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(54) **HEATING DEVICE WITH IRREVERSIBLE THERMODYNAMIC CYCLE FOR HEATING INSTALLATIONS HAVING HIGH DELIVERY TEMPERATURE**

HEIZAGGREGAT MIT IRREVERSIBLEM THERMODYNAMISCHEM KREISLAUF ZUR BEHEIZUNG VON INSTALLATIONEN MIT HOHER AUSGANGSTEMPERATUR

DISPOSITIF DE CHAUFFAGE À CYCLE THERMODYNAMIQUE IRRÉVERSIBLE, POUR INSTALLATIONS DE CHAUFFAGE AYANT UNE HAUTE TEMPÉRATURE DE SORTIE

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(73) Proprietor: **TEON S.R.L.**
20900 Monza (MB) (IT)

(72) Inventor: **PELLEGRINI, Gianfranco**
I-10137 Torino (IT)

(74) Representative: **Cinquantini, Bruno et al**
Notarbartolo & Gervasi S.p.A.
Corso di Porta Vittoria, 9
20122 Milano (IT)

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Description

FIELD OF THE INVENTION

[0001] The present invention relates to the field of heating devices for heating rooms. In particular, the subject of the present invention is a heating device with irreversible thermodynamic cycle for heating installations having high delivery temperature (higher than 80 °C). The device according to the invention stands out for the high coefficients of performance achievable (higher than 3-4) and consequently for the high energy saving.

STATE OF THE ART

[0002] The use of devices known under the name of "heat pumps" for heating rooms is widely known. Such devices are based on the principle of removing thermal energy from a source having a lower temperature (also called cold source) to transfer it to a source having a higher temperature (also called hot source). Energy is transferred by means of the circulation of an operating fluid in a circuit comprising an evaporator, a compressor, connected with the outlet of the evaporator, a condenser in series to the compressor and expansion means connected between the condenser and the evaporator.

[0003] It is also known that heat pumps may be of the air-water, air-air or water-water type. In the case of air-air and air-water pumps the cold source is represented by the air outside the room to be climatized, while in the case of water-water pumps the cold source consists of a water flow, e.g. a ground water flow or deep-running water flow (geothermal flow). In the first case of the ground water, the temperature of the source is generally in a range between +7 and +12 °C during the whole year, while in the case of deep-running water the temperature may reach up to 14-15 °C. Keeping the operating conditions the same, the heat pumps of the water-water type have higher coefficients of performance (COP) compared to the air-air or air-water pumps.

[0004] In the case of a cold source consisting of ground water, by means of the traditional heat pumps a maximum water delivery temperature the water (i.e. of the hot source) of about 65 °C is obtained.

[0005] It has been observed that, in most cases, the current technical solutions allow a high delivery temperature to be obtained however to the detriment of the coefficient of performance. In other words, almost all of the technical solutions currently on the market do not allow high delivery temperatures (i.e. above 65 °C) and high coefficients of performance (i.e. above 3) to be obtained at the same time.

[0006] In order to at least partially overcome this drawback, it is possible to make "two-stage" heat pumps. Referring to the scheme of figure 1, such devices comprise a first and a second circuit, through which corresponding operating fluids flow. A first circuit, called low temperature (LOW) circuit, absorbs thermal energy from a water flow usually of geothermal origin (Hgeo) for evaporating a first operating fluid (R1) in an evaporator (E1). The first operating fluid is compressed by means of a compressor (C1) and is then condensed in a heat exchanger (T). It is then expanded by means of an expansion valve (V1) to be returned to the evaporator (E1).

[0007] By means of the heat exchanger (T), the thermal energy of condensation is used to evaporate a second operating fluid (R2) circulating in the second operating circuit, also called high temperature (HIGH) circuit. In other words, the heat exchanger (T) acts as a condenser for the operating fluid of the low temperature circuit and as an evaporator for the one relating to the high temperature circuit. Similarly to the low temperature circuit, also the high temperature circuit comprises a compressor (C2) and an expansion valve (V2) between which a condenser (T2) is provided for condensing the second operating fluid. The latent heat of condensation of the second fluid (R2) is transferred to a delivery water flow, i.e. a water flow intended to be used as heating water or alternatively as water for sanitary use.

