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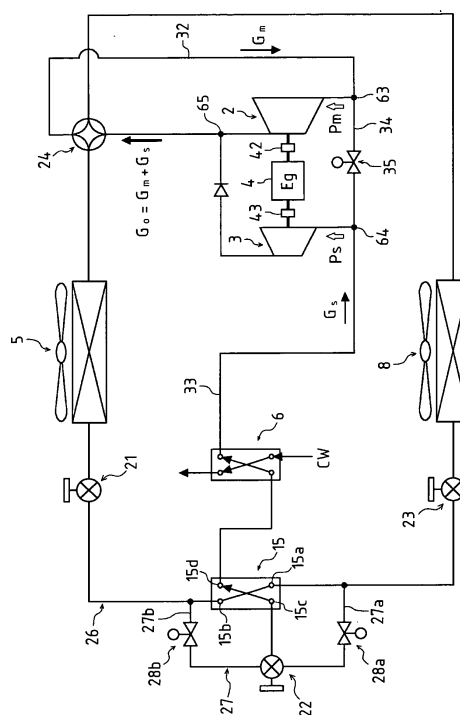
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(54) **ENGINE HEAT PUMP**

(57) An object of the invention is to provide an engine heat pump whose compression work is reduced without increase of consumed electric power, thereby improving its driving efficiency (energy efficiency). An engine heat pump comprises: an engine (4); a main compressor (2) driven by the engine (4); a sub compressor (3) delivering a refrigerant to be joined to a refrigerant delivered from the main compressor (2); an indoor heat exchanger (8); an outdoor heat exchanger (5); an expansion valve (23) for the indoor heat exchanger; an expansion valve (21) for the outdoor heat exchanger; and a supercooling heat exchanger (15) disposed on a liquid refrigerant passage (main passage 26) of a connection passage between the indoor heat exchanger (8) and the outdoor heat exchanger (5). In the supercooling heat exchanger (15), a supercooling liquid refrigerant branched into a branching passage (27(27a, 27b)) supercools a liquid refrigerant before being branched. The sub compressor (3) is driven by the engine (4) so as to compress the supercooling liquid refrigerant. A ratio (R) of a capacity of the sub compressor (3) to a total capacity of the main compressor (2) and the sub compressor (3) ranges between 20% and 29%.

FIG. 1



Description

Technical Field

- 5 **[0001]** The invention relates to a system of an engine heat pump. Particularly, the invention relates to a technology for reducing the total compression work of the engine heat pump without increase of consumed electric power.

Background Art

- 10 **[0002]** There is a well-known conventional engine heat pump including a compressor driven by an engine as disclosed in Japanese Laid Open Gazette No. 2004-20153. A compression work of the engine heat pump is shared between a main compressor and a sub compressor. An evaporation pressure set for one compressor (the sub compressor) is kept higher than that for the other compressor (the main compressor) so as to reduce the compression work by the one compressor, thereby reducing the total compression work of the engine heat pump.
- 15 The reference document discloses an electromotive compressor serving as the compressor (sub compressor) subjected to the higher evaporation pressure. Namely, the engine heat pump is provided with an additional device (i.e., the electromotive compressor) requiring electric power. As a result, although the compression work is reduced, the consumed electric power is increased so as to reduce the essential merit of the engine heat pump, that is, reduction of consumed electric power.

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Brief Summary of the Invention

Object of the Invention

- 25 **[0003]** An object of the invention is to provide an engine heat pump whose compression work is reduced without increase of consumed electric power, thereby improving its driving efficiency (energy efficiency).

Way for Attaining the Object

- 30 **[0004]** An engine heat pump according to the invention comprises: an engine; a main compressor driven by the engine; a sub compressor delivering a refrigerant to be joined to a refrigerant delivered from the main compressor; an indoor heat exchanger; an outdoor heat exchanger; an expansion valve for the indoor heat exchanger; an expansion valve for the outdoor heat exchanger; and a supercooling heat exchanger disposed on a liquid refrigerant passage of a connection passage between the indoor heat exchanger and the outdoor heat exchanger. In the supercooling heat exchanger, a
- 35 supercooling liquid refrigerant branched into a branching passage supercools a liquid refrigerant before being branched. The sub compressor is driven by the engine so as to compress the supercooling liquid refrigerant. A ratio of a capacity of the sub compressor to a total capacity of the main compressor and the sub compressor ranges between 20% and 29%. In the engine heat pump according to the invention, an engine exhaust heat recovery unit is disposed in parallel to the outdoor heat exchanger. The supercooling liquid refrigerant is evaporated by the engine exhaust heat recovery unit and
- 40 compressed by the sub compressor.

Effect of the Invention

- 45 **[0005]** In the engine heat pump of the invention, the sub compressor is driven by the engine so as to compress the supercooling refrigerant subjected to a higher evaporation pressure (refrigerant suction pressure) than the evaporation pressure of refrigerant compressed by the main compressor. Therefore, the engine heat pump requires no additional electric power required for an electromotive sub compressor, so that the total compression work in a refrigeration cycle is reduced while ensuring or improving the cooling capacity by the supercooling performance of the supercooling heat exchanger.
- 50 **[0006]** Further, a capacity ratio of the sub compressor relative to a total capacity of the main compressor and the sub compressor the sub compressor is set within a certain range. Therefore, while the cooling efficiency for the cooling operation is ensured or improved, the capacity of the supercooling heat exchanger for the heating operation is ensured. That is, in the present invention, since the common engine drives the main compressor and the sub compressor, sufficient driving efficiency (energy efficiency) is ensured whether the engine heat pump is operated for cooling or for heating.
- 55 In the engine heat pump of the invention, since a capacity ratio of the sub compressor to a total capacity of the main compressor and the sub compressor the sub compressor is set within a certain range, while the total compression work for the cooling operation is reduced, the total compression work for the heating operation is also reduced because it requires no additional electric power.

[0007] When the engine heat pump is operated for heating, the liquid refrigerant is supercooled so as to improve the capacity of refrigerant per unit mass and unit flow rate for absorbing heat from an outdoor air, thereby reducing the total amount of refrigerant flowing in the refrigeration cycle. As a result, the total compression work is reduced so as to improve the driving efficiency (energy efficiency) of the engine heat pump.

Brief Description of the Drawings

[0008]

Fig. 1 is a diagram of a refrigerant circuit of an engine heat pump according to the invention.
 Fig. 2 is a block diagram of control units for the refrigerant circuit of the engine heat pump.
 Fig. 3 is a Moliere chart due to the refrigerant circuit.
 Fig. 4 is a graph of COP relative to the capacity share of a sub compressor.
 Fig. 4 is a graph of temperature of refrigerant in a supercooling exchanger relative to the capacity share of a sub compressor.

Description of Notations

[0009]

2 Main Compressor
 3 Sub compressor
 4 Engine
 5 Outdoor Heat Exchanger
 6 Engine Exhaust Heat Recovery Unit
 8 Indoor Heat Exchanger
 15 Supercooling Heat Exchanger
 21 Expansion Valve for Outdoor Heat Exchanger
 22 Expansion Valve for Supercooling Heat Exchanger
 23 Expansion Valve for Indoor Heat Exchanger
 26 Main Passage
 27a Divisional Passage
 27b Divisional Passage

Best Mode for Carrying out the Invention

[0010] A refrigerant circuit system and a refrigeration cycle of an engine heat pump according to the invention will be described with reference to Figs. 1.

[0011] The engine heat pump of the invention comprises: an engine 4; a main compressor 2 driven by engine 4; a sub compressor 3 driven by engine 4; an indoor heat exchanger 8; an outdoor heat exchanger 5; an expansion valve 23 for the indoor heat exchanger 8; an expansion valve 21 for the outdoor heat exchanger; and a supercooling heat exchanger 15.

