pressure of the passing primary refrigerant is further reduced in the gas-liquid separation expansion valve **605**. The post-separation gas tube **634** directs the primary refrigerant present in the gas-phase region of the gas-liquid separator **632** to the injection merging point D. The post-separation gas tube on-off valve **635** is capable of switching between a state that permits and a state that does not permit the passage of the primary refrigerant in the post-separation gas tube **634**. The configuration is otherwise the same as the configuration in the first embodiment described above, and a description is therefore omitted.

[0214] With the heat pump system **601**, pressure of the primary refrigerant in the expansion valve **5a** and/or the primary bypass expansion valve **5b** is reduced to a pressure that is lower than a critical pressure that is the same as the primary refrigerant flowing through the intermediate-pressure tube **23** to thereby achieve a gas-liquid two-phase state. The primary refrigerant in a liquid state is reduced to the pressure of the primary refrigerant flowing through the low-pressure tube **20** in the gas-liquid separation expansion valve **605**. The post-separation gas tube **634** extends from the gas-phase region of the gas-liquid separator **632**, and the primary refrigerant in a gas state therefore flows to the post-separation gas tube **634** because the primary refrigerant in a liquid state is not liable to become mixed therein. The primary refrigerant taken in by the high-stage-side compressor **25** is thereby made less likely to enter a wet state after merging with the primary refrigerant flowing through the intermediate-pressure tube **23** at the injection merging point D. Liquid compression in the high-stage-side compressor **25** can thereby be prevented while the refrigerant density taken in by the high-stage-side compressor **25** is increased and efficiency is improved. In the depressurization of the primary refrigerant in the expansion valve **5a**, the pressure is not reduced to the pressure of the primary refrigerant flowing through the low-pressure tube **20** and the pressure is only reduced to nearly the pressure of the primary refrigerant flowing through the intermediate-pressure tube **23**. Therefore, it is possible to minimize the occurrence of liquid compression in the high-stage-side compressor **25**, which can occur when the temperature of the primary refrigerant flowing through the post-separation gas tube **634** is

excessively reduced. Also, the amount of primary refrigerant moving toward the high-pressure water heat exchanger 50 can be increased in a range in which the primary refrigerant taken in by the high-stage-side compressor 25 does not enter a wet state, even when the temperature of the primary refrigerant passing through the intermediate-pressure-water heat exchanger 40 has dropped excessively due to secondary refrigerant-temperature equalization control.

<6-2> Modification of the Sixth Embodiment

(A)

[0215] The heat pump system 601 of the sixth embodiment described above may be a heat pump system 601A which does not have the third high-pressure-water heat exchanger 53 as described in the fifth embodiment, as shown in FIG. 12. The configuration is otherwise the same as the configuration in the first embodiment described above, and a description is therefore omitted.

<7> Seventh Embodiment

[0216] A heat pump system 701 of the seventh embodiment may be a system in which the position of the injection merging point D in the heat pump system 1 of the first embodiment is an injection merging point 7D, which is located at a midway point in the first intermediate-pressure tube 23a for connecting the discharge side of the low-stage-side compressor 21 and the downstream-side end section of the second intermediate-pressure tube 23b inside the intermediate-pressure-water heat exchanger 40, as shown in FIG. 13. The configuration is otherwise the same as the configuration in the first embodiment described above, and a description is therefore omitted.

[0217] In the heat pump system 701, there are cases in which the discharge refrigerant temperature of the low-stage-side compressor 21 becomes excessively high for the secondary refrigerant for air warming heated in the intermediate-pressure-water heat exchanger 40, in the case that, e.g., the low-stage-side compressor 21 is operated to increase drive efficiency using the same compression ratio as the compression ratio of the high-stage-side compressor 25 while the compression ratio of the high-stage-side compressor 25 is increased so that a target temperature can be obtained as the discharge refrigerant temperature of the high-stage-side compressor 25. Even in such cases, it is possible to keep the temperature of the secondary refrigerant for air warming from becoming too high by providing the injection merging point 7D at a midway point in the first intermediate-pressure tube 23a.

[0218] In the heat pump system 701 as well, the temperature and pressure of the primary refrigerant to be taken in by the high-stage-side compressor 25 after the primary refrigerant passing in through the injection channel 70 has merged in the injection merging point 7D and after having passed through the intermediate-pressure-water heat exchanger 40 are values detected by the high-stage intake pressure sensor 24P and the high-stage intake temperature sensor 24T, and the controller 11 can ascertain the values and perform control for inhibiting the primary refrigerant taken in by the high-stage-side compressor 25 from entering a wet state.

<8> Eighth Embodiment

[0219] A heat pump system 801 of the eighth embodiment is a system in which the order of the economizer heat exchanger 7 and the primary-refrigerant-to-primary-refrigerant

heat exchanger 8 in the heat pump system 1 of the first embodiment is reversed, as shown in FIG. 14. In other words, the heat pump system 801 of the eighth embodiment is a system in which an 83rd intermediate-pressure point 8H in the downstream side of the third low-pressure point M is provided in place of the third high-pressure point H in the heat pump system 1 of the first embodiment, and an injection channel 870 branches from the 83rd intermediate-pressure point 8H. An 810th high-pressure tube 827j connects the fourth high-pressure point I and the downstream-side end section of the sixth high-pressure tube 27f inside the third high-pressure-water heat exchanger 53. An 87th high-pressure tube 827g connects the third low-pressure point M and the 83rd intermediate-pressure point 8H. An 88th high-pressure tube 827h connects the 83rd intermediate-pressure point 8H and the upstream-side end section of an 89th high-pressure tube 827i inside the economizer heat exchanger 7. The configuration is otherwise the same as the configuration in the first embodiment described above, and a description is therefore omitted.

[0220] In this heat pump system 801, the primary refrigerant taken by the low-stage-side compressor 21 can be warmed in the primary-refrigerant-to-primary-refrigerant heat exchanger 8 by the relatively warm primary refrigerant before it is cooled in the economizer heat exchanger 7. The amount of primary refrigerant moving toward the high-pressure water heat exchanger 50 can be thereby increased in a range in which the primary refrigerant to be taken in by the high-stage-side compressor 25 does not enter a wet state, even when the temperature of the primary refrigerant passing through the intermediate-pressure-water heat exchanger 40 becomes excessively reduced by secondary refrigerant-temperature equalization control.

<9> Ninth Embodiment

[0221] A heat pump system 901 of the ninth embodiment is a system for warming water for hot-water supply in the second high-pressure-water heat exchanger 52 as well in the heat pump system 1 of the first embodiment, as shown in FIG. 15. In other words, the heat pump system 901 of the ninth embodiment is a system provided with a 95th upstream connection tube 995x, a 95th hot-water supply heat pump tube 995d, and a 95th downstream connection tube 995y in place of the fourth hot-water supply heat pump tube 95d in the heat pump system 1 of the first embodiment; and provided with an upstream-connection temperature sensor 95Tx for detecting the temperature of the water for hot-water supply passing through the 95th upstream connection tube 995x, and a downstream-connection temperature sensor 95Ty for detecting the temperature of the water for hot-water supply passing through the 95th downstream connection tube 995y. The configuration is otherwise the same as the configuration in the first embodiment described above, and a description is therefore omitted.

[0222] In this heat pump system 901, the loss of heat released from the fourth high-pressure tube 27d can be minimized and effectively used in the second high-pressure-water heat exchanger 52 because, e.g., the water for hot-water supply flowing through the 95th hot-water supply heat pump tube 995d can absorb the heat that cannot be absorbed by the secondary refrigerant for air warming flowing through the second high-pressure-side branching channel 68b among the heat released from the fourth high-pressure tube 27d. Also, the size of the heat exchanger required for heating the water

for hot-water supply to a required water temperature can be made compact because a portion is provided in which both the secondary refrigerant for air warming and the water for hot-water supply simultaneously receive the heat of the primary refrigerant.

<10> Tenth Embodiment

[0223] A heat pump system 1x of the tenth embodiment is a system configured so that the hot-water supply circuit 90 in the heat pump system 1 of the first embodiment is removed, as shown in FIG. 16. In other words, the heat pump system 1x of the tenth embodiment is a system in which the first high-pressure water heat exchanger 51, the third high-pressure-water heat exchanger 53, and the hot-water supply circuit 90 in the heat pump system 1 of the first embodiment are removed; a fourteenth upstream high-pressure tube 127a is provided in place of the first high-pressure tube 27a, the second high-pressure tube 27b, and the third high-pressure tube 27c; and a fourteenth downstream high-pressure tube 127e is provided in place of the fifth high-pressure tube 27e, the sixth high-pressure tube 27f, and the seventh high-pressure tube 27g. The fourteenth upstream high-pressure tube 127a connects the discharge side of the high-stage-side compressor 25 and the upstream-side end section of the fourth high-pressure tube 27d inside the second high-pressure-water heat exchanger 52. The fourteenth downstream high-pressure tube 127e connects the third high-pressure point H and the downstream-side end section of the fourth high-pressure tube 27d inside the second high-pressure-water heat exchanger 52. The configuration is otherwise the same as the configuration in the first embodiment described above, and a description is therefore omitted.