[0008] Figure 2 illustrates, in an enthalpy-pressure (H-P) diagram, the thermodynamic cycles (ABCD - A'B'C'D') performed by the two operating fluids flowing through the two circuits illustrated in figure 1. The diagram in figure 1 shows the working temperatures relating to an operating condition achievable with this type of device. Considering e.g. the fluid R12 as the operating fluid for the low temperature circuit, then the related evaporation (Te1) and condensation (Tc1) temperatures may be assumed to be, e.g., 10 °C and 42 °C, respectively at condensation and evaporation pressures respectively of 4.2 bar and 10 bar; considering also for the second operating circuit the fluid R12, the temperatures of evaporation (Te2) and condensation (Tc2) are established at 40 °C and 87 °C for respective pressures corresponding to 9.6 bar and 27 bar. Of course, temperature and pressure values vary according to the type of operating fluid used and therefore the values indicated are an example of a possible and known operating mode.

[0009] Still from the diagram of figure 2 it is noted that the expansions (steps A-D and A'-D') of the two operating fluids occur once complete condensation is achieved, i.e. when the corresponding fluid has a vapour quality equal to 0. In particular, such expansions through the throttling valves V1 and V2 are configured as substantially isenthalpic transformations which lead to a simultaneous decrease of the temperature and pressure of the operating fluid. The compression steps (B-C and B'-C') substantially determine the electric power which is absorbed by the two-stage heat pump for carrying out the thermodynamic cycles. Such an electrical power directly influences the calculation of the coefficient of performance (COP).

[0010] Even though the solution just described above and more generally two-stage heat pumps so conceived allow higher delivery temperatures to be reached compared to single-stage heat pumps, they have much lower coefficients of performance. Even in the case of the use of water from geothermal sources with temperatures higher than 30 °C, it is observed that the coefficient of performance for these thermal machines rarely reaches values exceeding 3. In other words, for this type of devices it is difficult to achieve high performance and low energy consumption which can justify their use and increased technological content. DE-A-33 11 505 discloses a heating device according to the preamble of claim 1.

[0011] Therefore, based on these considerations, it is the primary task of the present invention to provide a heating device which allows overcoming the drawbacks which currently accompany single-stage and two-stage heat pumps used for heating water used for room heating and as sanitary water. It is one object of the present invention within the scope of this task to provide a heating device which allows a high coefficient of performance and at the same time a high delivery temperature (above 80 °C) of the water intended for a heating installation and/or having sanitary use also in the presence of a cold source consisting of ground water. It is another object of the present invention to provide a heating installation which is reliable and easy to manufacture at competitive costs. This object is achieved with a heating device having the features of claim 1.

SUMMARY

[0012] The present invention relates to a heating device with irreversible thermodynamic cycle comprising a first, low temperature, operating circuit for the circulation of a first operating fluid. Such a first circuit comprises evaporating means of the first operating fluid provided to exchange thermal energy with a supply water flow (ground water or water of geothermal origin) for the purpose of extracting therefrom the thermal energy required for evaporation. The first circuit further comprises compression means, condensation means and expansion means of the first operating fluid. The device according to the invention also comprises a second, high temperature, operating circuit for the circulation of a second operating fluid. Such a second circuit comprises evaporating means of the second fluid which perform the evaporation of the same by means of the thermal energy deriving from the condensation of the operating fluid circulating in the low temperature circuit. The second operating circuit further comprises compression means for compressing the second operating fluid after the evaporation thereof and condensation means for condensing the same second fluid after the compression thereof. Such condensation means of the second fluid exchange thermal energy with a delivery water flow to allow it to be heated. The second operating circuit is further provided with expansion means of the second fluid. The heating device according to the invention is characterized in that the first circuit comprises first cooling means operatively provided between the condensation means and the expansion means of the first circuit. Such first cooling means cool the first operating fluid while heating a first partial flow of the supply water flow. Moreover, according to the invention, the second circuit comprises second cooling means operatively provided between the condensation means and the expansion means of the second fluid so as to cool the second fluid after the condensation thereof and so as to heat a second partial flow of said supply water flow, which is independent from the first partial flow.