[0012] Supercooling heat exchanger 15 is disposed on a main passage 26 serving as a liquid refrigerant passage of a connection passage between indoor heat exchanger 8 and outdoor heat exchanger 5, so that, in supercooling heat exchanger 15, a supercooling liquid refrigerant branched into a branching passage 27 (including passages 27a and 27b) supercools a liquid refrigerant before being branched. The engine heat pump uses a refrigeration cycle caused by these components. Supercooling heat exchanger 15 includes connection points 15a and 15b to be connected to main passage 26, and connection points 15c and 15d to be connected to branching passage 27. A plurality of indoor heat exchangers 8 may be provided to the engine heat pump.

Main compressor 2, driven by engine 4, absorbs and compresses a vapor refrigerant separated from a liquid refrigerant by an accumulator (not shown), and delivers vapor refrigerant having a high temperature and a high pressure. A four-way valve 24 introduces the vapor refrigerant delivered from main compressor 2 into a predetermined direction. Four-way valve 24 also receives a vapor refrigerant to be supplied to main compressor 2. Therefore, a passage 32, serving as a suction line of main compressor 2, is interposed between a refrigerant inlet of main compressor 2 and four-way valve 24, so as to introduce the vapor refrigerant from four-way valve 24 into main compressor 2.

[0013] A supercooling liquid refrigerant is separated into branching passage 27 and passed through supercooling heat exchanger 15. An accumulator (not shown) separates a vapor refrigerant from the supercooling liquid refrigerant passed through supercooling heat exchanger 15. Sub compressor 3, driven by engine 4, absorbs and compresses the vapor

refrigerant separated from the supercooling liquid refrigerant, and delivers a vapor refrigerant having a high temperature and a high pressure.

An expansion valve 22 for the supercooling heat exchanger is provided on branching passage 27, so as to cool a refrigerant as the supercooling liquid refrigerant supplied to supercooling heat exchanger 15 for supercooling the liquid refrigerant before being branched. Sub compressor 3 absorbs the supercooling liquid refrigerant whose heat has been exchanged by supercooling heat exchanger 15. Therefore, a passage 33, serving as a suction line of sub compressor 3, is interposed between supercooling heat exchanger 15 and a refrigerant inlet of sub compressor 3.

Branching passage 27 connected to main passage 26 includes branching passage 27a between indoor heat exchanger 8 and supercooling heat exchanger 15. Branching passage 27 also includes branching passage 27b between outdoor heat exchanger 5 and supercooling heat exchanger 15. An on-off valve 28a is interposed between branching passage 27a and expansion valve 22, and an on-off valve 28b is interposed between branching passage 27b and expansion valve 22. During either a cooling cycle or a heating cycle, on-off valves 28a and 28b are selectively switched on or off for supercooling the liquid refrigerant in main passage 26 before being branched.

The refrigerant delivered from sub compressor 3 is joined to the refrigerant delivered from main compressor 2 at a confluence point 65 before four-way valve 24. Four-way valve 24 changes the flow direction of the joined refrigerant from confluence point 65, so as to perform either the cooling cycle or the heating cycle. An oil separator (not shown) is interposed between confluence point 65 and four-way valve 24. The oil separator separates a refrigerator oil from the high-temperature and high-pressurized vapor refrigerant, and returns the refrigerator oil to suction sides of main compressor 2 and sub compressor 3, so as to smoothly lubricate compressors 2 and 3.

The above construction makes a refrigeration cycle, which is switched between the cooling cycle and the heating cycle depending on the change of the flow direction of refrigerant by four-way valve 24.

[0014] In the cooling cycle, the respective refrigerants compressed by main compressor 2 and sub compressor 3 join each other at confluence point 65, and the joined refrigerant is sent to outdoor heat exchanger 5 through four-way valve 24. The refrigerant is radiated and condensed by outdoor heat exchanger 5, and sent to supercooling heat exchanger 15. The refrigerant flows into supercooling heat exchanger 15 through connection point 15b, and flows out from supercooling heat exchanger 15 through connection point 15a. The liquid refrigerant supercooled by supercooling heat exchanger 15 is expanded by expansion valve 23, and evaporated by the endothermic action of indoor heat exchanger 8. The vapor refrigerant is supplied to main compressor 2 through four-way valve 24, and compressed by main compressor 2 to be delivered again.

A part of the liquid refrigerant flowing from outdoor heat exchanger 5 through main passage 26 is branched into branching passage 27a so as to serve as the supercooling liquid refrigerant. The branched refrigerant is expanded and cooled by expansion valve 22 so as to be changed into a wet low-temperature refrigerant. The wet low-temperature refrigerant flows into supercooling heat exchanger 15 through connection point 15c, supercools the refrigerant flowing in main passage 26, and flows out from supercooling heat exchanger 15 through connection point 15d. In this cycle, on-off valve 28a is opened and on-off valve 28b is closed, so that the refrigerant flowing in main passage 26 is not branched into branching passage 27b, but into branching passage 27a. The supercooling refrigerant branched into branching passage 27a supercools the undivided liquid refrigerant before being branched.

[0015] In this way, the liquid refrigerant flowing through main passage 26 is supercooled so as to improve the efficiency of the refrigeration cycle. The supercooling liquid refrigerant is supplied to sub compressor 3 so as to be compressed and delivered again.

On the other hand, in the heating cycle, the respective refrigerants compressed by main compressor 2 and sub compressor 3 join each other at confluence point 65, and the joined refrigerant is sent to indoor heat exchanger 8 through four-way valve 24. The refrigerant is radiated and condensed (liquefied) by indoor heat exchanger 8, and sent to supercooling heat exchanger 15. The refrigerant flows into supercooling heat exchanger 15 through connection point 15a, and flows out from supercooling heat exchanger 15 through connection point 15b. The liquid refrigerant supercooled by supercooling heat exchanger 15 is expanded by expansion valve 21, and evaporated by the endothermic action of outdoor heat exchanger 5. The vapor refrigerant is supplied to main compressor 2 through four-way valve 24, and compressed by main compressor 2 to be delivered again.

A part of the liquid refrigerant flowing from indoor heat exchanger 8 through main passage 26 is branched into branching passage 27b so as to serve as the supercooling liquid refrigerant. The branched refrigerant is expanded and cooled by expansion valve 22 so as to be changed into a wet low-temperature refrigerant. The wet low-temperature refrigerant flows into supercooling heat exchanger 15 through connection point 15c, supercools the refrigerant flowing in main passage 26, and flows out from super cooling heat exchanger 15 through connection point 15d. In this cycle, on-off valve 28a is closed and on-off valve 28b is opened, so that the refrigerant flowing in main passage 26 is not branched into branching passage 27a, but into branching passage 27b. The supercooling refrigerant branched into branching passage 27b supercools the undivided liquid refrigerant before being branched.

[0016] The liquid refrigerant flowing out from supercooling heat exchanger 15 is radiated and evaporated by engine exhaust heat recovery unit 6, and supplied to sub compressor 3 so as to be compressed and delivered again.

A system for controlling the engine heat pump of the present invention will be described with reference to Fig. 2.

[0017] A controller 25 serves as a control unit provided in the engine heat pump of the present invention. Controller 25 is connected to expansion valve 21 for the outdoor heat exchanger, expansion valve 22 for the supercooling heat exchanger, and expansion valve 23 for the indoor heat exchanger, so as to control the respective expansion valves.