[0224] With this heat pump system 1x, it is possible to obtain the same effects as those of the first embodiment, even when the hot-water supply circuit 90 is not provided.

<11-1> Eleventh Embodiment

[0225] A heat pump system 2x of the eleventh embodiment is a system in which the water for hot-water supply flowing through the hot-water supply circuit 90 in the heat pump system 1 of the first embodiment undergoes heat exchange with the primary refrigerant in the intermediate-pressure-water heat exchanger 40 as well as the high-pressure water heat exchanger 50 side in the same manner as the secondary refrigerant for air warming, as shown in FIG. 17. In other words, the heat pump system 2x of the eleventh embodiment is provided with a second intermediate-pressure-water heat exchanger 153 for performing heat exchange between the primary refrigerant passing through the intermediate-pressure-water heat exchanger 40 and the water for hot-water supply. A second branching hot-water supply heat pump tube 195b branches away at a midway point of the second hot-water supply heat pump tube 95b and then extends to the downstream-side end section of the second intermediate-pressure-water heat exchanger 153. The second intermediate-pressure-water heat exchanger 153 causes heat exchange to be performed between the water for hot-water supply flowing into a third branching hot-water supply heat pump tube 195c via the second branching hot-water supply heat pump tube 195b and the primary refrigerant flowing into the eleventh intermediate-pressure tube 123c, which is a part of the third intermediate-pressure tube 23c, after having passed through the intermediate-pressure-water heat exchanger 40.

[0226] The water for hot-water supply which has passed through the third branching hot-water supply heat pump tube 195c inside the second intermediate-pressure-water heat exchanger 153 flows to a branching hot-water supply mixing valve 193 via a fourth branching hot-water supply heat pump tube 195d, and merges with the water for hot-water supply that has flowed in through the fourth hot-water supply heat pump tube 95d. The water for hot-water supply which has merged in the branching hot-water supply mixing valve 193 flows into the fifth hot-water supply heat pump tube 95e inside the first high-pressure water heat exchanger 51 via a merging hot-water supply communication tube 196. The configuration is otherwise the same as the configuration in the first embodiment described above, and a description is therefore omitted.

[0227] With this heat pump system 2x, in the case that, e.g., the temperature of the water for hot-water supply flowing out from the hot-water storage tank 91 to the heat pump circuit 10 side is at a normal temperature, which is the temperature of city water, there may be cases in which it is more efficient to cool the water for hot-water supply in a range in which liquid-compression does not occur in the high-stage-side compressor 25, even with the primary refrigerant that has been cooled while passing through the second intermediate-pressure tube 23b inside the intermediate-pressure-water heat exchanger 40. In such a case, in the heat pump system 2x of the eleventh embodiment, cold water for hot-water supply can be heated using the heat of the primary refrigerant flowing not only in the high-pressure water heat exchanger 50 side, but also between the downstream side of the intermediate-pressure-water heat exchanger 40 and the intake side of the high-stage-side compressor 25.

[0228] With such a configuration, the controller 11 furthermore controls the branching hot-water supply mixing valve 193 to thereby adjust the flow rate of the fourth branching hot-water supply heat pump tube 195d and the flow rate of the fourth hot-water supply heat pump tube 95d, whereby degradation of cycle efficiency of the heat pump circuit 10 can be minimized, even in the case that the cycle efficiency of the heat pump circuit 10 is slightly degraded by the secondary refrigerant-temperature equalization control described above.

[0229] For example, the controller 11 can control the branching hot-water supply mixing valve 193 to thereby increase the flow rate through the fourth branching hot-water supply heat pump tube 195d and to minimize degradation of the cycle efficiency of the heat pump circuit 10, in the case that the cycle efficiency of the heat pump circuit 10 is slightly degraded by a reduction in the flow rate through the intermediate-pressure-side branching channel 67 of the air-warming circuit 60 by the secondary refrigerant-temperature equalization control described above.

<11-2> Modification of the Eleventh Embodiment

(A)

[0230] In the heat pump system 2x of the eleventh embodiment, an example was described in which not only is heat exchange (intermediate-pressure-water heat exchanger 40) performed with the secondary refrigerant for air warming, but heat exchange (second intermediate-pressure-water heat exchanger 153) is also performed with the water for hot-water supply, in the intermediate-pressure tube 23 through which

the primary refrigerant is flowing from the low-stage-side compressor 21 toward the high-stage-side compressor 25.

[0231] However, the present invention is not limited thereto, and it is also possible to use a heat pump system capable of the following heat exchange in a range that does not depart from the spirit of the present invention.

[0232] For example, heat exchange can be performed in three locations so that heat exchange is performed between the water for hot-water supply, the secondary refrigerant for air warming, and the primary refrigerant in the high-pressure water heat exchanger 50 of the first embodiment in the intermediate-pressure tube 23 in which the primary refrigerant is flowing from the low-stage-side compressor 21 toward the high-stage-side compressor 25. In this case as well, heat exchange between the water for hot-water supply and the primary refrigerant flowing through the intermediate-pressure tube 23 is preferably performed in two separate locations, i.e., in the upstream side and the downstream side, where heat exchange is performed between the primary refrigerant and the secondary refrigerant for air warming, in the same manner as the high-pressure water heat exchanger 50.

(B)

[0233] The water for hot-water supply may undergo heat exchange in the intermediate-pressure tube 23 in which the primary refrigerant flows from the low-stage-side compressor 21 toward the high-stage-side compressor 25 without being made to undergo heat exchange with the primary refrigerant in the high-pressure water heat exchanger 50.

<12> Twelfth Embodiment

[0234] A heat pump system 3x of the twelfth embodiment is a system in which a bypass channel is provided in the air-warming circuit 60 in the heat pump system 1 of the first embodiment, as shown in FIG. 18. In other words, the heat pump system 3x of the twelfth embodiment is a system in which an air-warming bypass channel 69 for connecting the air-warming merging point Y and the hot-water supply merging point Z at a midway point of the air-warming-return tube 66 is further provided in the air-warming circuit 60 in the heat pump system 1 of the first embodiment, and a twelfth air-warming mixing valve 164 is provided in place of the air-warming mixing valve 64 in the first embodiment. In the twelfth air-warming mixing valve 164, an adjustment is made by instruction from the controller 11 to the mixing ratio between the cool secondary refrigerant for air warming which flows in from the air-warming bypass circuit 69 and which has just finished releasing heat in the radiator 61, the warmed secondary refrigerant for air warming flowing in via the intermediate-pressure-side branching channel 67, and the warmed secondary refrigerant for air warming flowing in via the high-pressure-side branching channel 68. The configuration is otherwise the same as the configuration in the first embodiment described above, and a description is therefore omitted.

[0235] In the heat pump system 1 of the first embodiment described above, there are cases in which a heat amount that exceeds the heat amount requested by the radiator 61 flows to the radiator 61, even when processing has been performed to reduce the heat amount while secondary refrigerant-temperature equalization control as described above has been performed. With this heat pump system 3x of the twelfth embodiment, the controller 11 operates the twelfth air-warming

mixing valve 164 and is capable of adjusting the flow rate of the secondary refrigerant for air warming flowing in through the air-warming bypass channel 69 toward the air-warming merging point Y when the heat amount to the radiator 61 is on the verge of becoming excessive as described above. The secondary refrigerant having a temperature that exceeds the temperature requested by the radiator 61 and the secondary refrigerant having a temperature that is less than that requested by the radiator 61 after heat release has ended in the radiator 61 are mixed. The controller 11 adjusts these mixing ratios in the twelfth air-warming mixing valve 164, whereby the temperature of the secondary refrigerant after mixing is adjusted to the temperature requested by the radiator 61.

[0236] The secondary refrigerant having the temperature requested by the radiator 61 can thereby be fed to the radiator 61 while the occurrence of liquid compression in the high-stage-side compressor 25 is minimized.

<13> Thirteenth Embodiment

[0237] A heat pump system 4x of the thirteenth embodiment is a system configured so that the economizer heat exchanger 7 and the third high-pressure point H are positioned between the third low-pressure point and the portion that branches the flow toward the primary-refrigerant-to-primary-refrigerant heat exchanger 8 and the flow toward the primary bypass 80, as shown in FIG. 19. In other words, the heat pump system 4x of the thirteenth embodiment is a system in which the fourth high-pressure point I of the heat pump system 1 of the first embodiment is modified to be a thirteenth high-pressure point 131 that is on the upstream side of the third high-pressure point H and is on the downstream side of the third high-pressure-water heat exchanger 53, and in which a thirteenth primary bypass 80x and a thirteenth injection channel 70x are provided. A seventh high-pressure tube 127g connects the thirteenth high-pressure point 131 and the downstream-side end section of the sixth high-pressure tube 27f inside the third high-pressure-water heat exchanger 53. A bypass upstream-economizer high-pressure tube 127n connects the thirteenth high-pressure point 131 and the third high-pressure point H. A bypass downstream-economizer high-pressure tube 127j connects the primary bypass expansion valve 5b and the downstream-side end section of the ninth high-pressure tube 27i inside the economizer heat exchanger 7. The configuration is otherwise the same as the configuration in the first embodiment described above, and a description is therefore omitted.

