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(54) **HEAT PUMP/AIR CONDITIONING APPARATUS WITH SEQUENTIAL OPERATION**

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Description

1. Field of invention

[0001] The invention relates to a system for providing a vapour compression cycle as for example an air-conditioning unit or a heat pump, with a thermal energy reservoir or storage that has a dual function, working either as an evaporator or as a gas cooler (condenser), and where the mode of operation depends on the temperature level of recurring temperatures of the energy source, the temperature of the energy storage, and the heat demand, all regulated to optimize heat production and to minimize power consumption. Furthermore the invention relates to a method for operating the system.

2. Description of the Prior Art

[0002] A conventional vapour compression cycle system for refrigeration, air-conditioning or heat pump purposes is shown in principle in Fig. 1. The system consists of a compressor 1, a condensing heat exchanger 2, a throttling valve or pressure reducing device 3 and an evaporating heat exchanger 4. These components are connected in a closed flow circuit 11, in which a refrigerant is circulated. The operating principle of a vapour compression cycle device is as follows: The pressure and temperature of the refrigerant is increased by the compressor 1, before it enters the gas cooler/condenser 2 where it is cooled and/or condensed, giving off heat. The high-pressure liquid is then throttled to the evaporator pressure by means of the pressure reduction device 3. In the evaporator 4, the refrigerant boils and absorbs heat from its surroundings. The vapour at the evaporator is drawn into the compressor 1, completing the cycle.

[0003] Conventional vapour compression cycle systems use refrigerants (as for instance R134A,) operating entirely at sub-critical pressure. A number of different substances or mixtures of substances may be used as a refrigerant. The choice of refrigerant is among other factors influenced by the condensation temperature, as the critical temperature of the fluid sets the upper limit for the condensation to occur. In order to maintain a reasonable efficiency it is normally desirable to use a refrigerant with critical temperature at 20-30°C above the condensation temperature. Near critical temperatures are avoided in design and operation of conventional systems, although some new systems operate near supercritical temperatures. This is for example the case for the heat pump described in UK patent application GB 2414289 A and in the patent application WO2005/106346 A1. Both of these applications describe the use of R410A as a refrigerant. A regulation method for transcritical heat pumping with R744 (CO_2) is described in patent EP 0 424 474 B2.

[0004] The present technology is treated in full detail in the literature and many patents cover this field of technology. The greenhouse gas effect of today's refrigerants pose a threat to the environment, as the refrigerant eventually will leak to the atmosphere. 1 kg of HFC refrigerant R410A released to the atmosphere corresponds to 1830 kg of CO_2 in global warming impact. R744 (CO_2) has a global warming potential of 1, whereas commonly used HFC refrigerants are from 1700 and up to more than 5000 kg CO_2 equivalent. It is therefore beneficial for the environment if R744 could be used as a refrigerant given that COP, (Coefficient of Performance) is as good as comparable HFC refrigerants. A lower COP will reduce the benefit by using R744 because CO_2 emissions from the power source increases. Some countries have made legislation that foresees a future ban on the use of strong greenhouse gases like the present HFCs for use in refrigeration processes. Environmental taxes are already levied on the use of HFC in Norway and in several other countries.

[0005] The COP for a heatpump that uses R744 (CO_2) is poor in a typical house-heating mode because of its low critical point of 31.2°C .This is thoroughly described in a Doctoral Thesis by Jørn Stene; Residential CO_2 heatpump systems for combined space heating and hot water heating (ISBN 82-471-6316-0). The increased CO_2 emissions from the energy source that powers the R744 refrigerant heat pump may outweigh the reduced green house gas effect from the potential release of HFC refrigerant to atmosphere. According to the Doctoral Thesis by Stene, the high-pressure hot R744 gas should reject usable heat well below the critical temperature of CO_2 (31,2°C) in order to achieve a good COP. This becomes difficult when indoor temperature is kept above 20°C and the media (water or air) used to heat space should have a temperature of at least 30°C to have a reasonable temperature difference for heat transfer. For heat to flow from the refrigerant to the heat distribution media the temperature of the refrigerant should thus be above 30°C. Cooling off the hot gas from the compressor high-pressure side in supercritical conditions to a level well below the critical point of CO_2 will increase the heat pump efficiency and particularly so when that heat is usable.

[0006] US 4,012,920 discloses a reversible heat pump that that has three coils to operate as either an evaporator or a condenser and for connecting either one of the other two coils to operate as a condenser or evaporator, respectively, so that heat can be exchanged in any combination between inside air, outside air and a storage fluid. However, the three coil arrangement can only work together two and two in cooling or heating mode, and never work with two of the coils performing as gas cooler/condenser simultaneously, which is essential for the principle of this invention when the heat storage is prepared for the next phase of operation.

[0007] US 3,523,575 disclose a reversible heat pump with a heat storage reservoir that can act both as help in cooling and in heating mode. However the heat pump has only two coils and the stored energy is only aimed at assisting in the evaporation/condensing process, not acting as the sole heat source for the heat pump.

[0008] From EP 1 811 246 it is known a heat pump according to the preamble of claim 1 employing CO2 as refrigerant and its operating method. The heat pump utilizes a heat source of natural water, e.g. well water, ground water, river water or sea water, effectively is applied to an air conditioning system in order to enhance heating/hot water supplying capacity and refrigeration capacity without requiring a large scale appurtenant facilities.

[0009] From WO 2008/037896 it is known a module that can be used for heat storage and transfer. The module includes a refrigerant compressor, a heat exchange/storage block located on a delivery side of the compressor, a heat exchange/storage medium-temperature block, another heat exchanger, or preferably a heat exchange/storage low-temperature block.

3. The object of this invention

[0010] There is a constant strive towards maximizing the output from the vapour compression cycle and minimizing the primary energy input to it. Bettering the components of the system e.g. heat transfer efficiency in condensing and evaporating heat exchangers, reduction in compressor losses and reduced throttling losses are areas where improvements of efficiency are made.

[0011] It is an object of the present invention to provide a new, simple and effective way of improving the overall efficiency of the vapour cloud compression cycle by using a heat storage as heat source at times when the temperature of the external heat source is low and to heat (load) the heat storage when the temperature of the external heat source is high and to increase gas cooling/condensing of the refrigerant by arranging for preheating of sanitary water when the heat storage serves as heat source.

[0012] The present invention is especially designed for a vapour compression cycle that uses CO2 (R744) as working fluid in transcritical refrigeration.

[0013] Still other objects of the present invention is to reduce noise from heat pumping by eliminating air and fan noises at certain times, to reduce time for de-icing of the evaporator that uses air as energy source and to increase longevity of compressor through more stable compressor load. There will be less use of the electrical resistance heater that is often placed in chassis of the outdoor heat pump unit because it can be turned off in the operating mode where the heat storage provides evaporation heat. Furthermore it is an objective to increase the feasibility of harnessing thermal energy from the sun. The current invention improves the efficiency of thermal solar collectors when they are heating a heat reservoir or storage, because they can feed usable heat to the system at low water temperatures. Still another object of the invention is to increase the heat pump work by heating a bigger portion of the warm water that is consumed. A two tank system with different temperature levels in the tanks should preferably be incorporated in the system, although it is also possible to use other tank arrangements. The dual temperature tank system provides an option to preheat parts of sanitary hot water at times when it is beneficial for the overall compression cycle in one of the tanks, and to blend this water with hot water from the other tank when consumption of warm sanitary water takes place. To achieve the object a system according to claim 1 and a method according to claim 4 are provided.

[0014] The present invention involves the control or regulation of energy flow between the heat storage and the refrigerant, the time for heating sanitary hot water, the room heating and for controlling when the evaporation heat is taken from the environment. This regulation is typically performed by valve regulation by actuation of valve positions, and by regulation of warm water production.

[0015] Regulation is based on the pattern of recurring temperatures of the environment, heat storage energy level, and the room heating and warm water needs. A control unit for controlling or regulating the system may include common control circuits and sensors.