Controller 25 is also connected to on-off valves 28a and 28b provided on respective branching passages 27a and 27b. On-off valves 28a and 28b are controlled as follows:

On-off valve 28a is opened for supercooling the liquid refrigerant in the cooling cycle, and closed in the other cases. On-off valve 28b is opened for supercooling the liquid refrigerant in the heating cycle, and closed in the other cases. Due to such a control of on-off valves 28a and 28b, whether it is the cooling cycle or the heating cycle, the liquid refrigerant is branched at the downstream side of supercooling heat exchanger 15, and the undivided whole liquid refrigerant before being branched is supercooled by supercooling heat exchanger 15.

[0018] Controller 25 is connected to engine 4 (or a control circuit of engine 4), so as to control the on-off operation of engine 4 for driving main compressor 2 and sub compressor 3.

In this system, controller 25 controls the opening of expansion valve 22 so that the wet refrigerant expanded by expansion valve 22 is overheated in passage 33 serving as the suction line of sub compressor 3. As discussed later, sub compressor 3 is selected (or configured) so that the refrigerant suction pressure of sub compressor 3 becomes higher than the refrigerant suction pressure of main compressor 2, that is, as shown in the Mollier chart of Fig. 3, a compression work ΔW_s by sub compressor 3 becomes smaller than a compression work ΔW_m by main compressor 2, thereby reducing the total compression work of the engine heat pump in comparison with only the compression work ΔW_m for the whole refrigerant.

The Mollier chart of the refrigeration cycle due to the above refrigerant circuit system will be described with reference to the flow of refrigerant in the refrigerant circuit system. The Mollier chart indicates a change of condition of refrigerant having a unit mass and a unit flow rate. The axis of abscissas indicates a specific enthalpy (kJ/kg), which is an energy of 1kg refrigerant. The axis of ordinate indicates a (absolute) pressure (MPa abs).

A cooling cycle regarding to the refrigeration cycle indicated as the Mollier chart will be described.

[0019] A point Am on the Mollier chart defines a condition of the refrigerant flowing in passage 32 serving as the suction line of main compressor 2. At point Am, the refrigerant is set to have a specific enthalpy h_2 (kJ/kg) and a pressure p_2 (MPa abs), and have a flow rate G_m . A point As on the Mollier chart defines a condition of the refrigerant flowing in passage 33 serving as the suction line of sub compressor 3. At point As, the refrigerant is set to have a specific enthalpy h_1 (kJ/kg) and a pressure p_1 (MPa abs), and have a flow rate G_s .

[0020] The refrigerants set to have the above conditions at the respective points are supplied to respective compressors 2 and 3, so as to be compressed. Main compressor 2 performs compression work ΔW_m relative to the refrigerant having the unit mass and the unit flow rate (a compression process AmB). Sub compressor 3 performs compression work ΔW_s relative to the refrigerant having the unit mass and the unit flow rate (a compression process AsB).

The high-pressurized refrigerants (vapor refrigerants) compressed by respective compressors 2 and 3 join each other at confluence point 65. At confluence point 65, the joined refrigerant has a total flow rate $G_o (=G_m+G_s)$. The joined high-pressurized vapor refrigerant is sent to outdoor heat exchanger 5 so as to be condensed and radiated, i.e., cooled, thereby being changed into the liquid refrigerant (a condensation process BC). In this regard, a point B defines a condition of the refrigerant flowing in the passage from confluence point 65 to outdoor heat exchanger 5. At point B, the refrigerant is set to have a specific enthalpy h_0 (kJ/kg).

The liquid refrigerant discharged from outdoor heat exchanger 5 is supercooled by the supercooling liquid refrigerant which has been branched into branching passage 27a at the downward side of supercooling heat exchanger 15 (a supercooling process CD). In this process, isothermal lines T_1 , T_2 and T_3 are set at respective constant temperatures $t_1(^{\circ}\text{C})$, $t_2(^{\circ}\text{C})$ and $t_3(^{\circ}\text{C})$ ($t_1>t_2>t_3$). The chart indicates that supercooling heat exchanger 15 supercools the liquid refrigerant flowing in main passage 26 so as to reduce the temperature of the liquid refrigerant flowing in main passage 26 from $t_1(^{\circ}\text{C})$ to $t_2(^{\circ}\text{C})$. The supercooled liquid refrigerant has a pressure p_0 (MPa abs) at a point D.

A part of the supercooled liquid refrigerant flowing in main passage 26 is branched, and the remaining supercooled liquid refrigerant is expanded by expansion valve 23 for the indoor heat exchanger, so as to be changed into a liquid refrigerant whose temperature and pressure are lower than those of an indoor air to be cooled (an expansion process DE_m). The low-temperature and low-pressurized liquid refrigerant is set to have a pressure p_2 (MPa abs) at a point Em. The liquid refrigerant having gotten pressure p_2 at point Em is sent to indoor heat exchanger 8, and indoor heat exchanger 8 absorbs heat from the indoor air so as to evaporate the refrigerant (an evaporation process EmAm). The evaporated refrigerant flows through passage 32 serving as the suction line of main compressor 2 so as to be supplied to main compressor 2. Namely, in evaporation process EmAm, pressure p_2 of the refrigerant is equaled to a refrigerant suction pressure P_m of main compressor 2, and the flow rate of refrigerant absorbed into main compressor 2 becomes G_m .

On the other hand, the supercooling liquid refrigerant branched into the branching passage is expanded so as to have the pressure and temperature which are lower than those of the liquid refrigerant at point C (an evaporation process DE_s). In this process, the temperature of the supercooling refrigerant is reduced from $t_2(^{\circ}\text{C})$ to $t_3(^{\circ}\text{C})$. A part of this

supercooled liquid refrigerant by supercooling heat exchanger 15 is branched into branching passage 27a so as to serve as the supercooling liquid refrigerant. The liquid refrigerant branched into branching passage 27a is set to have flow rate G_s .

[0021] In this process, the evaporation scale of the branched liquid refrigerant by expansion valve 22 (evaporation process DEs) is smaller than that of the liquid refrigerant by expansion valve 23 (evaporation process Dem) because of the following reason. The only requirement for supercooling the liquid refrigerant flowing in main passage 26 by the supercooling refrigerant branched into branching passage 27a is that the temperature of the supercooling refrigerant is lower than the liquid refrigerant (having the condition at point C) before being supplied to expansion valve 22. That is, the supercooling effect is ensured even when the pressure drop of the refrigerant having the condition at point D is such a small degree between pressure p_0 and pressure p_1 .

In supercooling heat exchanger 15, the supercooling refrigerant having the condition at point Es absorbs heat from the liquid refrigerant flowing in main passage 26, so as to supercool the liquid refrigerant flowing in main passage 26 (an evaporation process EsAs). The supercooling refrigerant, having finished supercooling, flows through passage 33 serving as the suction line of sub compressor 3 so as to be supplied to sub compressor 3.

[0022] In this regard, since a part of the liquid refrigerant flowing in main passage 26 is branched to have flow rate G_s , flow rate G_m of the liquid refrigerant supplied to indoor heat exchanger 8 is smaller than flow rate G_o of the whole refrigerant. However, since supercooling heat exchanger 15 supercools the undivided refrigerant before being branched, the endoergic capacity (cooling capacity) of the liquid refrigerant per unit mass and unit flow rate (kJ/kg) is increased so as to ensure or improve the cooling effect of outdoor heat exchanger 8.