[0013] The use of the first and of the second cooling means allows the supply flow to be heated effectively before it releases its thermal energy to the evaporating means of the first fluid. Such cooling means, by heating in parallel the first and the second partial flows, allow the thermal energy of the water flow to be increased, i.e. higher values of the coefficient of performance (COP) to be achieved. The use of operating fluids optimal for heating purposes practically makes the high and low thermodynamic cycles irreversible. Nevertheless such an irreversibility of the cycle advantageously allows coefficients of performance (COP) greater than 3-4 to be obtained also with a set-point temperature greater than 85 °C, this expression being used to indicate the temperature of the delivery water flow.

LIST OF THE DRAWINGS

[0014] Further features and advantages of the present invention shall become more apparent from the description of preferred, but not exclusive, embodiments of the heating device with irreversible thermodynamic cycle according to the present invention, provided for illustrating and non-limiting purposes, with the aid of the attached drawings, in which:

- figure 1 is a diagram relating to a two-stage heat pump of traditional type;
- figure 2 relates to an enthalpy-pressure (H-P) diagram relating to the two-stage heat pump schematized in figure 1;
- figure 3 relates to an operating diagram of a heating device according to the present invention;
- figure 4 is an enthalpy-pressure diagram relating to the irreversible thermodynamic cycle of the heating device in figure 3.

DETAILED DESCRIPTION OF THE INVENTION

[0015] Figure 3 is a diagrammatic view of a heating device 1 according to the present invention. The device 1 comprises a first operating circuit 10 (hereinafter also indicated as "low temperature circuit 10") inside which a first operating fluid R1 is active. The first operating circuit 10 comprises evaporating means 11 of the first operating fluid R1 (hereinafter also indicated with the expression "first evaporating means 11"). Such evaporating means 11 are configured to remove thermal energy from a supply water flow F (hereinafter also indicated as "supply flow F"), which may be a ground water flow or also a water flow of geothermal origin. In terms of thermodynamic cycle, the supply water flow F represents the cold source of thermal exchange. The first circuit further comprises compression means 21 of the first fluid R1 (hereinafter also indicated with the expression "first compression means 11"), which compress the same after evaporation.

[0016] The first circuit 10 comprises condensation means 31 of the first operating fluid R1 (hereinafter also indicated with the expression "first condensation means 31") for condensing the same after its compression. Expansion means 41 of the first fluid R1 (hereinafter also indicated with the expression "first expansion means 41") are provided to bring the pressure from the value at which the condensation of the first fluid R1 is achieved back to the value at which the related evaporation occurs.

[0017] The heating device 1 further comprises a second operating circuit 100 (hereinafter also indicated as "high temperature circuit 100") inside which a second operating fluid R2 is active. The second operating circuit 100 comprises second evaporating means 111 of the second fluid R2 (also indicated as second evaporating means 111), which evaporate the latter by exploiting the thermal energy deriving from the condensation of the first fluid R1 circulating in the low temperature circuit 10. As better specified below, the second evaporating means 111 and the first condensation means 31 are preferably integrated in a single heat exchanger indicated by the reference 50 in figure 3. In practice, by means of this exchanger 50 the condensation energy released due to the condensation of the first fluid R1 is directly transferred to the fluid R2 of the high temperature circuit 100 to allow the evaporation without intermediate passages.