4. General description of the invention

[0016] Accordingly, the present invention concerns a system for providing a vapour compression cycle. The system includes a flow loop or circuit with a compressor, a first heat exchanger downstream of the compressor, a second heat exchanger downstream of the first heat exchanger, a third heat exchanger downstream of the second heat exchanger and a first pressure reduction device downstream of the third heat exchanger, a fourth heat exchanger with a heat storage device or reservoir downstream of the first pressure reduction device, a second pressure reduction device downstream of the fourth heat exchanger, a fifth heat exchanger downstream of the second pressure reduction device and the flow loop is then connected back to the compressor completing the loop. The pressure reduction devices are common devices for throttling frequently used within the field of heat pumps and refrigeration circuit and may include expansion valves that are fixed or adjustable. Expansion valves may include thermodynamic energy expansion valves such as diaphragm electromagnetism valves, straight close valves and right angle close valves.

[0017] A bypass line with a shutoff valve, bypasses the fifth heat exchanger, and is connected at a first end between the fourth heat exchanger and the second pressure reduction device, and at a second end between the fifth heat exchanger and the compressor. A control unit controls at least the shutoff valve and the pressure reduction devices.

[0018] The first heat exchanger may be in heat exchange relationship with a high temperature water tank, the second heat exchanger may be in heat exchange relationship with a space (room) heating device and the third heat exchanger may be in heat exchange relationship with a water tank for preheating sanitary water.

[0019] A four way valve may be placed over the inlet and outlet of the compressor for switching between heating modes and cooling modes. A thermal solar panel may be connected to the heat storage tank and to one or both of the sanitary hot water tanks.

[0020] The refrigerant may be CO2.

[0021] Furthermore the invention includes a method for controlling the vapour compression cycle with the system defined above wherein opening the first pressure reduction device, closing the shutoff valve, and regulating the second pressure reduction device prepares a first heating mode, and where regulating the first pressure reduction device, with the second pressure reduction device or the bypass valve closed, prepares a second heating mode.

[0022] It is an essential feature that the system allows switching between first and second heating modes. The two modes are generally governed by outdoor temperature and the time of the day.

[0023] The heat exchanger connected to the heat storage may act as an evaporator when the ambient temperature of the fifth heat exchanger is at a low level, and it may act as gas cooler when the ambient temperature is at a high level.

[0024] The preheating of the sanitary water in a low temperature water tank should correspond with the use of the heat storage as an evaporator.

4. 1 Description of drawings:

[0025]

Fig. 1 shows a conventional vapour compression cycle device.

Fig. 2 shows the process cycle of this invention.

Fig. 3 shows typical data for outdoor temperature in Oslo winter.

Fig. 4 and 4b shows an embodiment of the present invention for room heating, hot water heating, hot water preheating and sanitary warm water outtake.

Fig. 5 and 6 shows log p H diagrams of CO2 to illustrate the process cycles.

Fig. 7 shows water flow in a two tank dual temperature solution.

5. Basic description

[0026] The invention will now be described in more detail, in the following referring to Fig 2.

[0027] The closed working fluid circuit consists of a refrigerant flow loop (11) where five heat exchangers are connected in series. The five heat exchangers are numbered (2h), (2r), (2p), (4) and (6). Heat exchangers (6) and (4) have a pressure reducing device upstream, numbered (5) and (3) respectively, enabling control of the pressure and temperature at the various sections of the flow loop. Further the flow loop has a bypass line with a shutoff valve (8) and a compressor (1). The fourth heat exchanger (6) allows the refrigerant to exchange heat with the heat storage medium in tank/closed compartment (7) at temperature (T1).

[0028] A regulator (14) governs the shown flow loop with its two modes of heating operation. Adjustment of the pressure reducing devices (5) and (3), and the position (shut or open) of the valve (8) in the bypass line determines if the operating mode one heating or operating mode two heating is to be used.

5.1 Operating mode one heating. Ref. Fig.2

[0029] Operating mode one heating and operating mode two heating of the present invention is used when the purpose of the apparatus is to heat an environment /building/water etc. Operating mode one heating is used when the temperature (T2) of the external environment of fifth heat exchanger (4) is at a high level in its cycle. If outdoor ambient air is the external environment (air is used as heat source), then it is likely that operating mode one heating would be during daytime, because the outdoor air temperature (T2) is systematically (but not always) higher during daytime than at night. (Fig. 3 shows the temperature measured each hour during a typical winter period in Oslo.) The pressure reduction device (5) upstream the fourth heat exchanger (6) can be set fully open, and the bypass line shutoff valve (8) is then closed. The second pressure reduction device (3) regulates the pressure level in the first heat exchanger (2h) and the second heat exchanger (2r) and the third heat exchanger (2p) and the fourth heat exchanger (6). The refrigerant boils off in the fifth heat exchanger (4). A compressor (1) increases the pressure and temperature of the refrigerant gas. Downstream the compressor (1), the refrigerant rejects heat in the first heat exchanger (2h) to the hot water tank and second heat exchanger (2r) to a heat distribution medium. The medium could be water or air. The refrigerant then passes the fully open the first pressure reduction device (5) and flows into the fourth heat exchanger (6) where heat in the refrigerant is rejected to a heat storage medium that could be water (or ice) in the heat storage (7) The high-pressure refrigerant is then throttled in the second pressure reduction device (3) before it flows to the fifth heat exchanger (4) and the flow circuit is complete.

5.2 Operating mode two heating. Ref. fig. 4

[0030] Operating mode two heating is used when the temperature (T2) of the external environment of the fifth heat exchanger (4) is a low point in its cycle. If outdoor air is used as heat source for the fifth heat exchanger (4), then it is likely that operating mode-two heating is at night time ref. Fig 3. The second pressure reduction device (5) is now shut and the bypass line shutoff valve (8) is open. (The shutoff valve (8) could be closed and the second pressure reduction device (5) could be set fully open if outdoor temperature (T2) is high enough to contribute to the evaporation.) The first pressure reduction device (5) is regulating pressure level in heat exchangers upstream of it. These valve positions make the media in heat storage (7) to the heat source for evaporation of the refrigerant. The fourth heat exchanger (6) enables the heat storage media to be the heat source that boils off the refrigerant. Compressor (1) sucks the vapour from the fourth heat exchanger (6) via the bypass line and raises the pressure and temperature of the refrigerant gas as it pumps the refrigerant in the refrigerant cycle. Downstream of the compressor (1), the refrigerant rejects heat in the second heat exchanger (2r) and (2p). Refrigerant pressure and temperature is throttled in the first pressure reduction device (5) to condensate in the fourth heat exchanger (6) where evaporation takes place and the cycle is complete.

5.3 Gains in using the two modes.

[0031] In a 24 hours period, through one day and one night, the gain of the arrangement described is that the nighttime evaporation temperature is increased by (T1) minus (T2). If the media in the heat storage is water it can be designed to have a lower temperature limit of approximately zero deg C. That is because the water in the heat storage has a temperature of zero deg. C until all of the water is frozen to ice. With a temperature differential of 5°C that can be quite normal in the northern hemisphere at wintertime, an improvement of COP for the process cycle of 12.5 percent can be anticipated. (According to Stene, a rise in evaporation temperature of 1°C will increase COP by 2.5 percent.)

[0032] The fourth heat exchanger (6) used at nighttime is virtually noiseless compared to the fifth heat exchanger (4) that uses forced air flow as heat source. Silence through the night is important for the use of any apparatuses in densely populated areas.

[0033] Ice build up at fins of the heat exchanger is a problem, because it reduces the efficiency of the heat transfer and de-icing is required when ice build-up become too severe. De-icing consumes energy, it produces water and it may affect longevity of the equipment as it implies temperature fluctuations in piping and increased valve switching. The present invention reduces problems related to de-icing to daytime.

6. Preferred embodiment (Fig 4)

[0034] The preferred embodiment of the invention is shown in Fig 4. This embodiment includes two hot water tanks, (9h) (hot water 27-65°C) and (9p) (preheating 7 - 27°C), in addition to a room heating device (Rhd) and the three flow adjustable circulation pumps (Ph) (hot water), (Pr) (room heating), (Pp) (preheating). The purpose of using two hot water tanks is to be able to separate the production of hot water at two different temperature levels, one temperature level for each operating mode. Heating of hot water can then take place at times when the physical state of other elements in the refrigerant flow circuit is benign for this purpose. Another benefit of using two tanks is that more water is heated by the heat pump compared with a traditional tank solution. Fig. 7 shows water volumes heated with a traditional one tank solution compared with a two tank dual temperature solution where warm sanitary tap water consists of hot water from the hot water tank tempered with preheated water from the low temperature water tank.