In this way, the evaporation scale of expansion valve 22 for expanding the liquid refrigerant having flow rate G_s branched into branching passage 27a is smaller than that of expansion valve 23 for expanding the remaining liquid refrigerant having flow rate G_m remaining after being branched, so as to ensure a small pressure drop of the supercooling liquid refrigerant from pressure p_0 to pressure p_1 , thereby ensuring a high evaporation pressure p_1 in evaporation process EsAs. Namely, the evaporation pressure of the branched supercooling liquid refrigerant having flow rate G_s is higher than the evaporation pressure of the remaining refrigerant having flow rate G_m after being branched, thereby greatly reducing compression work ΔW_s required for compression process AsB in comparison with compression work ΔW_m required for compression process AmB. As a result, the compression work by sub compressor 3 is very small relative to the compression work by main compressor 2, thereby reducing the total compression work of the engine heat pump. The reduction of the compression work will be detailed. The basis of comparison is the total compression work of the engine heat pump with the whole refrigerant having flow rate G_o under only compression work ΔW_m . In other words, this is a total compression work in a refrigerant circuit with no sub compressor, i.e., with only a single compressor performing compression work ΔW_m for the whole refrigerant having flow rate G_o . This compression work is equal to the total compression work when the pressure drop of the supercooling liquid refrigerant having flow rate G_s branched into branching passage 27a in evaporation process DEs is set between pressure p_0 and p_2 .

[0023] When the whole refrigerant having flow rate G_o is compressed by only compression work ΔW_m , the total compression work (the basis of comparison) is calculated according to a formula 1.

$$\text{Formula 1: } G_o \cdot \Delta W_m = G_o \cdot (h_0 - h_2), G_o = G_m + G_s$$

[0024] Due to the above-mentioned small pressure drop of the supercooling liquid refrigerant having flow rate G_s branched into branching passage 27a from pressure p_0 to pressure p_1 , the total compression work of the engine heat pump of the invention is calculated according to a formula 2.

$$\text{Formula 2: } (G_m \cdot \Delta W_m) + (G_s \cdot \Delta W_s) = \{G_m \cdot (h_0 - h_2)\} + \{G_s \cdot (h_0 - h_1)\}$$

[0025] As a result, the reduction of compression work due to the small pressure drop of the supercooling liquid refrigerant having flow rate G_s branched into branching passage 27a from pressure p_0 to pressure p_1 , i.e., due to the high evaporation pressure of the refrigerant having flow rate G_s is calculated according to a formula 3.

$$\text{Formula 3: } G_s \cdot (\Delta W_m - \Delta W_s) = G_s \cdot (h_1 - h_2)$$

In this way, sub compressor 3 is driven by engine 4 so as to compress the supercooling refrigerant subjected to evaporation

pressure (pressure of refrigerant to be supplied to the compressor) which is higher than that for the refrigerant compressed by main compressor 2. Therefore, the engine heat pump requires no additional electric power that is required for a conventional engine heat pump with an electromotive sub compressor. Therefore, the total compression work in the refrigeration cycle is reduced while ensuring or improving the cooling capacity due to the supercooling performance of supercooling heat exchanger 15.

A capacity ratio between main compressor 2 and sub compressor 3 in the engine heat pump of the invention will now be described.

[0026] The capacity ratio between main compressor 2 and sub compressor 3 is referred to as a ratio between a delivery capacity of main compressor 2 and a delivery capacity of sub compressor 3. The delivery capacity of each of compressors 2 and 3 is determined based on its volumetric capacity and rotary speed. The volumetric capacity is a volume of refrigerant supplied to each of compressors 2 and 3 per cycle of a rotor of corresponding compressor 2 or 3 (cc/cycle). Since main compressor 2 and sub compressor 3 are driven by common engine 4, the rotary speed of each of compressors 2 and 3 is determined by a pulley ratio (speed ratio) of corresponding compressor 2 or 3 relative to a pulley of engine 4.

[0027] Accordingly, the delivery capacity of each of compressors 2 and 3 is produced by multiplying the volumetric capacity and the pulley ratio. When main compressor 2 has a volumetric capacity V_m and a pulley ratio U_m , and sub compressor 3 has a volumetric capacity V_s and a pulley ratio U_s , the delivery capacity of main compressor 2 is produced by $V_m \cdot U_m$, and the delivery capacity of sub compressor 3 is produced by $V_s \cdot U_s$. As a result, the capacity ratio of sub compressor 3 to the total capacity of main compressor 2 and sub compressor 3 (hereinafter, referred to as a sub compressor capacity ratio R) is calculated by the following formula.

$$R = (V_s \cdot U_s) / \{(V_m \cdot U_m) + (V_s \cdot U_s)\}$$

[0028] According to the formula, when volumetric capacities V_m and V_s of respective compressors 2 and 3 are equal to each other, sub compressor capacity ratio R is determined according to pulley ratios U_m and U_s of respective compressors 2 and 3 to engine 4. When pulley ratios U_m and U_s are equal to each other, sub compressor capacity ratio R is determined according to volumetric capacities V_m and V_s . In the present invention, the delivery capacity of sub compressor 3 is smaller than that of main compressor 2.

[0029] Sub compressor capacity ratio $R(\%)$ ranges between 20% and 29%. Description will be given of this range of sub compressor capacity ratio R .

In the refrigerant circuit of the engine heat pump, variation of sub compressor capacity ratio R is reflected in variation of the ratio of flow rate G_s of the supercooling liquid refrigerant branched from main passage 26 into branching passage 27a (in the cooling cycle) or 27b (in the heating cycle) to flow rate G_o of the whole liquid refrigerant. As sub compressor capacity ratio R increases, the ratio of flow rate G_s to flow rate G_o increases. As sub compressor capacity ratio R decreases, the ratio of flow rate G_s to flow rate G_o decreases.

[0030] The determination of range of sub compressor capacity ratio R between 20% and 29% according to the invention will be described. The following description is based on that the supercooling liquid refrigerant (having flow rate G_s) branched from main passage 26 into branching passage 27a or 27b is referred to as "branched liquid refrigerant", and the remaining liquid refrigerant (having flow rate G_m) flowing in main passage 26 after being branched is referred to as "main liquid refrigerant".

The reason why the upper limit of the range of sub compressor capacity ratio R is 29% will be described.

[0031] The upper limit 29% is determined based on variation of driving efficiency (energy efficiency) in the cooling cycle (for cooling). In this regard, when the engine heat pump is operated for cooling, as sub compressor capacity ratio R increases, flow rate G_s of the branched liquid refrigerant increases, i.e., the amount of the supercooling liquid refrigerant for supercooling the undivided liquid refrigerant having flow rate G_o flowing in main passage 26 increases so as to enhance the cooling capacity of the main liquid refrigerant per unit mass and unit flow rate. However, as flow rate G_s of branched liquid refrigerant increases, flow rate G_m of the main liquid refrigerant decreases so as to reduce the cooling capacity of indoor heat exchanger 8. This phenomenon defines the variation of driving efficiency (energy efficiency) for defining the upper limit of sub compressor capacity ratio R .

The upper limit 29% of sub compressor capacity ratio R is determined based on measurement data indicated as graphs of Fig. 4.

[0032] The graphs of Fig. 4 are defined by the axis of abscissas indicating sub compressor capacity ratio R and the axis of ordinate indicating coefficient of performance (COP) in the refrigeration cycle. COP is obtained by dividing the cooling or heating capacity by the quantity of consumed fuel. As COP increases, the driving efficiency (energy efficiency) is enhanced. A dotted graph indicates COP of a refrigerant circuit system with no sub compressor, i.e., with only a single compressor.