[0238] With this heat pump system 4x, the primary refrigerant moving toward the expansion valve 5a is divided into a channel for cooling by the economizer heat exchanger 7 and a channel for cooling by the primary-refrigerant-to-primary-refrigerant heat exchanger 8, and it is therefore possible to adjust the amount of primary refrigerant that is cooled by either of the channels.

<14> Applicable Modifications of the Embodiments Described Above

[0239] Heat pump systems were described in detail above in the first to thirteenth embodiments. However, the present invention is not limited thereto; modes such as those described below of the heat pump systems of the embodiments are also included in the present invention in a range that does not depart from the spirit of the invention.

<14-1>

[0240] In the embodiments described above, examples were described for the case in which carbon dioxide is used as the primary refrigerant.

[0241] However, ethylene, ethane, and/or nitrogen oxide or the like, which are refrigerants other than carbon dioxide, may also be used as the primary refrigerant in any of the embodiments described above. In this case, the refrigerant to be used is preferably one that can be used with the discharge refrigerant pressure of the high-stage-side compressor 25 above critical pressure, and that can minimize the drive force of the compressors.

<14-2>

[0242] In the embodiments described above, examples were described for the case in which water circulates as the secondary refrigerant in the air-warming circuit 60.

[0243] However, in any of the embodiments described above, brine or the like may be used as another heat medium with no limitation being given to water as the secondary refrigerant.

<14-3>

[0244] In the embodiments described above, examples were described for the case in which the low-stage-side compressor 21 and the high-stage-side compressor 25 are provided.

[0245] However, in any of the embodiments described above, it is possible to provide a so-called single-shaft two-stage or a single-shaft multistage-type compression mechanism in which a shared drive shaft is used in the low-stage-side compressor 21 and the high-stage-side compressor 25. In this case, it is possible to increase drive efficiency by providing a 180-degree phase difference in the compression mechanisms.

<14-4>

[0246] In the embodiments described above, examples were described for the case in which the low-stage-side compressor 21 and the high-stage-side compressor 25 are connected in series.

[0247] However, in any of the embodiments described above, it is also possible to use a mode in which three or more compression mechanisms are connected in series. In such a case, heat load processing may be performed using the heat of the primary refrigerant flowing between each compression mechanism. Also, if the compression mechanisms are provided with two or more series-connection circuits, another compression mechanism may be further provided in parallel or in series.

<14-5>

[0248] In the embodiments described above, examples were described for the case in which control is performed to bring the temperature of the secondary refrigerant flowing through the intermediate-pressure-side branching channel 67 and high-pressure-side branching channel 68 of the air-warming circuit 60 into conformity with the temperature requested by the radiator 61.

[0249] However, in any of the embodiments described above, it is also possible to make the optimization of the cycle efficiency in the heat pump circuit 10 an absolute priority over feeding the heat amount requested by the radiator 61. In this case, there may be situations in which the heat amount fed to the radiator 61 is insufficient for maintaining optimal cycle efficiency of the heat pump circuit 10. In this case, it is possible to use a heat pump system 5x in which an external heat source section 60A for heating the passing secondary refrigerant for air warming is provided in the downstream

side of air-warming circuit 60, including the third high-pressure-side branching channel 68c or the third intermediate-pressure-side branching channel 67c from downstream side to prior to arriving at the radiator 61, as shown in FIG. 20. In this case, it is possible to adapt to the air-warming heat load while the cycle efficiency of the heat pump circuit 10 is kept in an optimal state, even when a situation occurs in which it is not possible to adapt to the air-warming heat load in order to keep the cycle efficiency of the heat pump circuit 10 optimal due to change in the environment in which the evaporator 4 is disposed, change in the air-warming load, or change in the hot-water supply load. A heat-supply section similar to the external heat source section 60A may be provided only to the hot-water supply circuit 90, or may be provided to both the air-warming circuit 60 and the hot-water supply circuit 90.

[0250] There may be cases in which an excessive heat amount is fed to the radiator 61 in order to keep the cycle efficiency of the heat pump circuit 10 optimal. In this case, it is possible to use a heat pump system 6x in which an external cooling source section 60B for cooling the passing secondary refrigerant for air warming is provided in the downstream side of air-warming circuit 60, including the third high-pressure-side branching channel 68c or the third intermediate-pressure-side branching channel 67c from downstream side to prior to arriving at the radiator 61, as shown in FIG. 21. The external cooling source section 60B may be one in which a portion of the water supply tube 94 through which external normal-temperature city water is flowing is bypassed by a water supply branching valve 94B and a water supply branching tube 194, and the normal-temperature city water and the secondary refrigerant for air warming flowing through the air-warming feed tube 65 are made to undergo heat exchange to thereby cool the secondary refrigerant for air warming flowing through the air-warming feed tube 65. In this case, it is possible to adapt to the air-warming heat load while the cycle efficiency of the heat pump circuit 10 is kept in an optimal state, even when a situation occurs in which it is not possible to adapt to the air-warming heat load in order to keep the cycle efficiency of the heat pump circuit 10 optimal due to change in the environment in which the evaporator 4 is disposed, change in the air-warming load, or change in the hot-water supply load. In the case that the water supply branching valve 94B is used, the heat provided in excess by the heat pump circuit 10 to the secondary refrigerant of the air-warming circuit 60 is recovered as heat for hot-water supply, whereby the efficiency of the heat pump system can also be increased. A heat-supply section similar to the external heat source section 60B may be provided only to the hot-water supply circuit 90, or may be provided to both the air-warming circuit 60 and the hot-water supply circuit 90.

<14-6>

[0251] In the embodiments described above, examples were described for the case in which there is no particular limitation to the relationship between the temperature requested by the radiator 61 of the air-warming circuit 60 and the temperature requested for the water for hot-water supply flowing through the sixth hot-water supply heat pump tube 95f and then into the hot-water storage tank 91 in the hot-water supply circuit 90, and the temperature of the primary refrigerant flowing through the intermediate-pressure-water heat exchanger 40 and/or the high-pressure water heat exchanger 50 of the heat pump circuit 10.

[0252] However, in any of the embodiments described above, it is possible to improve the cycle efficiency of the heat

pump circuit 10 under the presumed condition that the controller 11 controls the opening degree of the expansion valve 5a, the drive frequency of the low-stage-side compressor 21, the drive frequency of the high-stage-side compressor 25, and the like so that the temperature of the primary refrigerant flowing through the intermediate-pressure-water heat exchanger 40 exceeds the temperature requested by the radiator 61 of the air-warming circuit 60. In this case, the air-warming circuit 60 can produce secondary refrigerant at the temperature requested by the radiator 61 by using only the heat obtained by the secondary refrigerant flowing through the intermediate-pressure-side branching channel 67 side, which is the intermediate-pressure-water heat exchanger 40 side.

<14-7>

[0253] In the embodiments described above, examples were described for the case in which the compression ratio of the low-stage-side compressor 21 and the compression ratio of the high-stage-side compressor 25 are made equal in order to increase the cycle efficiency of the heat pump circuit 10.

[0254] However, in any of the embodiments described above, there is no limitation to the case in which the compression ratio of the low-stage-side compressor 21 and the compression ratio of the high-stage-side compressor 25 are made to be the same, and, for example, it is also possible to perform control so that the difference between the two compression ratios is reduced.

<14-8>

[0255] For example, in the case that the secondary refrigerant-temperature equalization control described in the embodiments above is carried out, the controller 11 performs a control to increase the flow rate through the air-warming pump 63 so as to shorten the time allowed for heat exchange in the case that the temperatures to be equalized exceed the temperature requested by the radiator 61. However, when control is performed to increase the flow rate through the air-warming pump 63 in this manner, the degree of superheat of the primary refrigerant taken in by the high-stage-side compressor 25 is liable to be reduced and a wet state is liable to form because the primary refrigerant in the intermediate-pressure-water heat exchanger 40 is cooled further.

[0256] In such a case, the controller 11 may perform low-stage intake degree-of-superheat control for increasing the degree of superheat of the primary refrigerant taken in by the low-stage-side compressor 21 without, e.g., modifying the target discharge temperature of the low-stage-side compressor 21 and without modifying the target discharge temperature of the high-stage-side compressor 25.