6.1 Operating mode one heating (Fig. 4)

[0035] In operating mode one heating, hot refrigerant gas from the compressor (1) is in heat exchange relationship with water, being circulated from the bottom of water tank (9h) - through the first heat exchanger (2h) and back to the top of water to tank 9h. The water is heated from app. 27°C to 65-90°C depending on refrigerant pressure and hot water temperatures and circulation rate. The heating capacity is regulated by means of the hot water circulation flow rate, the compressor (1) discharge pressure and flow rate.

[0036] Downstream of the first heat exchanger (2h), hot refrigerant gas is in heat exchange relationship with a conditioning fluid for room heating in the second heat exchanger (2r). Temperature levels of the conditioning fluid will in most cases vary between 27 and 45°C depending on local room heating systems. The heating capacity is regulated by the conditioning fluid flow rate, and the temperature and flow rate of the refrigerant hot gas.

[0037] The high-pressure refrigerant gas then flows through a third heat exchanger (2p) where no heat is rejected (there is no circulation of water in the heat exchanger (2p) in this mode). The refrigerant gas then flows further through the fully open pressure reduction device (5) before the hot refrigerant gas rejects heat to the media in heat storage (7) by means of the fourth heat exchanger (6). Bypass line shut off valve (8) is kept closed. Downstream the fourth heat exchanger (6) the refrigerant gas is flowing through the second pressure reduction device (3) where pressure is throttled whereafter liquid refrigerant flows to the fifth heat exchanger (4) where evaporation takes place before the refrigerant gas is sucked into the compressor (1) completing the cycle. Energy for heating of hot water and room heating must be adjusted as to fit with the compressor capacity. Generally the temperature of the water in hot water tank (9h) should be kept at set point during the period of operating mode one heating. Whenever hot water is consumed in this operating mode one heating, preheated water from tank (9p) enters tank (9h). (Ph) starts circulation through (2h) in order to heat the preheated water until the hot water tank is at set temperature again. Circulation rate should be adjusted so that outlet temperature of refrigerant from (2h) is higher than water/air inlet temperature of the second heat exchanger (2r). The system should be designed so that at the end of operating mode one heating, the water in tank (9p) should be at a temperature as close to city water temperature as possible, i.e. all preheated water should preferably have been consumed.

6.2 Operating mode two heating (Fig 4)

[0038] In operating mode two heating, shutoff valve (8) opens, the second pressure reduction device (5) closes and pressure reduction device (5) is operational. In this mode of operation, heat storage fluid in tank (7) serves as heat source to evaporate the refrigerant. The latent heat of the heat storage fluid is transferred to the refrigerant by the fourth heat exchanger (6) where liquid refrigerant boils off to form vapour. The vapour is sucked into the compressor (1). Compressor (1) raises pressure and temperature in the circulating refrigerant gas. The refrigerant passes through the first heat exchanger (2h) without rejecting heat as (Ph) is off in this mode of operation. Downstream the first heat exchanger (2h) hot refrigerant gas is in heat exchange relationship with a conditioning fluid for room heating in the second heat exchanger (2r). Temperature levels of the conditioning fluid will in most cases vary between 25 and 45°C depending on local room heating systems. The heating capacity is regulated by means of the conditioning fluid flow rate ((Pr) running speed) and flow and temperature of the refrigerant hot gas. The refrigerant gas then passes a third heat exchanger (2p) in which water to tank (9p) is circulated by means of (Pp). Water circulates from the bottom of the tank, via the heat exchanger (2p) where water is in heat exchange relationship with the refrigerant gas, and back to the top of the tank (9p). This way, water is preheated from mains water temperature of app. 7°C to app. 27°C. The rate of preheating is regulated by the water flow rate of (Pp). Cold water flow from (9p) is regulated to achieve maximum gas cooling of the refrigerant. That means that the flow should be adjusted as to use the entire period of operating in mode two heating for preheating of sanitary water. After leaving heat exchanger (2p) the high-pressure refrigerant gas is throttled in the first pressure reduction device (5) wherafter liquid refrigerant flows to the fourth heat exchanger (6) completing the cycle.

[0039] At the end of the nighttime period the temperature of heat storage medium in heat storage (7) will be lowered to a level where ice may have been formed given the heat storage medium was water. With a good heat transfer mechanism in the fourth heat exchanger (6) the whole tank may freeze.

[0040] This preferred embodiment of the invention shows that a controlled running of flow from the circulation device Ph, Pr and Pp in the different operating modes can provide gas cooling in operating mode two heating. Proper dimensioning of the hot water tanks (9h) and (9p) will assure enough daily hot water to a normal family dwelling.

[0041] The media that is used to boil off the refrigerant in operating mode two heating could be water or another phase change material. The phase change from liquid to solid should be facilitated in the energy storage (7) in order to increase the amount of energy that can be stored in a limited volume and also to get a stable evaporation temperature. Melting point for water is 0°C and freezing energy is 334 kJ/kg. A 300 litre tank contains app. 28 kWh for evaporation, which should be sufficient for a normal apartment. However, other tanks could be used and phase change may then be unnecessary. A 3000 litres tank (normal size for an indoor/outdoor oil storage tank) contains 52,5 kWh when water is cooled from 15°C to zero.

[0042] The heat storage media in heat storage (7) provides for gas cooling in operating mode one heating. This is usable heat as long as T1 >T2 in operating mode two.

7. Physics

[0043] Fig. 5 shows a pressure enthalpy diagram of a transcritical vapour compression cycle. In a transcritical vapour cycle the pressure and enthalpy of the hot gas from the discharge of compressor (1) (Fig. 1) is at state **a** (Fig 5). After giving off heat to a cooling agent e.g. hot water in (2) at constant pressure the refrigerant is cooled to state **b.** Throttling valve (3) (Fig. 1) takes the refrigerant to a two-phase gas/liquid mixture shown as state **c** (fig. 5). The throttling is a constant enthalpy process. The refrigerant absorbs heat in the fifth heat exchanger (4) (Fig. 1) by evaporating the liquid phase bringing it to state **d** (fig. 5) at the fifth heat exchanger (4) (Fig. 1) outlet, the refrigerant enters the compressor (1) (Fig. 1) making the cycle complete.

7.1 Operating mode one heating (Fig 5)

[0044] In operating mode one heating, the state of the refrigerant at outlet of compressor (1) (Fig. 2) is at **a.** The refrigerant is giving off heat to hot water in the first heat exchanger (2h) and to room heating media in the second heat exchanger (2r) (Fig. 2), bringing the refrigerant to state **b** at the inlet of the fourth heat exchanger (6) (Fig. 2). The refrigerant is further cooled, giving off heat to a suitable medium in heat storage (7) (Fig. 2), taking the refrigerant to state **b'** at the fourth heat exchanger (6) (Fig. 2) outlet. The state of the refrigerant in the heat rejection phase before throttling, is then moved from **b** to **b'.** The enthalpy difference **b-b'** represents the energy per unit of refrigerant flow that is available for storage in heat storage (7) (Fig. 2). From **b'** the refrigerant is throttled to point **c'.** The point **c'** represents evaporation pressure and temperature at the actual temperature (T2). The enthalpy **c'-c** is equivalent to **b-b'** and shows how the stored energy is harvested from the environment. The refrigerant absorbs heat in fifth heat exchanger (4) (Fig. 2), and moves from state **c'** to state d before it enters the compressor (1) and the cycle is complete.