[0033] As noticed from the graph, under the cooling operation, when sub compressor capacity ratio R is close to 10%,

COP is larger than COP with the single compressor, and is almost kept constant while sub compressor capacity R increases. After increasing sub compressor capacity ratio R passes a value close to 15%, COP decreases according to increase of sub compressor capacity ratio R. When increasing sub compressor capacity ratio R reaches approximate 30%, COP under the cooling operation becomes smaller than COP by the single compressor. Namely, this value (approximate 30%) of sub compressor capacity ratio R is a critical value (upper limit) for the engine heat pump of the present invention to ensure the improvement of driving efficiency (COP) with the reduction of the total compression work under the cooling operation. That is, when sub compression capacity ratio R is smaller than approximate 30%, COP under the cooling operation is kept larger than COP by the conventional system. This is the reason why the upper limit of sub compressor capacity ratio R is set to 29% according to the present invention. Incidentally, COP under the heating operation is constantly larger than COP by the conventional system regardless of sub compression capacity ratio R. The reason why the lower limit of the range of sub compressor capacity ratio R is 20% will be described.

[0034] During the heating operation, connection point 15a serves as a refrigerant inlet for introducing the refrigerant from main passage 26 to supercooling heat exchanger 15, and connection point 15b serves as a refrigerant outlet for discharging the refrigerant from supercooling heat exchanger 15 to main passage 26. The lower limit 20% of sub compressor capacity ratio R is determined based on the relation of the refrigerant temperature at connection point 15a (hereinafter referred to as "inlet temperature") to the refrigerant temperature at connection point 15b (hereinafter referred to as "outlet temperature") during the heating operation. In this regard, when the engine heat pump is operated for heating, as sub compressor capacity ratio R decreases, flow rate Gs of the branched liquid refrigerant branched into branching passage 27b decreases, i.e., the amount of the supercooling liquid refrigerant for supercooling the undivided liquid refrigerant having flow rate Go flowing in main passage 26 decreases so as to reduce the supercooling effect of supercooling heat exchanger 15. Accordingly, the branched liquid refrigerant tends to be easily evaporated. However, as flow rate Gs of branched liquid refrigerant decreases, flow rate Gm of the main liquid refrigerant increases so that the undivided liquid refrigerant having flow rate Go is insufficiently supercooled by supercooling heat exchanger 15. That is, in supercooling heat exchanger 15, the outlet temperature rises while the inlet temperature is kept constant. This outlet temperature rising relative to the inlet temperature in supercooling heat exchanger 15 is obstacle to the sufficient supercooling action of supercooling heat exchanger 15. Consequently, to ensure a satisfactory performance of supercooling heat exchanger 15 under the heating operation, sub compressor 3 has to be selected (or configured) in its capacity for ensuring the enough supercooling degree, i.e., a difference of the inlet temperature of the supercooling refrigerant from the outlet temperature of the supercooling refrigerant after supercooling has to be not lower than a certain level (e.g., 5°C). The lower limit of sub compression capacity range R is set so as to ensure the enough supercooling degree.

The lower limit 20% of sub compressor capacity ratio R is determined based on measurement data indicated as graphs of Fig. 5.

[0035] The graphs of Fig. 5 are defined by the axis of abscissas indicating sub compressor capacity ratio R and the axis of ordinate indicating the inlet or outlet temperature (°C) of supercooling heat exchanger 15 under the heating operation.

[0036] As noticed from the graph, the inlet temperature of supercooling heat exchanger 15 is substantially kept constant (32-33°C) regardless of sub compressor capacity ratio R. On the other hand, as sub compressor capacity ratio R decreases, the outlet temperature of supercooling heat exchanger 15 increases from a value lower than the inlet temperature to a value higher than the inlet temperature. To ensure the sufficient performance of supercooling heat exchanger 15, the relation between the inlet temperature and the outlet temperature is preferably set so that the outlet temperature is lower than the inlet temperature by a temperature difference that is not less than approximate 5°C. A critic value (lower limit) of sub compressor capacity ratio R for reducing the outlet temperature lower than the inlet temperature by the difference that is not less than approximate 5°C is 20%. This is the

reason why the lower limit of sub compressor capacity ratio R is set to 20% according to the present invention.

As mentioned above, in the engine heat pump of the present invention, the upper limit of sub compressor capacity ratio R is determined in due consideration of the cooling operation, and the lower limit of sub compressor capacity ratio R is determined in due consideration of the heating operation, so that sub compressor capacity ratio R ranges between 20% and 29%, thereby ensuring or improving the cooling capacity during the cooling operation, and thereby ensure the performance of supercooling heat exchanger 15 during the heating operation. As a result, whether the engine heat pump is operated for cooling or heating, the engine heat pump according to the present invention ensures a good driving efficiency (energy efficiency) because common engine 4 drives main compressor 2 and sub compressor 3, and because sub compressor capacity range R ranges between 20% and 29%.

Incidentally, in the refrigerant circuit of the engine heat pump of the present invention, a continuously variable transmission (CVT) can be adapted to transmit power from engine 4 to main compressor 2 and sub compressor 3.

[0037] In this case, the CVT is set so as to change a speed ratio between main compressor 2 and sub compressor 3 in due consideration of the critic values of sub compressor capacity ratio R in the respective cases of cooling operation and heating operation.

In the engine heat pump of the invention, during the cooling operation, any sub compressor ratio R is allowed only if it does not exceed the upper limit, and during the heating operation, any sup compressor ratio R is allowed only if it is not smaller than the lower limit. Therefore, the CVT is configured so as to change the speed ratio due to whether the engine heat pump is operated for cooling or heating, so that sub compressor capacity ratio R under the cooling operation is

less than approximate 30%, and sub compressor capacity ratio R under the heating operation is not less than 20%.
[0038] Due to the CVT having this configuration, the freedom degree in setting volumetric capacity V_s and pulley ratio U_s relative to volumetric capacity V_m and pulley ratio U_m can be enhanced. Each of optimal values of sub compressor capacity ratio R for each of the cooling cycle and heating cycle is easily determined so as to improve the driving efficiency (energy efficiency) in either of the cooling cycle and the heating cycle, because only the upper limit has to be determined for the cooling cycle, and only the lower limit has to be determined for the heating cycle.

The engine heat pump of the invention is provided with engine exhaust recovery unit 6 in parallel to outdoor heat exchanger 5. The supercooling liquid refrigerant branched from

main passage 26 is evaporated by engine exhaust heat recovery unit 6, and compressed by sub compressor 3.

Engine exhaust heat recovery unit 6 is adapted to evaporate the endothermic branched liquid refrigerant having passed through supercooling heat exchanger 15 under the heating operation. In engine exhaust heat recovery unit 6, the branched liquid refrigerant exchanges heat with engine cooling water CW having a higher temperature than the branched liquid refrigerant, i.e., absorbs heat from engine cooling water CW so as to be evaporated.

The refrigeration cycle in the heating cycle represented in the Moliere chart (Fig. 3) will be described. Description of the same parts as the above-mentioned parts in the cooling cycle will be omitted.

[0039] The high-pressurized vapor refrigerants having been compressed by respective compressors 2 and 3 join each other at confluence point 65, and the joined refrigerant is sent to indoor heat exchanger 8. Indoor heat exchanger 8 condenses the high-pressurized vapor refrigerant so as to radiate heat from the refrigerant into the indoor space, whereby the refrigerant is cooled and liquefied (a condensation process BC). In this regard, point B defines the condition of refrigerant on the way from confluence point 65 to indoor heat exchanger 8.

The liquid refrigerant discharged from indoor heat exchanger 8 is supercooled in supercooling heat exchanger 15 by the supercooling refrigerant branched into branching passage 27b at the downstream side of supercooling heat exchanger 15 (a supercooling process CD).