[0257] For example, the cycle of the heat pump circuit 10 is carried out and the flow rate through the air-warming pump 63 is increased, as shown by the dotted lines in the Mollier graph of FIG. 22. Here, the controller 11 increases the degree of superheat of the primary refrigerant taken in by the low-stage-side compressor 21 without modifying the target discharge temperature of the low-stage-side compressor 21 and without modifying the target discharge temperature of the high-stage-side compressor 25 by performing low-stage intake degree-of-superheat control. The cycle of the heat pump circuit 10 is carried out as shown by the solid lines in the Mollier graph of FIG. 22. Here, in a comparison of the cycle of the dotted lines and the cycle of the solid lines in relation to the point at which the high-stage-side compressor 25 takes in the primary refrigerant, the cycle of the solid lines moves in the direction away from the saturated vapor line, and the degree of superheat

increases in the Mollier graph of FIG. 22. The state of the primary refrigerant taken in by the high-stage-side compressor 25 is a state in which the degree of superheat is increased in progression away from the saturated vapor line, even when the degree of superheat of the primary refrigerant taken in by the high-stage-side compressor 25 is reduced by an increase in the air-warming load or by another change in the ambient conditions, albeit with a slight reduction in the intake refrigerant density of the low-stage-side compressor 21. Liquid compression in the high-stage-side compressor 25 is therefore less liable to occur. The target discharge temperature of the low-stage-side compressor 21 and the target discharge temperature of the high-stage-side compressor 25 are not modified even when the cycle shown by the solid lines in the Mollier graph of the FIG. 22 is carried out. Therefore, heating of the secondary refrigerant for air warming by heat exchange in the intermediate-pressure-water heat exchanger 40 and heating of the secondary refrigerant for air warming by heat exchange in the high-pressure water heat exchanger 50 can be sufficiently carried out. The efficiency of the heat pump circuit 10 can be improved because the compression ratio of the low-stage-side compressor 21 and the compression ratio of the high-stage-side compressor 25 can both be reduced.

[0258] The controller 11 controls the opening degree of the primary bypass expansion valve 5b in the heat pump circuit 10 having the primary bypass 80 and the primary bypass expansion valve 5b among the heat pump systems described in the embodiments and modifications above, making it possible for the above-described low-stage intake degree-of-superheat control to adjust the amount of heat exchange in the primary-refrigerant-to-primary-refrigerant heat exchanger 8. Thus, the degree of superheat of the primary refrigerant taken in by the low-stage-side compressor 21 can be adjusted.

<14-9>

[0259] In the embodiments described above, examples were described for cases in which the target discharge temperature of the low-stage-side compressor 21 and the target discharge temperature of the high-stage-side compressor 25 are the same.

[0260] However, in any of the embodiments described above, the controller 11 may control the drive frequency of the low-stage-side compressor 21, the drive frequency of the high-stage-side compressor 25, the opening degree of the expansion valve 5a, and other factors so that the target discharge temperature of the low-stage-side compressor 21 and the target discharge temperature of the high-stage-side compressor 25 are different. In this case, low-stage discharge temperature reduction control can be performed in order to reduce the target discharge temperature of the low-stage-side compressor 21.

[0261] For example, the cycle of the heat pump circuit 10 is carried out and the flow rate through the air-warming pump 63 is increased, as shown by the dotted lines in the Mollier graph of FIG. 23. Here, the controller 11 performs a control for reducing low-stage discharge temperature to thereby reduce the target discharge temperature of the low-stage-side compressor 21 and to increase the flow rate of the secondary refrigerant for air warming flowing through the second high-pressure-water heat exchanger 52 while the flow rate of the secondary refrigerant for air warming flowing through the intermediate-pressure-water heat exchanger 40 is reduced without modification of the target discharge temperature of the high-stage-side compressor 25. Here, the target discharge temperature of the low-stage-side compressor 21 is set to,

e.g., 65° C. so as to not become equal to or less than the temperature requested by the radiator 61 of the air-warming circuit 60. The cycle of the heat pump circuit 10 is thereby carried out as shown by the solid lines in the Mollier graph of FIG. 23. When a comparison is made of the dotted-lines cycle and the solid-lines cycle in relation to the point at which the high-stage-side compressor 25 takes in the primary refrigerant, the solid-lines cycle moves in the direction away from the saturated vapor line, and the degree of superheat increases in the Mollier graph of FIG. 23. Liquid compression in the high-stage-side compressor 25 is thereby made less likely to occur because the state of the primary refrigerant taken in by the high-stage-side compressor 25 is one that moves away from the saturated vapor line, whereby the degree of superheat increases, even when the flow rate through the air-warming pump 63 is increased and the degree of superheat of the primary refrigerant taken in by the high-stage-side compressor 25 is reduced. The target discharge temperature of the high-stage-side compressor 25 is not modified even when the cycle shown by the solid lines in the Mollier graph of FIG. 23 is carried out in this manner. Although the target discharge temperature of the low-stage-side compressor 21 is reduced, it is possible to maintain a state in which it is possible to adapt to the heat load because the flow rate of the secondary refrigerant for air warming passing through the intermediate-pressure-water heat exchanger 40 is similarly reduced. Also, the efficiency of the heat pump circuit 10 can be improved because the compression ratio of the low-stage-side compressor 21 and the compression ratio of the high-stage-side compressor 25 can both be reduced.

[0262] For example, the controller 11 controls the opening degree of the expansion valve 5a, the drive frequency of the low-stage-side compressor 21, the drive frequency of the high-stage-side compressor 25, and other factors, whereby the low-stage discharge temperature reduction control described above can be carried out.

<14-10>

[0263] In the embodiments described above, examples were described for cases in which the conditions of the ambient temperature environment of the radiator 61 in which the heat pump system is used are not particularly limited.

[0264] However, in any of the embodiments described above, it is possible to provide a limiting condition that the temperature of the secondary refrigerant, which has released heat in the radiator 61, be in a temperature range between the critical temperature of the carbon dioxide as the primary refrigerant and a temperature about five degrees lower than the critical temperature, as the service environment conditions of the heat pump system.

[0265] In the case that the heat pump system is used under such service conditions, the secondary refrigerant is used for heat loads having a temperature that is less than the critical temperature of carbon dioxide as the primary refrigerant. Therefore, heat exchange in the high-pressure water heat exchanger 50 can be carried out between the primary refrigerant in a state exceeding the critical pressure and the secondary refrigerant having a temperature that is less than the critical temperature; and heat release can be carried out in an area in which the slope of the isotherm of the primary refrigerant is smooth on a Mollier graph. It is therefore possible to perform operation that increases the enthalpy difference between the start and end of the primary refrigerant heat release step.

<14-11>

[0266] In the embodiments described above, examples were described for a system in which the controller 11 controls the air-warming mixing valve 64 and the flow rate through the air-warming pump 63 on the basis of the temperatures detected by the intermediate-pressure-side branching temperature sensor 67T and the high-pressure-side branching temperature sensor 68T, and it is not required that the flow rate of the secondary refrigerant in the third intermediate-pressure-side branching channel 67c and third high-pressure-side branching channel 68c of the air-warming circuit 60 be ascertained.

[0267] However, in any of the embodiments described above, it is possible to use a heat pump system 7x in which an intermediate-pressure-side branching channel flow rate meter 67Q for ascertaining the flow rate of the secondary refrigerant for air warming flowing through the intermediate-pressure-side branching channel 67, and a high-pressure-side branching channel flow rate meter 68Q for ascertaining the flow rate of the secondary refrigerant for air warming flowing through the high-pressure-side branching channel 68, in place of the intermediate-pressure-side branching temperature sensor 67T and the high-pressure-side branching temperature sensor 68T, are provided respectively, as shown in FIG. 24.

[0268] In this heat pump system 7x, the controller 11 controls the flow rate through the air-warming mixing valve 64 and/or the air-warming pump 63 on the basis of the flow rate ascertained by the intermediate-pressure-side branching channel flow rate meter 67Q and the flow rate ascertained by the high-pressure-side branching channel flow rate meter 68Q, so that the difference between the temperature of the secondary refrigerant for air warming flowing through the third intermediate-pressure-side branching channel 67c and the temperature of the secondary refrigerant for air warming flowing through the third high-pressure-side branching channel 68c is reduced. The controller 11 may perform control so that the temperature of the secondary refrigerant for air warming flowing through the third intermediate-pressure-side branching channel 67c and the temperature of the secondary refrigerant for air warming flowing through the third high-pressure-side branching channel 68c are the same temperature.