7.2 Operating mode two heating (Fig 6)

[0045] Fig. 6 shows a log pressure enthalpy diagram of a transcritical vapour compression cycle. Operating mode two heating is represented by points a, b", c", d. Operating mode two heating is run when the temperature (T2) (Fig. 2) is at a low point and the temperature of the heat storage media (T1) is high (after a period where the media in heat storage (7) has been used to cool the gas). Temperature (T1) could be between 0 and 20°C given the heat storage media is water and T1 should be greater than T2. The refrigerant status at outlet of compressor (1) (fig. 2) is at state **a.** After rejecting heat in the second heat exchanger (2r) the state of the refrigerant would be at point **b** and the state of the refrigerant leaving heat exchanger (2p) (Fig. 2) would be at **b".** The preheating of hot water brings the refrigerant from b to b". The first pressure reduction device (5) (fig. 2) lets the pressure of the refrigerant down to point **c"** at constant enthalpy. Heat from the media in heat storage (7) (fig. 2) is used to boil off the refrigerant in the fourth heat exchanger (6) (fig. 2) bringing the refrigerant to state **d.** The fifth heat exchanger (4) (fig. 2) is bypassed and the state of the refrigerant is at state **d** as it is sucked into the compressor (1) (fig. 2) completing the cycle. Energy for preheating of hot water in heat exchanger (2p) is then represented by the enthalpy difference **c** - **c".**

[0046] Point **c'** shows the evaporation pressure if the heat source were at (T2) (fig. 2) assuming that T1 > T2 and that no preheating of hot water in (2p) took place. Point **d'** is the corresponding state of refrigerant gas at compressor inlet.

[0047] The gain of this operating mode is that the evaporation temperature is lifted from **c'** to **c**, thus reducing the compressor work with **(a-d')** - **(a-d)** and the energy taken from the heat storage is increased by the enthalpy **(d-c")** - **(d-c').**

8. Use of a two tank dual temperature hot water system ref. fig. 4 and fig. 7

[0048] Fig. 7 shows differences in the amounts of water being heated by the heat pump when heating 100 litres of water for use at 40°C, when using two tanks at two temperatures for sanitary warm water supply compared to a conventional one tank system.

[0049] In operating mode one heating, the hot refrigerant gas in the first heat exchanger (2h) rejects heat at temperatures up to 90°C to a separate hot water tank (9h). Pump speed of circulation pump (Ph) governs the energy transfer and temperature approach of hot water in the first heat exchanger (2h). In operating mode one, circulation pump (Pp) is off, and no preheating of hot water is done in heat exchanger (2p). After giving off heat to room heating media in the second heat exchanger (2r), the hot refrigerant gas flows right through the heat exchanger (2p) before it goes to the fourth heat exchanger (6), where remaining heat is given off to thaw/heat the medium in the heat storage (7).

[0050] In operating mode two heating, the hot water circulation pump (Ph) is off and the hot refrigerant gas flows right through the first heat exchanger (2h) without giving of any heat, before it enters the second heat exchanger (2r) and gives off heat to a room heating media. After giving of heat to room heating media, the hot refrigerant gas flows to heat exchanger (2p) where heat is given off to water circulating from tank (9p). Energy outtake is regulated by means of circulation pump (Pp).

[0051] Tempered water from tank (9p) should be used to blend with hot water from tank (9h) before use. More of the sanitary water can then be heated by the heat pump at lower temperature than what is the case for traditional system. This is shown in fig. 7.

9. Solar thermal heating

[0052] A flow loop from a solar thermal collector may be connected to the heat storage tank (7). The fluid from the solar thermal collector in heat exchange relationship with the media in the heat storage (7) will then help thaw and heat the heat storage medium. In a conventional solar thermal system the differential temperature between ambient temperature and heat transfer fluid is relatively high in the winter season. A typical temperature differential of 50 - 60°C is common. A high temperature differential reduces the efficiency of the heat absorber because of radiation losses and convection losses in the absorber. Because of the low temperature requirement for thawing ice and raising temperature above zero degrees in the heat storage, wintertime efficiency of thermal collector increases with up to 50 percent compared to traditional systems.

[0053] In summer operation the solar thermal collector can produce hot water for sanitary use directly to the water tanks.

10. Cooling ref fig. 4b

[0054] For operating mode cooling, a four way valve 12 is introduced downstream of compressor (1). By rerouting the refrigerant flow, refrigerant heat may be dumped in the fifth heat exchanger (4) or in the fourth heat exchanger (6) depending on ambient temperature and actual temperature in the heat storage media in the heat storage tank (7).

[0055] When shutoff valve (8) is closed, heat is first dumped to ambient air through heat exchanger (4). Depending on the room cooling needs and temperature of the heat storage media in tank (7), the second pressure reduction device (3) or the first pressure reduction device (5) may be used to reduce the pressure to condensate the refrigerant to the second heat exchanger (2r) where room cooling media is in heat exchange relationship with the refrigerant. Circulation pumps (Pp) and (Ph) are normally stopped in this mode of operation.

Claims

1. A system including a high temperature water tank (9h), a room heating device (Rhd) and low temperature water tank (9p), and for providing a vapour compression cycle with two separate heating modes of operation, comprising a compressor (1);
a first gas cooling heat exchanger (2h) downstream of compressor (1);
a second gas cooling heat exchanger (2r) downstream of the first heat exchanger (2h); a third gas cooling heat exchanger (2p) downstream of the second heat exchanger (2r);
a first pressure reduction device (5) downstream of the third heat exchanger (2p) **characterised by**:

 a fourth heat exchanger (6) with a heat storage device (7) downstream of the first pressure reduction device (5);
 a second pressure reduction device (3) downstream of the fourth heat exchanger (6);
 a fifth heat exchanger (4) downstream of the second pressure reduction device (3) connected back to the compressor (1);
 a bypass line with a shutoff valve (8) bypassing the fifth heat exchanger (4),
 connected at a first end between the fourth heat exchanger (6) and the second pressure reduction device (3), and at a second end, between the fifth heat exchanger (4) and the compressor (1);
 at least one control unit (14) for controlling at least the shutoff valve (8) and the first pressure reduction device (5) and the second pressure reduction device (3) wherein the first heat exchanger (2h) is connected to the high temperature water tank (9h);
 the second heat exchanger (2r) is connected to the room heating device (Rhd); and
 the third heat exchanger (2p) is connected to the low temperature water tank (9p).

2. The system according to claim 1 where a thermal solar heat source is connected to the heat storage device (7) and to high temperature water tank (9h) and/or low temperature water tank (9p).

3. The system according to claim 1 to 2, wherein the refrigerant is CO2.

4. A method of controlling a vapour compression cycle in a system according to claim 1, allowing switching between a first mode of heating operation and a second mode of heating operation, wherein in the first mode of operation the method is **characterized by** the steps of:

 rejecting heat from a first heat exchanger (2h), a second heat exchanger (2r), and a fourth heat exchanger (6), whereby the first second and fourth heat exchanger are operating as gas coolers, and wherein the fourth heat exchanger (6) rejects heat to a heat storage device (7); and
 operating a fifth heat exchanger (4) as an evaporator.

5. A method of controlling a vapour compression cycle in a system according to claim 4, allowing switching between a first mode of heating operation and a second mode of heating operation, wherein in the second mode of operation the method is **characterized by** the steps of:

 rejecting heat from a second heat exchanger (2r) and a third heat exchanger (2p) whereby the second and third heat exchanger are operating as gas coolers; and
 operating a fourth heat exchanger (6) as an evaporator that absorbs heat from a heat storage device (7).

6. The method of controlling a vapour compression cycle according to claim 4 or 5 further including the step of:
governing the two modes of operation according to a heat source temperature (T2) and the time of the day.