A part of the supercooled liquid refrigerant flowing in main passage 26 is branched, and the remaining supercooled liquid refrigerant is expanded by expansion valve 21 so as to be changed into a low-temperature and low-pressurized liquid refrigerant (an expansion process DEm). The liquid refrigerant having the condition at point Em is sent to outdoor heat exchanger 5. In outdoor heat exchanger 5, the refrigerant absorbs heat from an outdoor air so as to be evaporated (an evaporation process $EmAm$). The vapor refrigerant flows in passage 32 serving as the suction line of main compressor 2 so as to be sucked to main compressor 2 again.

On the other hand, the supercooling liquid refrigerant branched into branching passage 27b is expanded by expansion valve 22 so as to have pressure and temperature lower than the pressure and temperature of the refrigerant at point C (an expansion process DEs). In this way, a part of the liquid refrigerant supercooled by supercooling heat exchanger 15 is branched into branching passage 27b so as to serve as the supercooling refrigerant having flow rate G_s .

In supercooling heat exchanger 15, the supercooling liquid refrigerant having the condition at point Es absorbs heat from the liquid refrigerant flowing in main passage 26 so as to supercool the liquid refrigerant flowing in main passage 26. The supercooling liquid refrigerant having passed through supercooling heat exchanger 15 is sent to engine exhaust heat recovery unit 6. In engine exhaust heat recovery unit 6, the supercooling liquid refrigerant exchanges heat with engine cooling water CW , i.e., absorbs heat from engine cooling water CW so as to be evaporated (an evaporation process $EsAs$). The evaporated refrigerant flows in passage 33 serving as the suction line of sub compressor 3 so as to be sucked into sub compressor 3 again.

In this way, in the engine heat pump under the cooling operation, the supercooling is performed so as to improve the driving efficiency (energy efficiency) by the following action.

[0040] The undivided liquid refrigerant having the total flow rate G_o flowing in main passage 26 is supercooled by supercooling heat exchanger 15. The supercooled liquid refrigerant increases its endothermic reaction capacity per unit mass and unit flow rate (kJ/kg). Therefore, when the supercooled liquid refrigerant reaches outdoor heat exchanger 5, the supercooled liquid refrigerant has an increased capacity per unit mass and unit flow rate for absorbing heat from the outdoor air. That is, the supercooled refrigerant can absorb heat as much as that absorbed by a non-supercooled refrigerant even if the supercooled refrigerant is less than the non-supercooled refrigerant. Therefore, flow rate G_m of the main liquid refrigerant sent into outdoor heat exchanger 5 under the heating operation can be reduced so as to reduce total flow rate G_o of the refrigerant circulating in the refrigeration cycle. As a result, the total compression work in the refrigeration cycle can be reduced so as to improve the driving efficiency (energy efficiency).

Since the branched supercooling liquid refrigerant is evaporated by engine exhaust heat recovery unit 6 provided in parallel to outdoor heat exchanger 5 and is compressed by sub compressor 3, the total compression work during the heating operation is reduced without requiring additional consumption of electric power while the total compression work

during the heating operation is reduced due to the above-mentioned range of sub compressor capacity ratio R.

[0041] Further, due to the supercooling of liquid refrigerant during the heating operation, the capacity of the refrigerant for absorbing heat from the outdoor air per unit mass and unit flow rate so as to reduce the quantity of whole refrigerant flowing in the refrigeration cycle.

As a result, the total compression work is reduced so as to enhance the driving efficiency (energy efficiency).

In the above-mentioned engine heat pump, main compressor 2 and sub compressor 3, which are driven by engine 4, may be operable individually. Due to this configuration, one or both of main compressor 2 and sub compressor 3 is/are selectively driven in correspondence to the rate of air-conditioning load, thereby improving the driving efficiency (energy efficiency).

In this case, as shown in Fig. 1, a main compressor clutch 42 is interposed between main compressor 2 and engine 4 so as to selectively drivingly connect or disconnect main compressor 2 to and from engine 4, and a sub compressor clutch 43 is interposed between sub compressor 3 and engine 4, so as to selectively drivingly connect or disconnect sub compressor 3 to and from engine 4.

[0042] Passage 33 serving as the suction line of sub compressor 3 to passage 32 serving as the suction of main compressor 2 through a connection passage 34 with an on-off valve 35. On-off valve 35 on connection passage 34 is opened to connect passages 32 and 33 to each other, and closed to separate passages 32 and 33 from each other, thereby forming a refrigerant circuit in correspondence to each of a low air-conditioning load rate, a middle air-conditioning load rate and a high air-conditioning load rate.

[0043] In this regard, as shown in Fig. 2, controller 25 is connected to main compressor clutch 42 and sub compressor clutch 43 so as to control the engagement and disengagement operation of each of clutches 42 and 43, and is connected to on-off valve 35 so as to control the opening and closing operation of on-off valve 35.

Due to this configuration, an example of the control of clutches 42 and 43 and valve 35 corresponding to load during either of the cooling operation and the heating operation is performed as follows. During the cooling operation, when the air-conditioning load rate is low, only sub compressor 3 is driven. When the air-conditioning load rate is middling, only main compressor 2 is driven. When the air-conditioning load rate is high, both compressors 2 and 3 are driven and supercooling heat exchanger 15 performs the supercooling operation. On the other hand, during the heating operation, when the air-conditioning load rate is low, only sub compressor 3 is driven. When the air-conditioning load rate is middling, only main compressor 2 is driven and engine exhaust heat recovery unit 6 performs the heat exchange. When the air-conditioning load rate is high, both compressors 2 and 3 are driven, supercooling heat exchanger 15 performs the supercooling operation, and engine exhaust heat recovery unit 6 performs the heat exchange.

[0044] With respect to the magnitude of air-conditioning load rate, the low air-conditioning load rate ranges between 0% and 15%, the middle air-conditioning load rate ranges between 15% and 60%, and the high air-conditioning load rate ranges between 60% and 100%. The engine heat pump is controlled for cooling as follows:

[0045] When the air-conditioning load rate is low, only sub compressor 3 is driven. In this case, controller 25 disengages main compressor clutch 42, and opens on-off valve 35. Therefore, the driving power of engine 4 is transmitted to only sub compressor 3, and passage 32 serving as the suction line of main compressor 2 is connected to passage 33 serving as the suction line of sub compressor 3, so that sub compressor 3 compresses the whole refrigerant having flow rate Go. Supercooling heat exchanger 15 is exercised or not depending on the control of opening and closing operation of expansion valve 22 for the supercooling heat exchanger. To exercise supercooling heat exchanger 15 for the supercooling, the pressure relation at a confluence point 64 (see Fig. 1) is considered to reduce the pressure loss. Controller 25 controls the opening degrees of respective expansion valves 22 and 23 so as to substantially equalize the pressure of refrigerant from passage 33 to the pressure of refrigerant from passage 32.

When the air-conditioning load rate is middling, only main compressor 2 is driven. In this case, controller 25 disengages sub compressor clutch 43, so as to transmit the driving power of engine 4 to only main compressor 2, so that main compressor 2 compresses the whole refrigerant having flow rate Go. To exercise supercooling heat exchanger 15 for the supercooling, controller 25 opens on-off valve 35 and controls the opening degrees of respective expansion valves 22 and 23, so that, at a confluence point 63 (see Fig. 1), the pressure of refrigerant from passage 33 is substantially equal to the pressure of refrigerant from passage 32.

When the air-conditioning load rate is high, both main and sub compressors 2 and 3 are driven, and supercooling heat exchanger 15 is exercised for the supercooling. In this case, controller 25 engages main compressor clutch 42 and sub compressor clutch 43, and closes on-off valve 35. Therefore, the driving power of engine 4 is transmitted to main and sub compressors 2 and 3, and simultaneously, passages 32 and 33 are separated from each other, so that main compressor 2 compresses the refrigerant having flow rate Gm, and sub compressor 3 compresses the supercooling refrigerant having flow rate Gs.