[0018] The second operating circuit 100 further comprises compression means 121 of the second operating fluid R2 (also indicated as "second compression means 121") to increase the pressure and superheat the same fluid after the related evaporation obtained by means of the second evaporating means 111. Condensation means 131 of the second fluid R2 (also indicated as "second condensation means 131") are provided to transfer the thermal energy of the condensation (hereinafter also indicated as "latent heat of condensation") to a "delivery" water flow Hman, this expression being used to indicate, e.g., a water flow intended for a utility for domestic or sanitary use. Such a delivery water flow Hman represents the hot source for the thermodynamic cycle relating to the device 1. Second expansion means 141 of the second operating fluid R2 are provided to bring the second operating fluid R2 from the related condensation pressure Pc2 back to the evaporation pressure Pe2. According to the invention, the first circuit 10 comprises cooling means 72 of the first operating fluid R1 (hereinafter also indicated as "first cooling means 72") operatively provided between the first condensation means 31 and the first expansion means 41. Such first cooling means 72 of the first operating fluid R1 serve the function of cooling the first operating fluid R1 which leaves the condensation means 31, at the same time heating a first partial flow F1 of the supply water flow F intended to release (after the heating of the first partial flow F1) its thermal energy to the first operating fluid R1 by means of the evaporating means 11 of the same low temperature circuit 10.

[0019] Again according to the invention, the second circuit 100 comprises cooling means 71 of the second operating fluid R2 (hereinafter also indicated as "second cooling means 71") operatively provided between the second condensation means 131 and the second expansion means 141. Such second cooling means 71 serve the function of undercooling the second operating fluid R2 which leaves the condensation means 131 of the second circuit 100, at the same time heating a second partial flow F2 of said supply water flow F intended to release (after the heating of the second partial flow F2) its thermal energy to the first operating fluid R1 by means of the evaporating means 11 of the same low temperature circuit 10.

[0020] In other words, the first cooling means 72 and the second cooling means 71 serve the function of removing the thermal energy from the respective operating fluids R1, R2 to increase at the end the thermal level of the supply flow F, i.e. the thermal level of the cold source of the heating device 1. In particular, the first partial flow F1 and the second partial flow F2 are hydraulically "independent", i.e. they are heated independently by means of the first cooling means 72 and second cooling means 71, respectively. Substantially, the supply water flow F coming from a source, e.g. of geothermal type, is split at least into the first partial flow F1 and the second partial flow F2, which are heated essentially in "parallel" to be then brought again together, so as to restore the amount of the supply water flow F intended for the evaporation means 11 of the first fluid R1 of the first circuit 10.

[0021] It has been observed that this solution allows a high level of heating of the supply water flow F to be obtained, since the thermal level increase obtained for the first partial flow F1 combines with the one obtained for the second partial flow F2. The first cooling means 72 and the second cooling means 71 are substantially configured as heat exchanges capable of removing part of the thermal energy of the first fluid R1 and of the second fluid R2, respectively, to raise the thermal level of the related partial flows F1, F2, i.e. of the water flow F comprising them. As the coefficient of performance (COP) depends on the difference between the temperature of the hot source and that of the cold source, this solution allows COPs higher than those of traditional two-stage heat pumps considered above to be obtained, keeping other operating conditions the same. The heating substantially "in parallel" of the two partial flows F1, F2 allows the size of the cooling means 71, 72 to be contained, as only a part (the first partial flow F1 or the second partial flow F2) of the supply flow F flow through them. This allows (again keeping other operating conditions the same) a significant increase of temperature of the supply flow F to be obtained. The reduced size of the cooling means 71, 72 advantageously results in a containment of the overall volume of the heating device 1.