Patentansprüche

1. System mit einem Hochtemperaturwassertank (9h), einer Raumheizvorrichtung (Rhd) und einem Niedrigtemperaturwassertank (9p) und zum Bereitstellen eines Dampfverdichtungskreislaufs mit zwei separaten Heizbetriebsmodi, umfassend einen Kompressor (1);
einen ersten Gaskühlungswärmetauscher (2h) stromabwärts des Kompressors (1);
einen zweiten Gaskühlungswärmetauscher (2r) stromabwärts des ersten Gaskühlungswärmetauschers (2h);
einen dritten Gaskühlungswärmetauscher (2p) stromabwärts des zweiten Wärmetauschers (2r); eine erste Druckreduzierungsvorrichtung (5) stromabwärts des dritten Wärmetauschers (2p), **gekennzeichnet durch**:

 einen vierten Wärmetauscher (6) mit einer Wärmespeichervorrichtung (7) stromabwärts der ersten Druckreduzierungsvorrichtung (5);
 eine zweite Druckreduzierungsvorrichtung (3) stromabwärts des vierten Wärmetauschers (6);
 einen fünften Wärmetauscher (4) stromabwärts der zweiten Druckreduzierungsvorrichtung (3), der wieder mit dem Kompressor (1) verbunden

ist;
eine Umgehungsleitung mit einem Absperrventil (8), die den fünften Wärmetauscher (4) umgeht und an einem ersten Ende zwischen dem vierten Wärmetauscher (6) und der zweiten Druckreduzierungsvorrichtung (3) und an einem zweiten Ende zwischen dem fünften Wärmetauscher (4) und dem Kompressor (1) verbunden ist;
wenigstens eine Steuereinheit (14) zum Steuern von wenigstens dem Absperrventil (8) und der ersten Druckreduzierungsvorrichtung (5) und der zweiten Druckreduzierungsvorrichtung (3), wobei der erste Wärmetauscher (2h) mit dem Hochtemperaturwassertank (9h) verbunden ist; wobei der zweite Wärmetauscher (2r) mit der Raumheizvorrichtung (Rhd) verbunden ist; und wobei der dritte Wärmetauscher (2p) mit dem Niedrigtemperaturwassertank (9p) verbunden ist.

2. System nach Anspruch 1, wobei eine thermische Solarwärmequelle mit der Wärmespeichervorrichtung (7) und dem Hochtemperaturwassertank (9h) und/oder dem Niedrigtemperaturwassertank (9p) verbunden ist.

3. System nach Anspruch 1 bis 2, wobei das Kältemittel CO2 ist.

4. Verfahren zum Steuern eines Dampfverdichtungskreislaufs in einem System nach Anspruch 1, das ein Umschalten zwischen einem ersten Heizbetriebsmodus und einem zweiten Heizbetriebsmodus ermöglicht, wobei das Verfahren im ersten Heizbetriebsmodus durch folgende Schritte gekennzeichnet ist:

Abgeben von Wärme von einem ersten Wärmetauscher (2h), einem zweiten Wärmetauscher (2r) und einem vierten Wärmetauscher (6), wodurch der erste, zweite und vierte Wärmetauscher als Gaskühler arbeiten, und wobei der vierte Wärmetauscher (6) Wärme an eine Wärmespeichervorrichtung (7) abgibt; und
Betreiben eines fünften Wärmetauschers (4) als einen Verdampfer.

5. Verfahren zum Steuern eines Dampfverdichtungskreislaufs in einem System nach Anspruch 4, das ein Umschalten zwischen einem ersten Heizbetriebsmodus und einem zweiten Heizbetriebsmodus ermöglicht, wobei das Verfahren im zweiten Heizbetriebsmodus durch folgende Schritte gekennzeichnet ist:

Abgeben von Wärme von einem zweiten Wärmetauscher (2r) und einem dritten Wärmetauscher (2p), wodurch der zweite und dritte Wärmetauscher als Gaskühler arbeiten; und
Betreiben eines vierten Wärmetauschers (6) als einen Verdampfer, der Wärme von einer Wärmespeichervorrichtung (7) aufnimmt.

6. Verfahren zum Steuern eines Dampfverdichtungskreislaufs in einem System nach Anspruch 4 oder 5, ferner folgenden Schritt beinhaltend:
Regeln der zwei Betriebsmodi gemäß einer Wärmequellentemperatur (T2) und der Tageszeit.

Revendications

1. Système comprenant un réservoir d'eau à haute température (9h) comprenant un dispositif de chauffage de salle (Rhd) et un réservoir d'eau à basse température (9p), et donnant lieu à un cycle de compression de vapeur comprenant deux modes séparés de chauffage en fonctionnement, comprenant un compresseur (1) ;
un premier échangeur de chaleur (2h) à refroidissement de gaz en aval du compresseur (1) ; un deuxième échangeur de chaleur à refroidissement de gaz (2r) en aval du premier échangeur de chaleur (2h) ;
un troisième échangeur de chaleur à refroidissement de gaz (2p) en aval du deuxième échangeur de chaleur (2r) ;
un premier dispositif de réduction de pression (5) situé en aval du troisième échangeur de chaleur (2p) ;
caractérisé par
un quatrième échangeur de chaleur (6) doté d'un dispositif de stockage de chaleur (7) situé en aval du premier dispositif de réduction de pression (5) ;
un second dispositif de réduction de pression (3) situé en aval du quatrième échangeur de chaleur (6) ;
un cinquième échangeur de chaleur (4) situé en aval du deuxième dispositif de réduction de pression (3) raccordé en retour au compresseur (1) ;
une conduite de dérivation dotée d'une soupape d'arrêt (8) contournant le cinquième échangeur de chaleur (4), raccordée à une première extrémité entre le quatrième échangeur de chaleur (6) et le second dispositif de réduction de pression (3) et à une seconde extrémité, entre le cinquième échangeur de chaleur (4) et le compresseur (1) ;
au moins une unité de commande (14) permettant de commander au moins la soupape d'arrêt (8) et le premier dispositif de réduction de pression (5) et le second dispositif de réduction de pression (3)
le premier échangeur de chaleur (2h) étant raccordé au réservoir d'eau à haute température (9h) ;
le deuxième échangeur de chaleur (2r) étant raccordé au dispositif de chauffage de salle (Rhd) ; et
le troisième échangeur de chaleur (2p) étant raccordé au réservoir d'eau à basse température (9p).

2. Système selon la revendication 1, où une source de chaleur thermique solaire est raccordée au dispositif de stockage de chaleur (7) et au réservoir d'eau à haute température (9h) et/ou au réservoir d'eau à basse température (9p).

3. Système selon la revendication 1 ou 2, dans lequel le réfrigérant est le CO2.

4. Procédé de régulation d'un cycle de compression de vapeur dans un système conforme à la revendication 1, qui permet la commutation entre un premier mode de fonctionnement de chauffage et un second mode de fonctionnement de chauffage, le procédé **se caractérisant par** les étapes suivantes dans le premier mode de fonctionnement :

> le rejet de chaleur à partir d'un premier échangeur de chaleur (2h), d'un deuxième échangeur de chaleur (2r) et d'un quatrième échangeur de chaleur (6), moyennant quoi les premier, deuxième et quatrième échangeurs de chaleur fonctionnent en tant que refroidisseurs à gaz, et dans lequel le quatrième échangeur de chaleur (6) rejette de la chaleur vers un dispositif de stockage de chaleur (7) ; et
> le fonctionnement d'un cinquième échangeur de chaleur (4) en tant qu'évaporateur.

5. Procédé de régulation d'un cycle de compression de vapeur dans un système conforme à la revendication 4, qui permet la commutation entre un premier mode de fonctionnement de chauffage et un second mode de fonctionnement de chauffage, le procédé **se caractérisant par** les étapes suivantes dans le second mode de fonctionnement :

> le rejet de chaleur à partir d'un deuxième échangeur de chaleur (2r) et d'un troisième échangeur de chaleur (2p), moyennant quoi les deuxième et troisième échangeurs de chaleur fonctionnent en tant que refroidisseurs à gaz ; et
> le fonctionnement d'un quatrième échangeur de chaleur (6) en tant qu'évaporateur qui absorbe de la chaleur à partir d'un dispositif de stockage de chaleur (7).

6. Procédé de régulation d'un cycle de compression de vapeur dans un système conforme à la revendication 4 ou 5, comprenant en outre l'étape suivante :
la commande des deux modes de fonctionnement selon une température de source de chaleur (T2) et l'heure du jour.