The engine heat pump is controlled for heating as follows:

[0046] When the air-conditioning load rate is low, only sub compressor 3 is driven. In this case, the control pattern with controller 25 is similar to the control during the cooling operation under the low air-conditioning load rate.

When the air-conditioning load rate is middling, only main compressor 2 is driven and engine exhaust heat recovery unit

6 performs the heat exchange. In this case, controller 25 disengages sub compressor clutch 43 to transmit the driving power of engine 4 to only main compressor 2, and opens on-off valve 35 to exercise engine exhaust heat recovery unit 6 for the heat exchange, so that main compressor 2 compresses the whole refrigerant having flow rate G_o joined at confluence point 63. Further, when supercooling heat exchanger 15 is exercised for the supercooling, controller 25 opens on-off valve 35 and controls the opening degrees of respective expansion valves 22 and 23, so that, at confluence point 63, the pressure of refrigerant from passage 33 is substantially equal to the pressure of refrigerant from passage 32. When the air-conditioning load rate is high, both main and sub compressors 2 and 3 are driven, supercooling heat exchanger 15 is exercised for the supercooling, and engine exhaust heat recovery unit 6 is exercised for the heat exchange. In this case, controller 25 engages main compressor clutch 42 and sub compressor clutch 43, and closes on-off valve 35. Therefore, the driving power of engine 4 is transmitted to main and sub compressors 2 and 3, and simultaneously, passages 32 and 33 are separated from each other, so that main compressor 2 compresses the refrigerant having flow rate G_m , and sub compressor 3 compresses the supercooling refrigerant having flow rate G_s under the heat exchanging by engine exhaust heat recovery unit 6.

In this way, one or both of main and sub compressors 2 and 3 is/are selectively driven in correspondence to the rate of required air-conditioning load, so as to reduce a load part of the engine heat pump where the fuel efficiency of engine 4 is reduced, thereby improving the driving efficiency (energy efficiency) of the engine heat pump.

Industrial Applicability

[0047] The present invention is broadly applicable to various engine heat pumps each of which has an engine and a compressor driven by the engine, so as to reduce the compression work of the engine heat pump without increase of consumed electric power, thereby improving the driving efficiency (energy efficiency).

Claims

1. An engine heat pump comprising:

an engine;
 a main compressor driven by the engine;
 a sub compressor delivering a refrigerant to be joined to a refrigerant delivered from the main compressor;
 an indoor heat exchanger;
 an outdoor heat exchanger;
 an expansion valve for the indoor heat exchanger;
 an expansion valve for the outdoor heat exchanger; and
 a supercooling heat exchanger disposed on a liquid refrigerant passage of a connection passage between the indoor heat exchanger and the outdoor heat exchanger, wherein, in the supercooling heat exchanger, a supercooling liquid refrigerant branched into a branching passage supercools a liquid refrigerant before being branched,
characterized in that the sub compressor is driven by the engine so as to compress the supercooling liquid refrigerant, and a ratio of a capacity of the sub compressor to a total capacity of the main compressor and the sub compressor ranges between 20% and 29%.

2. The engine heat pump according to claim 1, further comprising:

an engine exhaust heat recovery unit disposed in parallel to the outdoor heat exchanger, wherein the supercooling liquid refrigerant is evaporated by the engine exhaust heat recovery unit and compressed by the sub compressor.