[0022] It is noted that the heating in parallel of the first partial flow F1 and of the second partial flow F2 by means of the respective cooling means 72, 71 advantageously allows the operation of the heating device 1 to be optimized also in the case in which the maximum power thereof is not required. In terms of design, the first cooling means 72 are in fact advantageously independent from the second cooling means 71. In particular, the latter are sized so that the second partial flow F2 may be greater than the first partial flow F1 flowing through the first cooling means 72. When the maximum operating power of the heating device 1 is not required, the condensation means 131 of the second circuit 100 determine a partial condensation which is completed downstream of the second cooling means 71. Thereby part of the latent heat of condensation is directly transferred to the second partial flow F2. This significantly increases the final temperature of the supply flow F intended for the evaporation means 11 of the first circuit 10. The increased amount of thermal energy available for evaporating the first operating fluid R1 advantageously decreases the thermal power required for the compression of the same first fluid. This aspect consequently results in an advantageous increase of the coefficient of performance COP.

[0023] With reference to the diagram in figure 3, the heating device 1 according to the invention preferably comprises a delivery manifold 81 and a return manifold 82 of the supply water flow F. In particular, the delivery manifold 81 comprises a first inlet for the supply flow F from a source of ground water or water of geothermal origin. The delivery manifold 81 further comprises a first outlet for the first partial flow F1 intended for the first cooling means 72 of the first operating fluid R1 circulating in the first circuit 10. The delivery manifold 82 also comprises at least a second outlet for the second partial flow F2 intended instead for the second cooling means 71 of the second operating fluid R2 circulating in the second circuit 100.

[0024] The return manifold 82 comprises at least a first inlet for the first partial flow F1 heated by means of the heat exchange with the first cooling means 72 of the first operating fluid R1. The return manifold 82 also comprises a second inlet for the second partial flow F2 heated by means of the heat exchange with the second cooling means 71 of the second operating fluid R2. The return manifold 82 further comprises a main outlet for the supply water flow F intended for the evaporating means 11 of the first circuit 10.

[0025] According to the invention, the supply water flow F thus reaches the delivery manifold 81 by means of the first inlet and is then split at least into the first flow F1 and into the second flow F2. Said first partial flow F1 reaches, by means of the first inlet, the first cooling means 72, at which the same first partial flow F1 is heated with a consequent cooling of the first operating fluid R1. At the same time, thus in parallel, the second partial flow F2 reaches, by means of the second outlet, the second cooling means 71, at which it is heated with a consequent cooling of the second operating fluid R2. After having been heated, the first partial flow F1 and the second partial flow F2 reach the return manifold 82 to restore the supply flow F, which reaches, by means of the main outlet of the same return manifold 82, the evaporation means 11 of the first circuit 10 to release its thermal energy to the first operating fluid R1.

[0026] Figure 4 depicts the thermodynamic cycle relating to the two operating fluids R1 and R2, in an enthalpy-pressure diagram of the heating device 1 according to the diagram in figure 3. Referring to the first operating circuit 10, the first evaporating means 11 perform the evaporation of the first fluid R1 by removing thermal energy from the supply water flow F. Such an evaporation is indicated in the diagram by the isobaric transformation $6 \rightarrow 7$ and occurs at a pressure Pe1 and at a constant temperature Te1. Following the complete evaporation of the first fluid R1, the latter is compressed by means of the first compression means 21 (transformation $1 \rightarrow 2$). Such a compression determines a temperature increase up to the value T3 corresponding to point 2 of the diagram and a related increase of pressure (up to the value Pc1) and enthalpy.

[0027] The first fluid R1 leaving the compression means 21 reaches the first condensation means 31 to be condensed at the corresponding constant pressure of condensation Pc1 and at a constant temperature of condensation Tc1. As still apparent from figure 4, the first fluid R1 first undergoes a de-overheating (transformation $2 \rightarrow 3$) at a constant pressure inside the first condensation means 31, before the condensation. The second evaporating means 111 of the high temperature circuit 100 perform the evaporation of the second operating fluid R2 by exploiting the latent heat of condensation of the first fluid R1. As already indicated above, the first condensation means 31 and the second evaporating means 111 are preferably integrated in a single heat exchanger 50 which directly transfers the thermal energy released from the first fluid R1 to the second fluid R2.