[0269] The controller 11 ascertains the temperature of the primary refrigerant flowing through the intermediate-pressure-water heat exchanger 40 by using the intermediate-pressure temperature sensor 23T, and ascertains the flow rate of the primary refrigerant flowing through the intermediate-pressure-water heat exchanger 40 by drive frequency of the low-stage-side compressor 21, the temperature detected by the intermediate-pressure temperature sensor 23T, and the pressure detected by the high-stage intake pressure sensor 24P. The controller 11 ascertains the temperature of the secondary refrigerant for air warming passing through the first intermediate-pressure-side branching channel 67a by using the temperature detecting by an air-warming return temperature sensor 66T. The controller 11 furthermore ascertains the flow rate of the secondary refrigerant for air warming flowing through the intermediate-pressure-side branching channel 67 by using the intermediate-pressure-side branching channel flow rate meter 67Q. The controller 11 thereby calculates the heat amount obtained by the secondary refrigerant for air warming and calculates a predicted value as the temperature of the secondary refrigerant for air warming passing through the third intermediate-pressure-side branching channel 67c,

on the basis of the temperature difference and the flow rates of the primary refrigerant and secondary refrigerant for air warming in the intermediate-pressure-water heat exchanger 40.

[0270] The controller 11 ascertains the temperature and flow rate of the primary refrigerant flowing through the second high-pressure-water heat exchanger 52 on the basis of the high-pressure temperature sensor 27T, the high-pressure pressure sensor 27P, the drive frequency or other parameters of the high-stage-side compressor 25, the hot-water supply intermediate temperature sensor 95T, the flow rate through the hot-water supply pump 92, and other factors. The controller 11 ascertains the temperature of the secondary refrigerant for air warming passing through the first high-pressure-side branching channel 68a by using the temperature detected by the air-warming return temperature sensor 66T. The controller 11 furthermore ascertains the flow rate of the secondary refrigerant flowing through the high-pressure-side branching channel 68 by using the high-pressure-side branching channel flow rate meter 68Q. The controller 11 thereby calculates the heat amount obtained by the secondary refrigerant for air warming and calculates a predicted value as the temperature of the secondary refrigerant for air warming passing through the third high-pressure-side branching channel 68c, on the basis of the temperature difference and the flow rates of the primary refrigerant and the secondary refrigerant for air warming in the second high-pressure-water heat exchanger 52.

[0271] The controller 11 controls the air-warming mixing valve 64 and/or the air-warming pump 63 so that the difference between the temperature of the secondary refrigerant for air warming passing through the third intermediate-pressure-side branching channel 67c and the temperature of the secondary refrigerant for air warming passing through the third high-pressure-side branching channel 68c thus calculated in the manner described above is reduced. The specific details of control using the temperature of the secondary refrigerant for air warming that passes through the third intermediate-pressure-side branching channel 67c and the temperature of the secondary refrigerant for air warming that passes through the third high-pressure-side branching channel 68c thus calculated herein are the same as the details described in the embodiments above.

[0272] In this manner, it is possible to reduce the difference between the temperature of the secondary refrigerant for air warming that passes through the third intermediate-pressure-side branching channel 67c and the temperature of the secondary refrigerant for air warming that passes through the third high-pressure-side branching channel 68c, even with the heat pump system 7x in which the intermediate-pressure-side branching temperature sensor 67T and the high-pressure-side branching temperature sensor 68T are not provided.

<14-12>

[0273] It is also possible to use a heat pump system 8x provided with a feed tube flow rate meter 65Q in place of the high-pressure-side branching channel flow rate meter 68Q provided in the manner described above in Modification <14-11>, as shown in FIG. 25.

[0274] The feed tube flow rate meter 65Q is capable of ascertaining the flow rate of the secondary refrigerant for air warming that passes through the air-warming feed tube 65. The flow rate of the secondary refrigerant for air warming that flows through the high-pressure-side branching channel 68 can be obtained by subtracting the flow rate ascertained by the

intermediate-pressure-side branching channel flow rate meter 67Q from the flow rate of the air-warming feed tube 65, which can be ascertained by the feed tube flow rate meter 65Q, by using the intermediate-pressure-side branching channel flow rate meter 67Q and the feed tube flow rate meter 65Q. The control method and calculation method can otherwise be carried out in the same manner as Modification <14-11>.

[0275] The feed tube flow rate meter 65Q may also be provided in place of the intermediate-pressure-side branching channel flow rate meter 67Q rather than as a substitute for the high-pressure-side branching channel flow rate meter 68Q.

<14-13>

[0276] It is also possible to use a heat pump system 9x in which a feed tube temperature sensor 65T is provided in place of the high-pressure-side branching temperature sensor 68T described above in the embodiments, as shown in FIG. 26.

[0277] The feed tube temperature sensor 65T is capable of ascertaining the temperature of the secondary refrigerant for air warming that passes through the air-warming feed tube 65. The heat amount of the secondary refrigerant for air warming that flows through the high-pressure-side branching channel 68 can be ascertained by calculating the heat amount of the secondary refrigerant for air warming that flows through the air-warming feed tube 65, from the temperature of the air-warming feed tube 65, which can be ascertained by the feed tube temperature sensor 65T, and subtracting therefrom the heat amount of the secondary refrigerant for air warming flowing through the intermediate-pressure-side branching channel 67 as obtained from the temperature ascertained by the intermediate-pressure-side branching temperature sensor 67T, by using the intermediate-pressure-side branching temperature sensor 67T and the feed tube temperature sensor 65T. In the case that the flow rate of the secondary refrigerant for air warming flowing through the high-pressure-side branching channel 68 can be ascertained, it is possible to ascertain the temperature of the secondary refrigerant flowing through the high-pressure-side branching channel 68 from the thus-ascertained heat amount of the secondary refrigerant for air warming that flows through the high-pressure-side branching channel 68. Thus, control after the temperature of the secondary refrigerant for air warming flowing through the intermediate-pressure-side branching channel 67 and the temperature of the secondary refrigerant for air warming flowing through the high-pressure-side branching channel 68 are thus-ascertained can be the same as that described in the embodiments above.

[0278] The feed tube temperature sensor 65T may be provided in place of the intermediate-pressure-side branching tube temperature sensor 67T rather than in place of the high-pressure-side branching temperature sensor 68T.

[0279] As described above, in the case that the feed tube temperature sensor 65T is provided, the controller 11 may control the air-warming mixing valve 64 and the air-warming pump 63 so as to reduce the difference between the temperature of the secondary refrigerant for air warming detected by the feed tube temperature sensor 65T and the temperature of the secondary refrigerant for air warming ascertained by another temperature sensor (e.g., the intermediate-pressure-side branching temperature sensor 67T). In this case as well, the same effects as the embodiments described above can be obtained.

<14-14>

[0280] In the embodiments described above, examples were described for the case in which the temperatures of the

secondary refrigerant flowing through the third intermediate-pressure-side branching channel 67c and the third high-pressure-side branching channel 68c are equalized in the secondary refrigerant-temperature equalization control.

[0281] However, no limit is imposed by the case in which the temperatures are made perfectly identical; in any of the embodiments described above, it is also possible to use control in which the difference between the temperature of the secondary refrigerant flowing through the third intermediate-pressure-side branching channel 67c and the temperature of the secondary refrigerant flowing through the third high-pressure-side branching channel 68c is merely reduced.

[0282] It is furthermore possible to perform control so as to satisfy a condition that the difference be equal to or less than a predetermined value, rather than making a reduction in the difference between the temperature of the secondary refrigerant flowing through the third intermediate-pressure-side branching channel 67c and the temperature of the secondary refrigerant flowing through the third high-pressure-side branching channel 68c.

<14-15>

[0283] In the embodiments described above, examples were described for the case in which the flow rate ratio in the air-warming mixing valve 64 is controlled when the secondary refrigerant-temperature equalization control is carried out.

[0284] However, in any of the embodiments described above, no limitation is imposed by control in which the difference between the temperature of the secondary refrigerant for air warming flowing through the third intermediate-pressure-side branching channel 67c and the temperature of the secondary refrigerant for air warming flowing through the third high-pressure-side branching channel 68c is reduced by controlling the flow rate ratio in the air-warming mixing valve 64. For example, also included in the present invention is the case in which the controller 11 performs a control for increasing the flow rate through the air-warming pump 63 or reducing the flow rate through the air-warming pump 63 to thereby reduce the difference between the temperature of the secondary refrigerant for air warming flowing through the third intermediate-pressure-side branching channel 67c and the temperature of the secondary refrigerant for air warming flowing through the third high-pressure-side branching channel 68c.

[0285] For example, in the case that the temperature of the secondary refrigerant flowing through the third high-pressure-side branching channel 68c is less than the temperature of the secondary refrigerant flowing through the third intermediate-pressure-side branching channel 67c, the controller 11 performs a control to reduce the flow rate through the air-warming pump 63 to thereby make it possible to reduce the temperature difference in a state in which the temperature difference between the secondary refrigerant for air warming and the primary refrigerant undergoing heat exchange in the second high-pressure-water heat exchanger 52 is greater than the temperature difference between the secondary refrigerant for air warming and the primary refrigerant undergoing heat exchange in the intermediate-pressure-water heat exchanger 40. In this case, the time for receiving heat from the primary refrigerant in either of the heat exchangers is extended by reducing the flow rate through the air-warming pump 63, but the reason for the considerable effect of increasing the temperature by increasing the time duration is due to the secondary refrigerant for air warming that passes through the second

high-pressure-water heat exchanger 52 side in which the temperature difference between the primary refrigerant and the secondary refrigerant is considerable.