FIG. 1

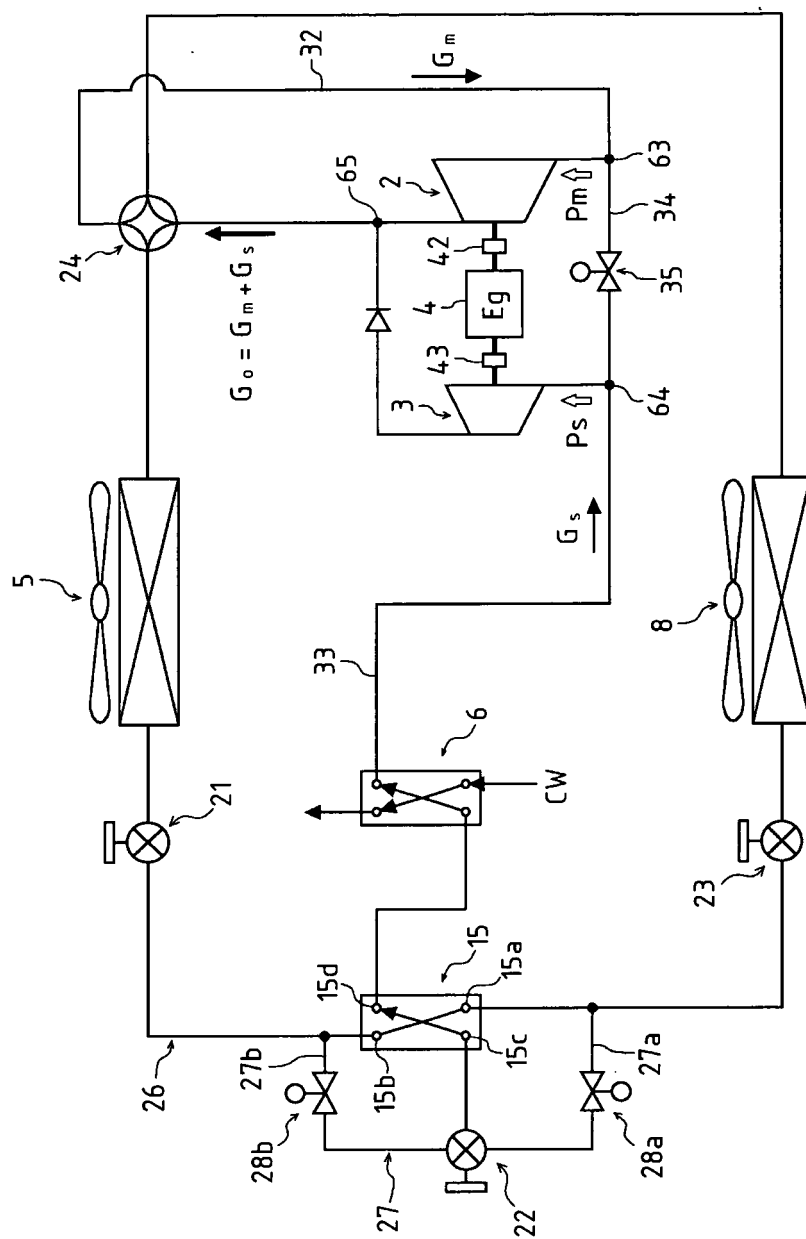


FIG. 2

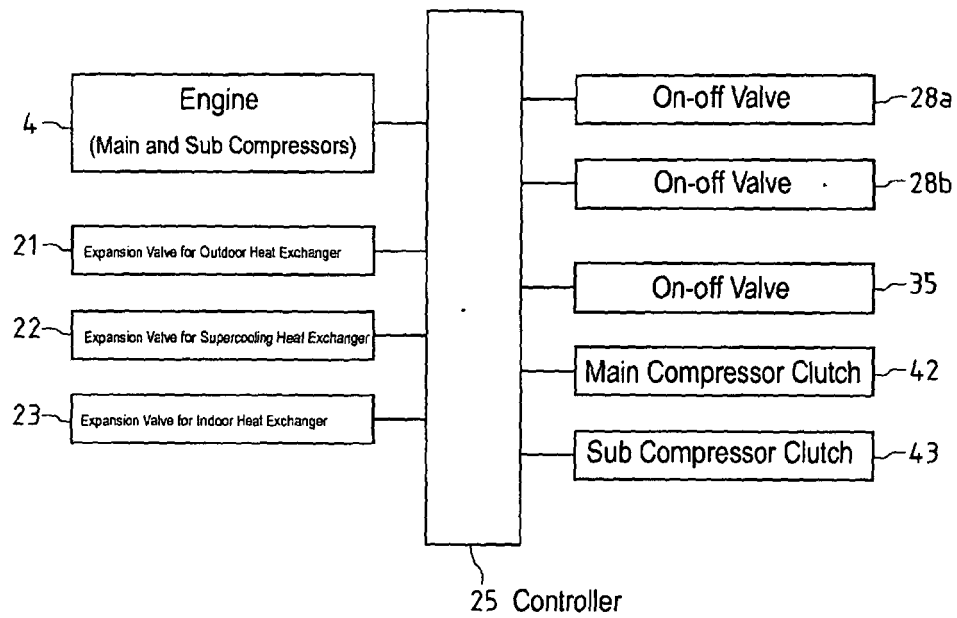


FIG. 3

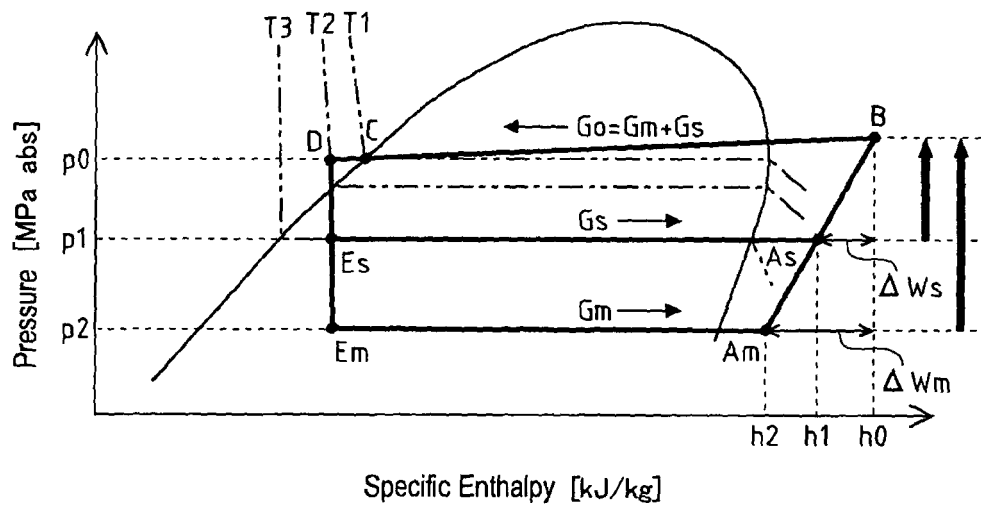


FIG. 4

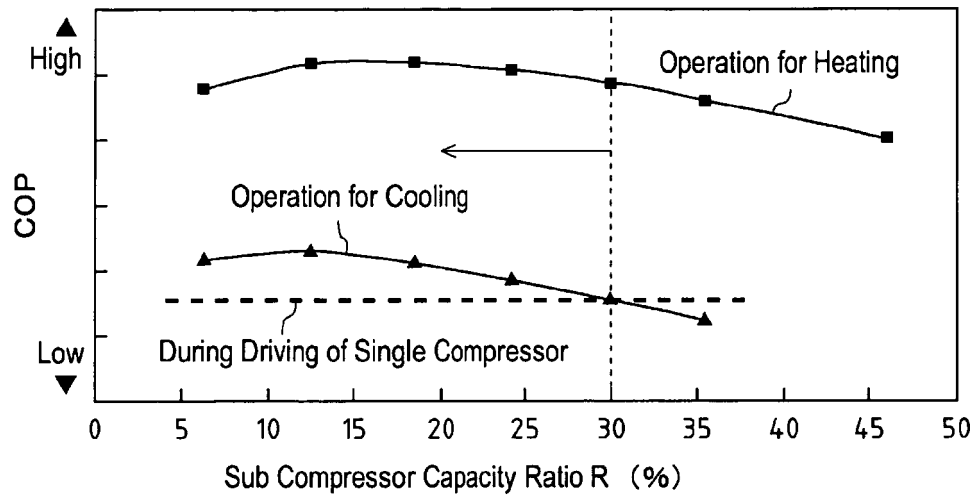
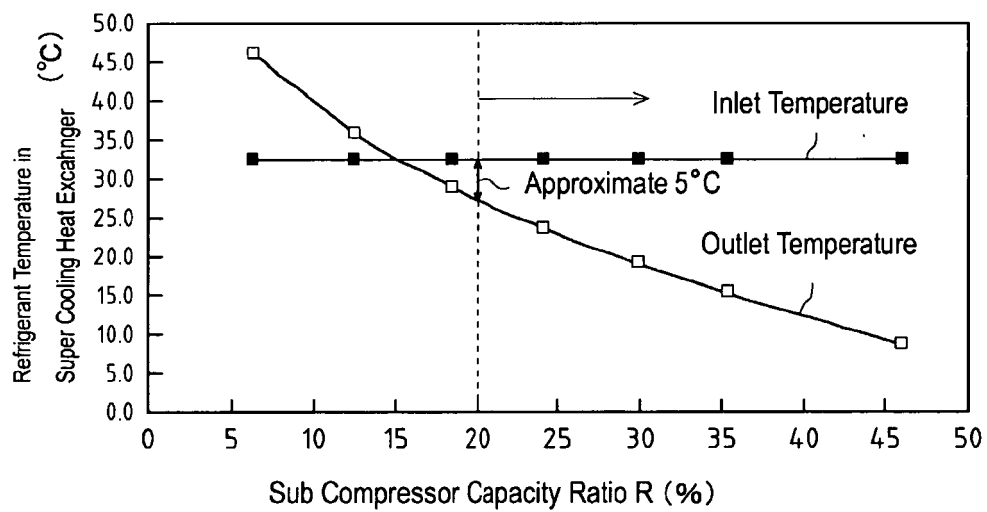


FIG. 5



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2005/007411

A. CLASSIFICATION OF SUBJECT MATTER

Int.Cl.⁷ F25B1/00, 27/00, 31/00

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

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

Int.Cl.⁷ F25B1/00, 27/00, 31/00

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

Jitsuyo Shinan Koho 1922-1996 Jitsuyo Shinan Toroku Koho 1996-2005
 Kokai Jitsuyo Shinan Koho 1971-2005 Toroku Jitsuyo Shinan Koho 1994-2005

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	JP 60-226669 A (Sanyo Electric Co., Ltd.), 11 November, 1985 (11.11.85), Page 1, lower right column, line 10 to page 2, upper left column, line 12; Figs. 1, 3 (Family: none)	1, 2
Y	JP 11-193966 A (Tokyo Gas Co., Ltd.), 21 July, 1999 (21.07.99), Par. Nos. [0020] to [0023]; Fig. 1 (Family: none)	1, 2
Y	JP 6-18121 A (Nippondenso Co., Ltd.), 25 January, 1994 (25.01.94), Par. No. [0003]; Fig. 4 (Family: none)	2



Further documents are listed in the continuation of Box C.



See patent family annex.

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"&" document member of the same patent family

Date of the actual completion of the international search

07 July, 2005 (07.07.05)

Date of mailing of the international search report

26 July, 2005 (26.07.05)

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REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 2004020153 A [0002]