[0028] When the evaporation of the second operating fluid R2 is completed, the second compression means 121 increase at the same time the pressure and the temperature of the same fluid ($1 \rightarrow 2'$). The overheated second fluid R2 then flows through the second condensation means 131, in which the vapour is first de-overheated (transformation $2 \rightarrow 3'$) and then condensed until the complete liquid state is achieved ($3' \rightarrow 4'$). The latent heat of condensation is thermally transferred by means of the second condensation means 131 to the delivery water flow Hman, which undergoes a corresponding heating which may advantageously exceed 80 °C.

[0029] When the condensation is completed, the second fluid R2 in the liquid state flows through the second cooling means 71 to be further cooled at a constant pressure Pc2 (step $4' \rightarrow 5'$). As already indicated above, the second cooling means 71 recover part of the thermal energy remaining in the condensed second fluid R2 while heating the second partial flow F1 of the supply water flow F, i.e. the cold source. Following this post-condensation cooling, the second operating fluid R2 in the liquid state is expanded isenthalpically as well as isothermally up to the pressure of evaporation Pe2, by means of the second expansion means 141. In practice the passage of the fluid R2 inside the second expansion means 141 determines just a pressure reduction, but not a temperature variation, as the fluid R2 is always kept at the liquid state during this transformation. In other words, unlike traditional heat pumps, the expansion step is not accompanied by a loss of thermal energy. On the contrary, during the undercooling of the liquid R2, an enthalpy amount (indicated by rectangle A in figure 4) is recovered which, by means of the second partial flow F2, is directly transferred to the supply fluid F. This means that the amount of thermal energy which may be removed from the supply fluid F itself to evaporate the first operating fluid R1 of the first circuit 10, is increased. As already emphasized above, in accordance with the objects of the present invention, the heating of the supply fluid F results in an increase of the temperature of the cold source, i.e. in an advantageous increase of the COP, keeping the energy consumption for the compression of the two operating fluids R1, R2 the same.

[0030] As already indicated above, in the case in which the heating device 1 is not required to operate at maximum power conditions, then the condensation means 131 of the second circuit, 100 determine a partial condensation, which is completed by the second cooling means 71. The thermal energy deriving from the completion of such a condensation is recovered and transferred to the first partial flow F1 and finally to the supply water flow F. The heating device 1 according to the invention is in practice capable of operating under all operating conditions with high values of COP.

[0031] As indicated above, the first cooling means 72 are provided between the condensation means 31 and the expansion means 41 of the first operating fluid R1 to undercool the first operating fluid R1 condensed by means of the same condensation means 31. The first cooling means 72 allow the temperature of the first partial flow F1, i.e. ultimately of the supplying flow F, to be increased. In the case in which the first operating fluid R1 is completely condensed in the condensation means 31, the first cooling means 72 further undercool the same first fluid R1 by removing a further part of the thermal energy (indicated by the letter B in figure 4). In the case in which the condensation means 31 of the first circuit 10 determine instead a partial (not complete) condensation, by means of the first cooling means a part of the thermal energy (indicated by the letter C in the diagram in figure 4) deriving from the completion of the condensation of the first fluid R1 is recovered and is transferred to the first partial flow F1 and finally to the supply water flow F.

[0032] In practice, this solution allows an amount of thermal energy of condensation to be positively exploited which would otherwise be lost, as it occurs in traditional two-stage pumps. From the diagram in figure 2 it can be noted in fact that, due to the "bell-like" shape of the liquid-vapour diagram, the amount of energy deriving from the condensation of the low temperature fluid (first fluid R1) is greater than the one which may actually be absorbed by the high temperature fluid (second fluid R2). Therefore, in traditional solutions a part of the thermal energy of condensation is substantially lost, as it may not be absorbed by the second high temperature fluid R2. Differently, in the device 1 in figure 3, the energy which may not be received by the second fluid R2 is advantageously used to heat the cold source (supply water flow F) to the benefit of an increase of the COP.