[0286] In the case that the temperature of the secondary refrigerant flowing through the third high-pressure-side branching channel 68c is higher than the temperature of the secondary refrigerant flowing through the third intermediate-pressure-side branching channel 67c, the controller 11 performs a control to increase the flow rate through the air-warming pump 63 to thereby make it possible to reduce the temperature difference in a state in which the temperature difference between the secondary refrigerant for air warming and the primary refrigerant undergoing heat exchange in the second high-pressure-water heat exchanger 52 is greater than the temperature difference between the secondary refrigerant for air warming and the primary refrigerant undergoing heat exchange in the intermediate-pressure-water heat exchanger 40. In this case, the time for receiving heat from the primary refrigerant in either of the heat exchangers is reduced by increasing the flow rate through the air-warming pump 63, but the reason for the considerable effect of reducing the temperature by reducing the time duration is due to the secondary refrigerant for air warming that passes through the second high-pressure-water heat exchanger 52 side in which the temperature difference between the primary refrigerant and the secondary refrigerant is considerable.

[0287] In the case that the temperature of the secondary refrigerant flowing through the third high-pressure-side branching channel 68c is less than the temperature of the secondary refrigerant flowing through the third intermediate-pressure-side branching channel 67c, the controller 11 performs a control to increase the flow rate through the air-warming pump 63 to thereby make it possible to reduce the temperature difference in a state in which the temperature difference between the secondary refrigerant for air warming and the primary refrigerant undergoing heat exchange in the second high-pressure-water heat exchanger 52 is less than the temperature difference between the secondary refrigerant for air warming and the primary refrigerant undergoing heat exchange in the intermediate-pressure-water heat exchanger 40. In this case, the time for receiving heat from the primary refrigerant in either of the heat exchangers is reduced by increasing the flow rate through the air-warming pump 63, but the reason for the considerable effect of reducing the temperature by reducing the time duration is due to the secondary refrigerant for air warming that passes through the intermediate-pressure-water heat exchanger 40 side in which the temperature difference between the primary refrigerant and the secondary refrigerant is considerable.

[0288] In the case that the temperature of the secondary refrigerant flowing through the third high-pressure-side branching channel 68c is greater than the temperature of the secondary refrigerant flowing through the third intermediate-pressure-side branching channel 67c, the controller 11 performs a control to reduce the flow rate through the air-warming pump 63 to thereby make it possible to reduce the temperature difference in a state in which the temperature difference between the secondary refrigerant for air warming and the primary refrigerant undergoing heat exchange in the second high-pressure-water heat exchanger 52 is less than the temperature difference between the secondary refrigerant for air warming and the primary refrigerant undergoing heat exchange in the intermediate-pressure-water heat exchanger 40. In this case, the time for receiving heat from the primary

refrigerant in either of the heat exchangers is increased by reducing the flow rate through the air-warming pump 63, but the reason for the considerable effect of increasing the temperature by extending the time duration is due to the secondary refrigerant for air warming that passes through the intermediate-pressure-water heat exchanger 40 side in which the temperature difference between the primary refrigerant and the secondary refrigerant is considerable.

<14-16>

[0289] In the embodiments described above, examples were described for the case in which no particular control is performed in terms of the relationship between the temperature of the primary refrigerant flowing through the intermediate-pressure-water heat exchanger 40 and the temperature of the primary refrigerant flowing through the second high-pressure-water heat exchanger 52.

[0290] However, in any of the embodiments described above, it is also possible for the controller 11 to control the hot-water supply pump 92 so as to, e.g., adjust and bring the temperature of the primary refrigerant flowing into the second high-pressure-water heat exchanger 52 close to the temperature of the primary refrigerant flowing into the intermediate-pressure-water heat exchanger 40 by adjusting the flow rate of the water for hot-water supply that is passing through the first high-pressure water heat exchanger 51.

[0291] For example, in the case that the target discharge temperature of the high-stage-side compressor 25 is set higher than the target discharge temperature of the low-stage-side compressor 21, it is not possible bring the inlet temperature of the primary refrigerant of the intermediate-pressure-water heat exchanger 40 and the inlet temperature of the primary refrigerant of the second high-pressure-water heat exchanger 52 close together unless the temperature of the primary refrigerant discharged from the high-stage-side compressor 25 is reduced. In such a case, the controller 11 may control the hot-water supply pump 92 on the basis of the temperature detected by the hot-water supply intermediate temperature sensor 95T so that the water for hot-water supply required for cooling the primary refrigerant in the first high-pressure water heat exchanger 51 is fed.

[0292] In this case, the temperature of the secondary refrigerant for air warming flowing through the third intermediate-pressure-side branching channel 67c and the temperature of the secondary refrigerant for air warming flowing through the third high-pressure-side branching channel 68c are more readily brought close together because the temperature near the primary-refrigerant inlet of the intermediate-pressure-water heat exchanger 40 corresponding to the outlet side of the secondary refrigerant for air warming and the temperature near the primary-refrigerant inlet of the second high-pressure-water heat exchanger 52 corresponding to the outlet of the secondary refrigerant for air warming are proximate values. For example, the temperature is more readily equalized when the flow rate through the air-warming pump 63 is reduced. Since the temperature of the secondary refrigerant for air warming flowing through the third intermediate-pressure-side branching channel 67c and the temperature of the secondary refrigerant for air warming flowing through the third high-pressure-side branching channel 68c are more readily brought close together, it is possible to minimize the amount of degradation of the cycle efficiency in the heat pump circuit 10 caused by secondary refrigerant-temperature equalization control.

<14-17>

[0293] In the embodiments described above, examples were described for the case in which the control of the heat pump circuit 10 side is not particularly specified.

[0294] However, the operating conditions are changed by carrying out secondary refrigerant-temperature equalization control in the air-warming circuit 60, and there are cases in which degradation of the cycle efficiency of the heat pump circuit 10 can be reduced or improved.

[0295] Here, the compression ratio in the low-stage-side compressor 21 tends to increase in the case that, e.g., the target discharge temperature of the low-stage-side compressor 21 is increased in order to handle the air-warming heat load, as shown in the Mollier graph of FIG. 27 (see the change from the dotted lines to the dot-dash lines). In accordance with the preceding, there is also an increase in the compression ratio of the high-stage-side compressor 25 with which alignment is being attempted. Therefore, the required drive force is increased and energy consumption is also increased.

[0296] In contrast, the controller 11 may, e.g., modify the operating conditions from the cycle of the dotted lines to the cycle of the solid lines (see the change from the dotted lines to the solid lines), as shown in the Mollier graph of FIG. 28. In other words, it is possible to perform low-stage intake degree-of-superheat control for increasing the degree of superheat of the primary refrigerant taken in by the low-stage-side compressor 21 in the case that the target discharge temperature of the low-stage-side compressor 21 is increased. The compression ratio of the low-stage-side compressor 21 required for achieving the target discharge temperature of the low-stage-side compressor 21 can be minimized thereby. In association therewith, it is also possible to minimize the compression ratio of the high-stage-side compressor 25. The required drive force can be further minimized thereby.

[0297] It is conversely possible to perform low-stage intake degree-of-superheat control for reducing the degree of superheat of the primary refrigerant taken in by the low-stage-side compressor 21 in the case that the cycle state is modified so that the target discharge temperature of the low-stage-side compressor 21 is reduced. The specific volume of the primary refrigerant taken in by the low-stage-side compressor 21 can be reduced while an increase in the compression ratio of the high-stage-side compressor 25 is suppressed by minimizing an increase in the compression ratio of the low-stage-side compressor 21. It is therefore possible to ensure a circulation amount and to increase capacity while suppressing an increase in the compression ratio.

[0298] For example, in the control described above, the controller 11 controls the opening degree of the primary bypass expansion valve 5b, making it possible to adjust the amount of heat exchange in the primary-refrigerant-to-primary-refrigerant heat exchanger 8 in the heat pump circuit 10 having the primary bypass 80 and the primary bypass expansion valve 5b among the heat pump systems described in the embodiments and modifications above. Thus, the superheat degree of the primary refrigerant taken in by the low-stage-side compressor 21 can be adjusted.

<14-18>

[0299] In the embodiments described above, examples were described for the case in which the control of the heat pump circuit 10 side is not particularly specified.

[0300] However, the operating conditions are changed by carrying out secondary refrigerant-temperature equalization control in the air-warming circuit 60, and there are cases in

which degradation of the cycle efficiency of the heat pump circuit **10** can be reduced or improved.

[0301] Here, there may be cases in which a high temperature is not required as the temperature of the primary refrigerant flowing through the intermediate-pressure-water heat exchanger **40** in the case that the temperature of the secondary refrigerant for air warming is not significantly reduced in the radiator **61** due to, among other factors, a lower air-warming heat load.