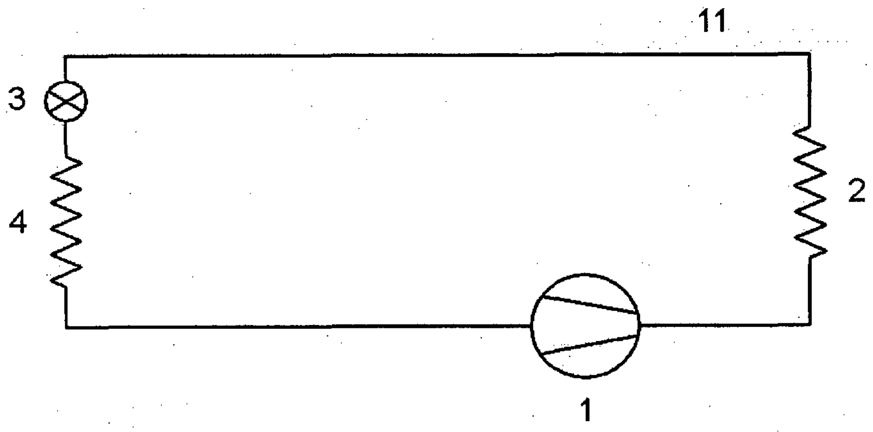

FIG. 1

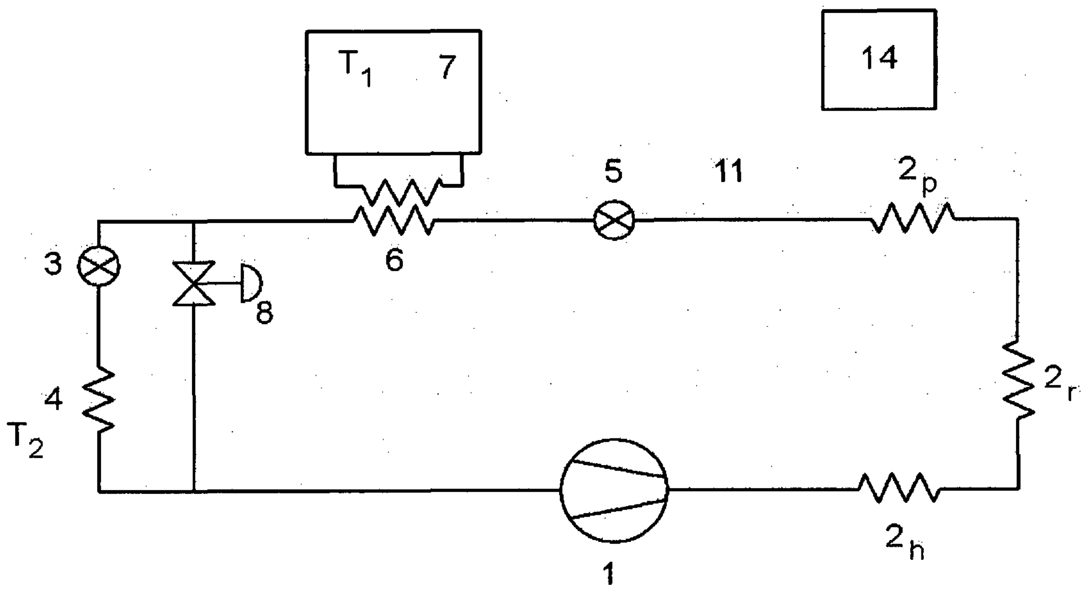

FIG. 2

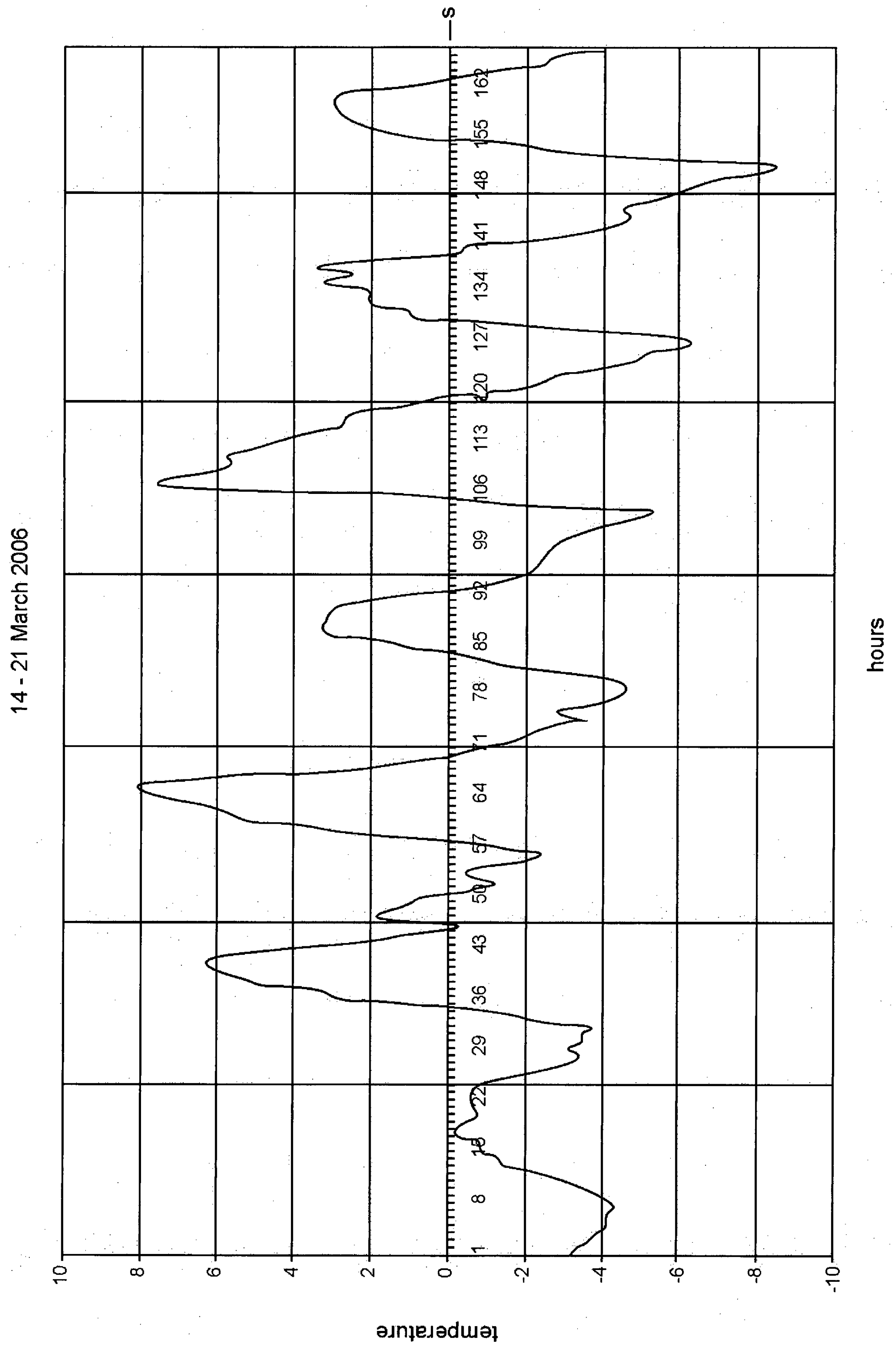
14 - 21 March 2006
temperature
10
8
6
4
2
0
-2
-4
-6
-8
-10
1
8
15
22
29
36
43
50
57
64
71
78
85
92
99
106
113
120
127
134
141
148
155
162
hours
s