[0033] Once the condensation of the first fluid R1 is completed, the first cooling means 72 further cool the same first fluid R1 so as to recover a further amount of thermal energy (indicated by the letter B in the diagram in figure 4). Also this further thermal energy is advantageously transferred to the first delivery F1, i.e. to the cold source (water flow F) so as to increase the energy content thereof.

[0034] The first compression means 21 and/or the second compression means 121 may consist of volumetric compressors of the type normally used for the manufacture of traditional heat pumps, or of other operatively equivalent means. In this respect, according to a preferred embodiment, the first compression means 21 of the first fluid R1 and/or the second compression means 121 of the second fluid R2 are configured so as to exchange thermal energy with a third partial flow F3 and with a fourth partial flow F4 of said supply water flow F (independent from said first partial flow F1 and from said second partial flow F2), respectively. This solution advantageously allows an overheating of the lubricants used for the operation of the mechanical components of the volumetric compressors to be limited, in order to improve the reliability and the life thereof. In fact, assuming to set a set point temperature of the device at about 85 °C, the temperature of condensation shall be at about 87-88 °C which means having a higher superheating after the compression (even 160 °C may be reached, depending on the coolant used). However, in the operating conditions the internal friction of the mechanical components which make up the volumetric compressors may reach even 150 °C. Such a temperature is obviously dangerous for the integrity of the lubricants used and therefore for the reliability of the appliance. By means of the solution in figure 3, the temperature of an overheating may remain confined into an acceptable range.

[0035] Again with reference to the solution in figure 3, it is noted that the delivery manifold 81 comprises a third outlet by means of which the third partial flow F3 of the supply water flow F reaches the compression means 21 of the first circuit 10 to remove thermal energy therefrom. The return manifold 82 comprises a third inlet through which the third partial flow F3 flows after the heating thereof obtained with the consequent cooling of the compression means 21 of the first circuit 10.

[0036] The delivery manifold 81 also comprises a fourth outlet for the fourth partial flow F4 of the supply water flow F. Such a fourth partial flow F4 reaches the compression means 121 of the second circuit 100 to remove thermal energy therefrom. The return manifold 82 comprises a fourth inlet through which the fourth partial flow F4 flows after the heating thereof obtained with the consequent cooling of the compression means 121 of the second circuit 100.

[0037] It is noted that the third partial flow F3 and the fourth partial flow F4 are independent from the first partial flow F1 and from the partial flow F2. In particular, the third partial flow F3 and the fourth partial flow F4 are heated substantially "in parallel" with each other and in parallel with the first partial flow F1 and with the second partial flow F2. According to this solution, the supply water flow F in the delivery manifold 81 is split into the four partial flows indicated by F1, F2, F3 and F4, which, after having been heated independently from one other, are conveyed into the return manifold 82 to restore again the amount of the supply flow F. The thermal level of such a supply flow F will be determined by the combination of the thermal levels of the four partial flows F1, F2, F3 and F4. It has been observed that the thermal energy removed from the compression means 21, 121 significantly increases the thermal level of the supply flow F thus contributing to obtaining further increased COP values (greater than 3). According to a preferred embodiment illustrated in figure 3, the delivery manifold 81 also comprises a fifth outlet hydraulically connected with a fifth input of the return manifold 82 by means of a compensating line L. The latter serves the function of compensating the various head losses of the four partial flows F1, F2, F3 and F4 which arise when the same partial flows flow from the delivery manifold 81 to the return manifold 82.