[0302] In contrast, the controller **11** may, e.g., modify the operating conditions from the cycle of the dotted lines to the cycle of the solid lines (see the change from the dotted lines to the solid lines), as shown in the Mollier graph of FIG. 29. In other words, it is possible to perform control so that the degree of superheat of the primary refrigerant taken in by the low-stage-side compressor **21** is also reduced while the target discharge temperature of the low-stage-side compressor **21** is reduced. The compression ratio of the high-stage-side compressor **25** and the compression ratio of the low-stage-side compressor **21** become substantially the same, and it is possible to achieve efficient operation in which the drive forces of the low-stage-side compressor **21** and the high-stage-side compressor **25** are minimized. It is possible to adapt to the load even when the target discharge temperature of the low-stage-side compressor **21** is reduced in this manner, because the heat load requested by the radiator **61** is reduced. The compression drive force can thereby be further reduced while adapting to load fluctuations.

[0303] For example, in the control described above, the controller **11** controls the opening degree of the primary bypass expansion valve **5b**, making it possible to adjust the amount of heat exchange in the primary-refrigerant-to-primary-refrigerant heat exchanger **8** in the heat pump circuit **10** having the primary bypass **80** and the primary bypass expansion valve **5b** among the heat pump systems described in the embodiments and modifications above. Thus, the superheat degree of the primary refrigerant taken in by the low-stage-side compressor **21** can be adjusted.

INDUSTRIAL APPLICABILITY

[0304] The refrigeration apparatus of the present invention is capable of improving cycle efficiency in the processing of the heat load performed by the secondary refrigerant, and is therefore particularly useful in the case that application is made to a heat pump system for processing a heat load using a heat pump circuit provided with a multistage compression-type compression element.

REFERENCE SIGNS LIST

[0305] **1** Heat pump system
 [0306] **4** Evaporator
 [0307] **4/f** Fan
 [0308] **5a** Expansion valve
 [0309] **5b** Primary bypass expansion valve
 [0310] **7** Economizer heat exchanger
 [0311] **8** Primary-refrigerant-to-primary-refrigerant heat exchanger
 [0312] **10** Heat pump circuit
 [0313] **20** Low-pressure tube
 [0314] **20a to f** First to sixth low-pressure tube
 [0315] **20P** Low-pressure pressure sensor
 [0316] **20T** Low-pressure temperature sensor
 [0317] **21** Low-stage-side compressor

[0318] **23** Intermediate pressure tube
 [0319] **23a to d** First to fourth intermediate pressure tube
 [0320] **23T** Intermediate-pressure temperature sensor
 [0321] **24P** High-stage intake pressure sensor
 [0322] **24T** High-stage intake temperature sensor
 [0323] **25** High-stage-side compressor
 [0324] **27** High-pressure tube
 [0325] **27a to n** First to fourteenth high-pressure tube
 [0326] **27P** High-pressure pressure sensor
 [0327] **27T** High-pressure temperature sensor
 [0328] **40** Intermediate-pressure water heat exchanger
 [0329] **50** High-pressure water heat exchanger
 [0330] **51 to 53** First to third high-pressure water heat exchanger
 [0331] **60** Air-warming circuit
 [0332] **61** Radiator
 [0333] **61T** Radiator temperature sensor
 [0334] **62** Branch flow mechanism (first flow rate adjustment mechanism)
 [0335] **63** Air-warming pump (flow rate adjustment section)
 [0336] **64** Air-warming mixing valve
 [0337] **65** Air-warming feed tube
 [0338] **65T** Feed tube temperature sensor
 [0339] **65Q** Feed tube flow rate meter
 [0340] **66** Air-warming return tube
 [0341] **66T** Air-warming return temperature sensor
 [0342] **67** Intermediate-pressure-side branching channel
 [0343] **67T** Intermediate-pressure-side branching channel temperature sensor
 [0344] **67Q** Intermediate-pressure-side branching channel flow rate meter
 [0345] **67a to c** First to third intermediate-pressure-side branching channel
 [0346] **68** High-pressure-side branching channel
 [0347] **68T** High-pressure-side branching channel temperature sensor
 [0348] **68Q** High-pressure-side branching channel flow rate meter
 [0349] **69** Air-warming bypass channel (first heat load bypass channel)
 [0350] **70** Injection channel
 [0351] **72** First injection tube
 [0352] **74** Second injection tube
 [0353] **75** Third injection tube
 [0354] **76** Fourth injection tube
 [0355] **73** Injection expansion valve
 [0356] **90** Primary bypass
 [0357] **90** Hot-water supply circuit
 [0358] **91** Hot-water storage tank
 [0359] **92** Hot-water supply pump
 [0360] **93** Hot-water supply mixing valve
 [0361] **94** Water supply tube
 [0362] **94T** Hot-water supply water-intake temperature sensor
 [0363] **95** Hot-water supply heat pump tube
 [0364] **95a to f** First to sixth hot-water supply heat pump tube
 [0365] **95T** Hot-water supply intermediate temperature sensor
 [0366] **98** Hot-water supply tube
 [0367] **98T** Hot-water supply hot-water outlet temperature sensor
 [0368] **99** Hot-water supply bypass tube

- [0369] 164 Twelfth air-warming mixing valve (first heat load bypass flow rate adjustment mechanism)
- [0370] A Intake point
- [0371] B Low-stage discharge point
- [0372] C Intermediate-pressure water heat exchanger passage point
- [0373] D Injection merge point
- [0374] E High-stage discharge point
- [0375] F First high-pressure point
- [0376] G Second high-pressure point
- [0377] H Third high-pressure point
- [0378] I Fourth high-pressure point
- [0379] J Fifth high-pressure point
- [0380] K First low-pressure point
- [0381] L Second low-pressure point
- [0382] M Third low-pressure point
- [0383] N Fourth low-pressure point
- [0384] Q Injection intermediate-pressure point
- [0385] R Economizer post-heat-exchange point
- [0386] X Air-warming branching point
- [0387] Y Air-warming merging point
- [0388] W Water supply branching point
- [0389] Z Hot-water supply merging point

CITATION LIST

Patent Literature

- [0390] <Patent Literature 1> Japanese Laid-open Patent Application Publication No. 2004-177067

1. A heat pump system comprising:

a heat pump circuit configured to have a primary refrigerant circulated therethrough, the heat pump circuit including at least a low-stage-side compression element, a high-stage-side compression element, an expansion element, and an evaporator;

a first heat load circuit configured to have a first fluid circulated therethrough, the first heat load circuit having a first branching portion,

a second branching portion,

a first branching channel arranged to connect the first branching portion and the second branching portion,

a second branching channel arranged to connect the first branching portion and the second branching portion without merging with the first branching channel, and

a first heat-load-processing section;

a first heat exchanger arranged and configured to perform heat exchange between

the primary refrigerant flowing from a discharge side of the low-stage-side compression element toward an intake side of the high-stage-side compression element and

the first fluid flowing through the first branching channel;

a second heat exchanger arranged and configured to perform heat exchange between

the primary refrigerant flowing from the high-stage-side compression element toward the expansion element and

the first fluid flowing through the second branching channel;

a first flow rate adjustment element arranged and configured to adjust at least one of

a flow rate of the first fluid in the first branching channel and the

a flow rate of the first fluid in the second branching channel; and

a controller configured to perform flow rate adjustment control of the first flow rate adjustment element so as to maintain a state in which a predetermined temperature condition is satisfied, including a case in which a ratio between temperature of the first fluid flowing through a portion of the first branching channel that has passed through the first heat exchanger and temperature of the first fluid flowing through a portion of the second branching channel that has passed through the second heat exchanger is 1, or

reduce a difference between the temperature of the first fluid flowing through a portion of the first branching channel that has passed through the first heat exchanger and the temperature of the first fluid flowing through a portion of the second branching channel that has passed through the second heat exchanger,

the heat pump circuit, the first heat load circuit, the first heat exchanger, the second heat exchanger, the first flow rate adjustment element and the controller being configured and arranged such that the first fluid cooled in the first heat load circuit and not yet warmed can be fed to the first heat exchanger and the second heat exchanger.

2. The heat pump system according to claim 1, wherein the controller is further configured to control output of the low-stage-side compression element and the high-stage-side compression element so that

temperature of the primary refrigerant that flows into the first heat exchanger and temperature of the primary refrigerant that flows into the second heat exchanger both become a temperature equal to or greater than a first heat-load-corresponding temperature requested in the first heat-load-processing section, while causing

the temperature of the primary refrigerant flowing to the first heat exchanger to become a temperature equal to or greater than the first fluid flowing to the first heat exchanger, and

the temperature of the primary refrigerant flowing to the second heat exchanger to become a temperature equal to or greater than the first fluid flowing to the second heat exchanger.