FIG. 3

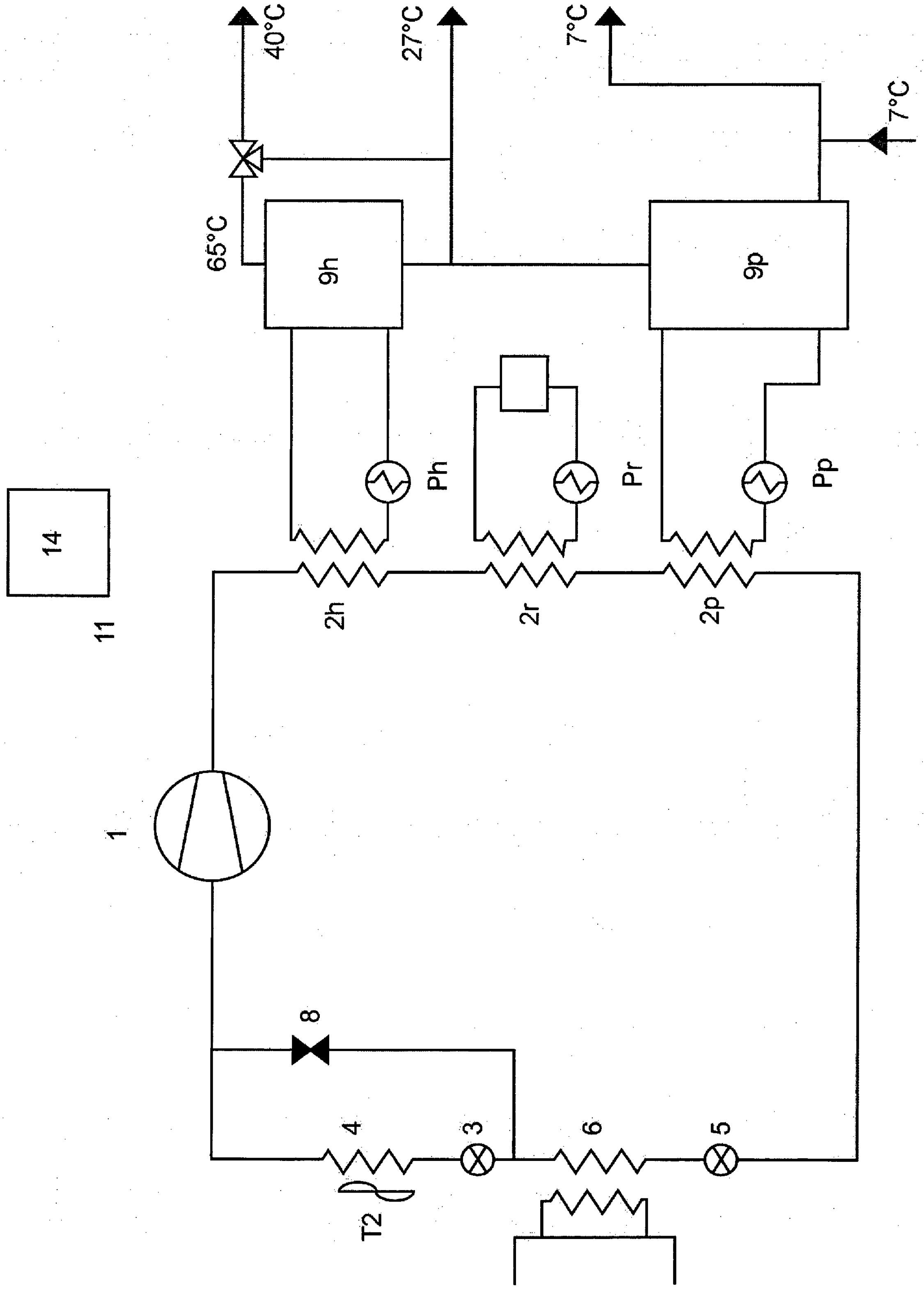

FIG. 4

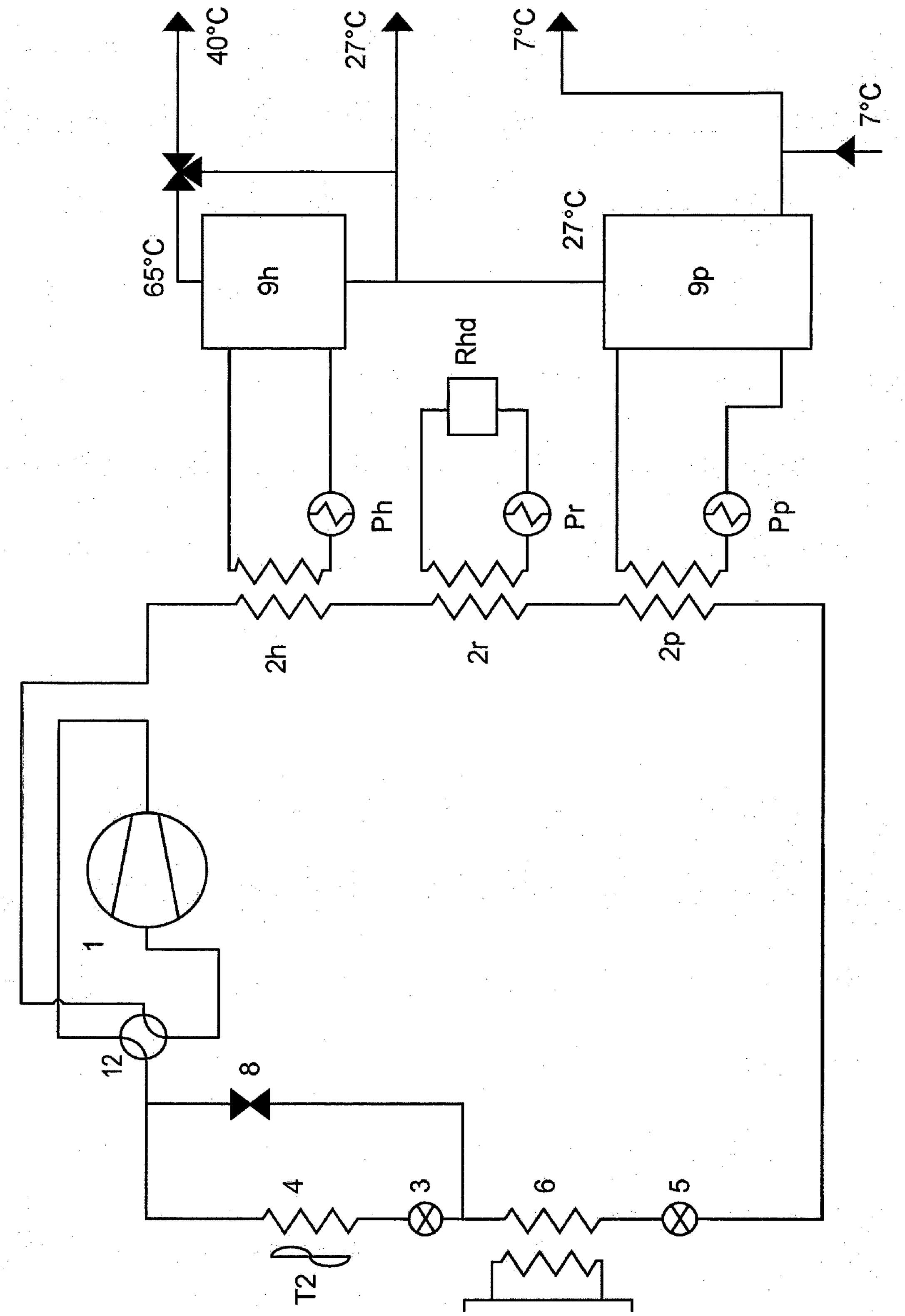

FIG. 4b

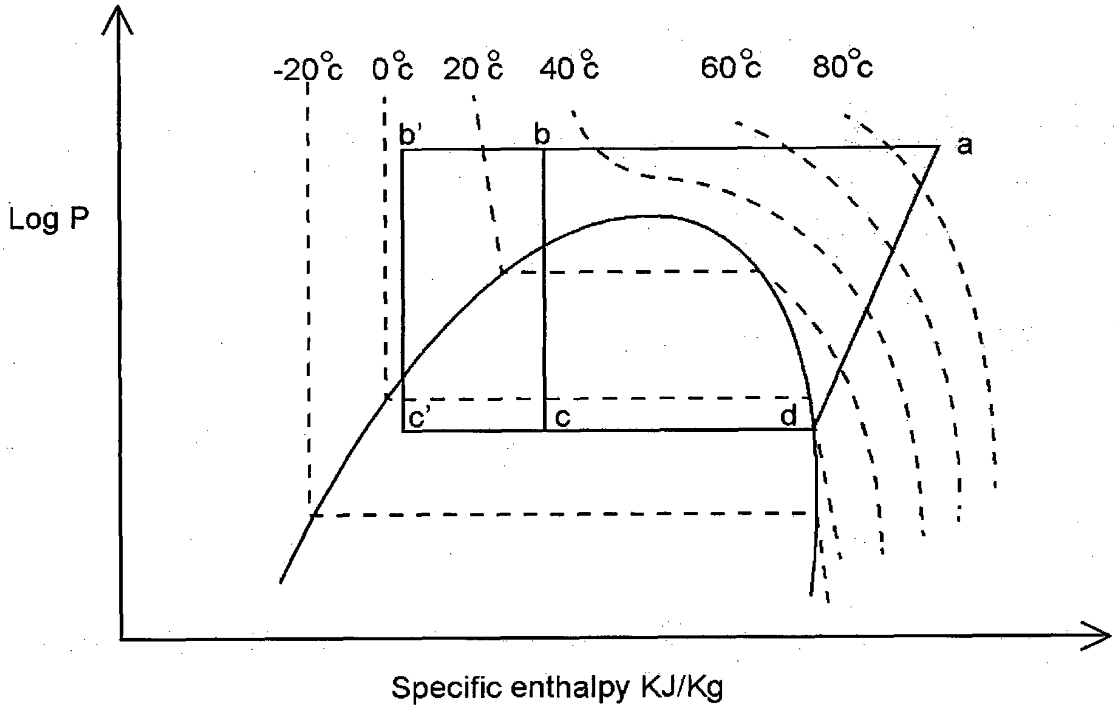
-20°c
0°c
20°c
40°c
60°c
80°c
b'
b
a
Log P
c'
c
d
Specific enthalpy KJ/Kg