3. The heat pump system according to claim 2, wherein the first heat load circuit further includes

a first heat load bypass circuit arranged to connect a portion between the first heat-load-processing section and the first branching portion, and a portion between the first heat-load-processing section and the second branching portion; and

a first heat-load-bypass flow-rate-adjustment element arranged and configured to adjust a flow rate of the first fluid that passes through the first heat load bypass circuit;

the controller is further configured to perform a control in the flow rate adjustment control so that a target value of the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger and a target value of the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger become a temperature that exceeds the first heat-load-corresponding temperature; and

the controller is further configured to operate the first heat-load-bypass flow-rate-adjustment element and adjust the flow rate of the first fluid flowing through the first

- heat load bypass circuit so that the temperature of the first fluid fed to the first heat-load-processing section becomes the first heat-load-corresponding temperature.
4. The heat pump system according to claim 2, wherein the controller is further configured to perform a control in the flow rate adjustment control so that a target value of the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger and a target value of the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger become the first heat-load-corresponding temperature.
 5. The heat pump system according to claim 2, wherein the controller is further configured to control at least one of the low-stage-side compression element, the high-stage-side compression element, and the expansion element in the flow rate adjustment control so as to:
 - maintain a state in which a predetermined compression ratio condition is satisfied, including a case in which a ratio between compression ratio in the low-stage-side compression element and compression ratio in the high-stage-side compression element is 1, or
 - reduce a difference between the compression ratio in the low-stage-side compression element and the compression ratio in the high-stage-side compression element.
 6. The heat pump system according to claim 5, wherein the controller is further configured to perform a low-stage intake degree-of-superheat control in order to increase a degree of superheat of the primary refrigerant taken in by the low-stage-side compression element in a case where the discharge temperature of the primary refrigerant of the low-stage-side compression element increases when the flow rate adjustment control is performed.
 7. The heat pump system according to claim 6, wherein the heat pump circuit further includes a primary-refrigerant-to-primary-refrigerant heat exchanger arranged and configured to cause heat exchange to be performed between the primary refrigerant taken in by the low-stage-side compression element and the primary refrigerant that has passed through the second heat exchanger and then flows toward the expansion element; and the controller is further configured to perform a low-stage intake degree-of-superheat control using the primary-refrigerant-to-primary-refrigerant heat exchanger.
 8. The heat pump system according to claim 5, wherein the controller is further configured to perform a control during load reduction in order to reduce a degree of superheat of the primary refrigerant taken in by the low-stage-side compression element while reducing a target value of the discharge temperature of the primary refrigerant of the low-stage-side compression element in a case where the temperature of the first fluid flowing from the first heat-load-processing section toward the first heat exchanger and the second heat exchanger has increased when the flow rate adjustment control is performed.
 9. The heat pump system according to claim 8, further comprising:
 - a second heat load circuit configured to have a second fluid circulated therethrough, the second heat load circuit having a second heat load section; and
 - a third heat exchanger arranged and configured to cause heat exchange to be performed between the second fluid circulating through the second heat load circuit and the primary refrigerant flowing from the high-stage-side compression element toward the second heat exchanger.
 10. The heat pump system according to claim 9, further comprising
 - a fourth heat exchanger arranged and configured to cause heat exchange to be performed between the second fluid flowing from the second heat-load-processing section toward the third heat exchanger, among the second fluid that passes through the second heat load circuit, and the primary refrigerant which has passed through the second heat exchanger and is thereafter flowing toward the expansion element.
 11. The heat pump system according to claim 9, wherein the controller is further configured to adjust a circulation amount of the second fluid circulating through the second heat load circuit so that the temperature of the primary refrigerant that passes through the third heat exchanger approximates a target value of the temperature of the primary refrigerant discharged by the low-stage-side compression element in a case where the target value of the temperature of the primary refrigerant discharged by the low-stage-side compression element is less than the target value of the temperature of the primary refrigerant discharged by the high-stage-side compression element.
 12. The heat pump system according to claim 9, wherein the second heat load processing section is a hot-water-supply tank; and the second fluid is water used for hot-water supply.
 13. The heat pump system according to claim 2, wherein the controller is further configured to operate the first flow rate adjustment element in the flow rate adjustment control in order to reduce flow rate of the first fluid having a lower temperature of
 - the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger; and
 - the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger.
 14. The heat pump system according to claim 13, wherein the first flow rate adjustment element arranged and configured to adjust a ratio between a flow rate of the first fluid flowing through the first branching channel and a flow rate of the first fluid flowing through the second branching channel; and the controller is further configured to operate the first flow rate adjustment element in the flow rate adjustment control in order to reduce the flow rate ratio of the first fluid having a lower temperature of
 - the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger; and
 - the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger,
 while keeping constant a flow rate of the first fluid fed to the first heat-load-processing section.
 15. The heat pump system according to claim 13, wherein the first flow rate adjustment element is further arranged and configured to adjust a flow rate of the first fluid fed to the first heat-load-processing section; and the controller is further configured, in the flow rate adjustment control, to reduce the flow rate of the first fluid fed to the first heat-load-processing section by operating the

first flow rate adjustment element in a case where a flow rate ratio is low for the first fluid having a lower temperature of

the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger; and

the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger.

16. The heat pump system according to claim **13**, wherein the first flow rate adjustment element includes

a ratio adjustment section configured to adjust a ratio between a flow rate of the first fluid flowing through the first branching channel and a flow rate of the first fluid flowing through the second branching channel, and

a flow rate adjustment section configured to adjust a flow rate of the first fluid fed to the first heat-load-processing section;

the controller is further configured to operate the first flow rate adjustment element in the flow rate adjustment control in order to

increase the flow rate of the first fluid having a temperature that exceeds the first heat-load-corresponding temperature and/or

reduce the flow rate of the first fluid having a temperature that is less than the first heat-load-corresponding temperature,

as determined from among the temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger and the temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger; and

the controller is further configured to reduce the flow rate of the first fluid fed to the first heat-load-processing section in proportion to the increase of the temperature of the first fluid fed to the first heat-load-processing section in a case where the temperature of the first fluid fed to the first heat-load-processing section has exceeded the first heat-load-corresponding temperature.

17. The heat pump system according to claim **1**, further comprising:

a first branching channel temperature detector arranged and configured to ascertain temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger; and

a second branching channel temperature detector arranged and configured to ascertain temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger.

18. The heat pump system according to claim **1**, further comprising:

a branching portion temperature detector arranged and configured to ascertain at least one of

a temperature of the first fluid flowing through the portion of the first branching channel that has passed through the first heat exchanger and

a temperature of the first fluid flowing through the portion of the second branching channel that has passed through the second heat exchanger; and

a merging portion temperature detector arranged and configured to ascertain a temperature of the first fluid flow-

ing toward the first heat-load-processing section after the first fluid which has passed through the first branching channel has merged with the first fluid which has passed through the second branching channel.

19. The heat pump system according to claim **1**, further comprising:

a first branching channel flow rate detector arranged and configured to ascertain a flow rate of the first fluid flowing through the first branching channel; and

a second branching channel flow rate detector arranged and configured to ascertain a flow rate of the first fluid flowing through the second branching channel.

20. The heat pump system according to claim **1**, further comprising:

a branching portion flow rate detector arranged and configured to ascertain at least one of a flow rate of the first fluid flowing through the first branching channel and a flow rate of the first fluid flowing through the second branching channel; and

a merging portion flow rate detector arranged and configured to ascertain a flow rate of the first fluid flowing toward the first heat-load-processing section after the first fluid flowing through the first branching channel and the first fluid flowing through the second branching channel have merged.

21. The heat pump system according to claim **1**, wherein the primary refrigerant flowing from the discharge side of the low-stage-side compression element toward the intake side of the high-stage-side compression element and the first fluid flowing through the first branching channel are in an opposing-flow relationship in the first heat exchanger; and

the primary refrigerant flowing from the high-stage-side compression element toward the expansion element and the first fluid flowing through the second branching channel are in an opposing-flow relationship in the second heat exchanger.

22. The heat pump system according to claim **1**, wherein the first heat-load-processing section is an air-warming heat exchanger used to warm air in a disposed target space; and

the first fluid is a secondary refrigerant.

23. The heat pump system according to claim **1**, wherein the low-stage-side compression element and the high-stage-side compression element utilize have a shared rotating shaft, which is rotatably driven such that compression work is performed.

24. The heat pump system according to claim **1**, wherein the controller is further configured to keep a discharge pressure of the high-stage-side compression element at a pressure that is equal to or greater than a critical pressure of the primary refrigerant in the flow rate adjustment control; and

the heat pump system is configured to be used in an environment in which ambient temperature of the first heat-load-processing section is a temperature equal to or less than a critical temperature of the primary refrigerant.

25. The heat pump system according to claim **1**, wherein the primary refrigerant is carbon dioxide.

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