FIG. 5

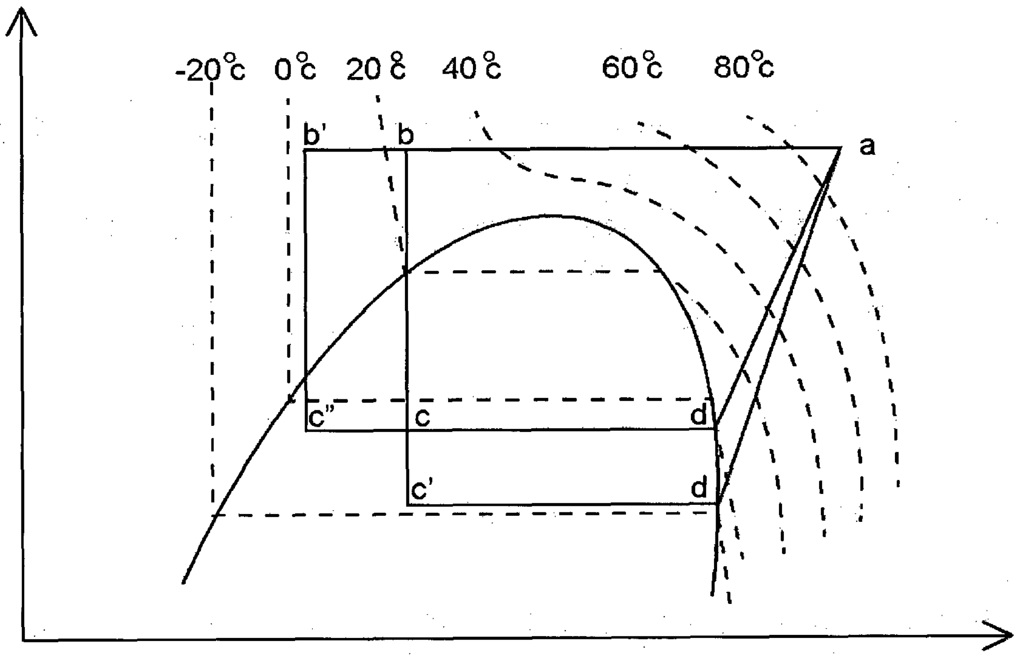
-20°c
0°c
20°c
40°c
60°c
80°c
b'
b
a
c"
c
d
c'
d

FIG. 6

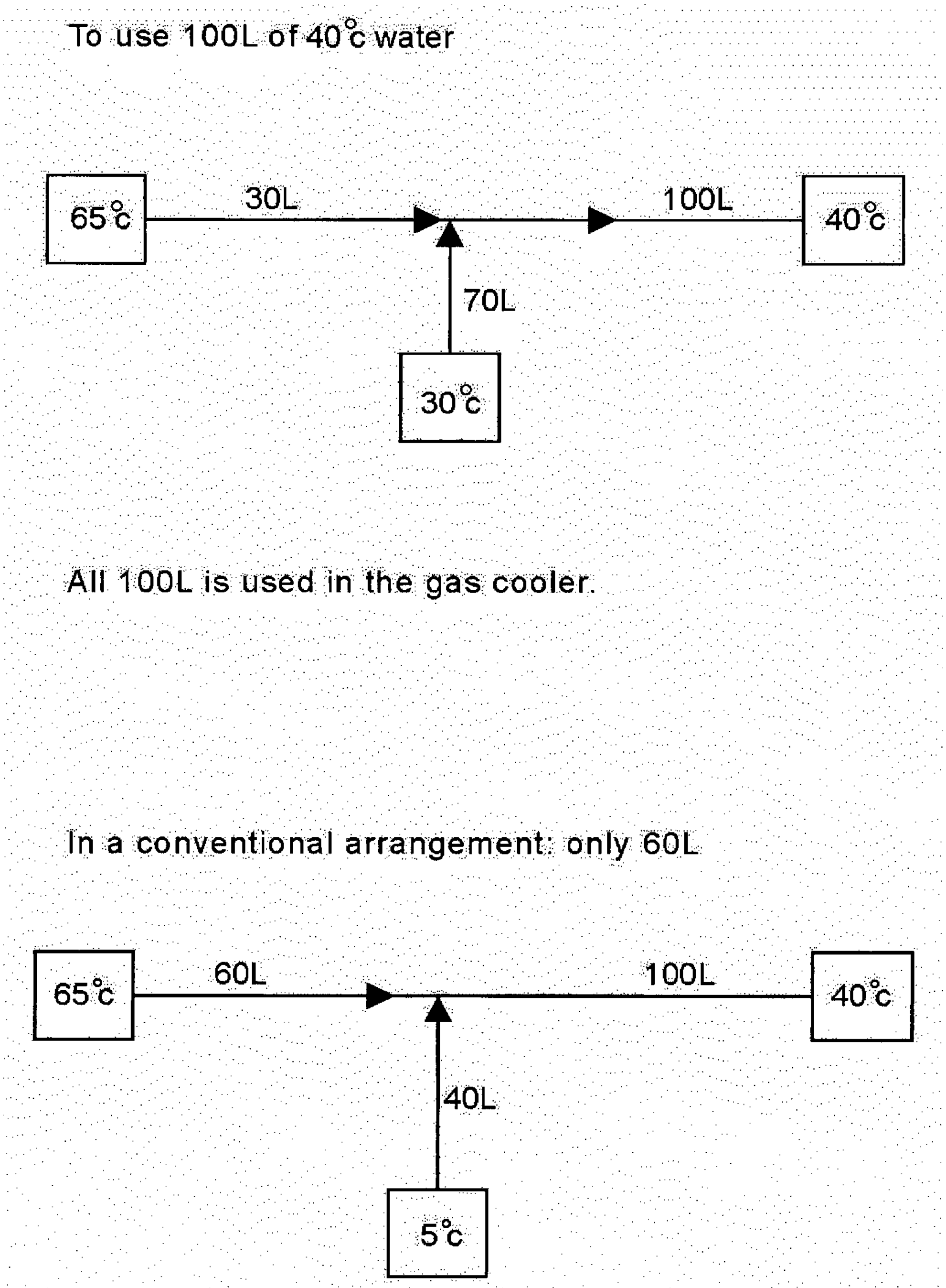
To use 100L of 40°c water
65°c
30L
100L
40°c
70L
30°c
All 100L is used in the gas cooler.
In a conventional arrangement: only 60L
65°c
60L
100L
40°c
40L
5°c

FIG. 7

REFERENCES CITED IN THE DESCRIPTION

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- WO 2005106346 A1 **[0003]**
- EP 0424474 B2 **[0003]**
- US 4012920 A **[0006]**
- US 3523575 A **[0007]**
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- **JØRN STENE.** Residential CO2 heatpump systems for combined space heating and hot water heating. *Doctoral Thesis,* ISBN 82-471-6316-0 **[0005]**