[0038] In a first possible operating mode, the two operating fluids R1, R2 may be of the same type and in particular, may have the same density. However, the technical solutions implemented above advantageously allow operating fluids with densities different from each other to be used and, in particular, allow a fluid R2 to be used in the high temperature circuit 100 having a lower density compared to the fluid circulating in the low temperature circuit 10. In fact, the higher enthalpy made available, compared to the traditional solutions, for the second fluid R2 allows the required mass flow rate thereof to be reduced. This allows, e.g., the consumption connected with the compression of the second fluid R2 to be reduced. In this regard, it has been observed that the heating device 1 reaches optimal operating performances when the fluid R 600 is used as the first operating fluid R1 in the low temperature circuit 10 and the fluid (Z)-2-Butene is used as the second operating fluid R2 in the high temperature circuit 100. Moreover, according to a further possible operating mode, a water solution (or even water alone) may be used as the operating fluid R2 circulating in the high temperature circuit 100, having taken into account the pressure and temperature conditions realized during the operation of the same high temperature circuit 100.

[0039] The solutions implemented for the heating device according to the invention allow the task and objects set above to be fully accomplished. In particular, the device according to the invention allows high coefficients of performance (COPs) to be obtained with clear advantages in terms of energy consumption. In other words, the device according to the invention allows a high setpoint temperature to be obtained with a low energy consumption of the system.

[0040] The device thus conceived may be subjected to many modifications and variations, all falling within the scope of the claims.

Claims

1. Heating device (1) with irreversible thermodynamic cycle, comprising:

    - a first circuit (10) for the circulation of a first operating fluid (R1), said first circuit (10) comprising:
    - evaporating means (11) of said first operating fluid (R1), said evaporating means (11) removing thermal energy from a supply water flow (F) for evaporating said first operating fluid (R1);
    - compression means (21) of said first operating fluid (R1) which compress said first fluid (R1) after the evaporation thereof;
    - condensation means (31) of said first operating fluid (R1) which condense said first fluid (R1) after the compression thereof;
    - expansion means (41) of said operating fluid (R1);

    \- a second circuit (100) for the circulation of a second operating fluid (R2), said second circuit (100) comprising:

        - evaporating means (111) of said second fluid (R2) which evaporate said second fluid (R2) by means of the thermal energy deriving from the condensation of said first fluid (R1) of said first circuit (10);
        - compression means (121) of said second fluid (R2) which compress said second operating fluid (R2) after the evaporation thereof;
        - condensation means (131) of said second fluid (R2) which condense said second operating fluid (R2) after the compression thereof, said condensation means (131) of said second fluid (R2) heating a delivery water flow (Hman) by means of the thermal

energy deriving from said condensation of said second fluid (R2);

- expansion means (141) of said second operating fluid (R2);

wherein:

- said first circuit (10) comprises first cooling means (72) operatively provided between said condensation means (31) and said expansion means (41) of said first circuit (10), said first cooling means (72) cooling said first operating fluid (R1)
- said second circuit (100) comprises second cooling means (71) operatively provided between said condensation means (131) and said expansion means (141) of said second fluid (R2) so as to cool said second fluid (R2) after the condensation thereof and

characterised in that

- said first cooling means (72) heats a first partial flow (F1) of said supply water flow (F);
- said second cooling means (71) heats a second partial flow (F2) of said supply water flow (F) independent from said first partial flow (F1);
- said device (1) further comprises a delivery manifold (81) and a return manifold (82) of said supply water:
- said delivery manifold (81) comprising at least a first inlet for said supply water flow (F), a first outlet for said first partial flow (F1) of said supply water flow (F) and at least a second outlet for said second partial flow (F2) of said supply water flow (F);
- said return manifold (82) comprising at least an inlet for said first partial flow (F1) and at least a second inlet for said second partial flow (F2) coming from said second cooling means (71), said return manifold (82) comprising a main outlet for said supply water flow (F) intended for said evaporating means (11) of said first circuit (10).

2. Device (1) according to claim 1, wherein said delivery manifold (81) comprises a third outlet for a third partial flow (F3) of said supply flow independent from said first partial flow (F1) and from said second partial flow (F2), said first compression means (21) of said first circuit (10) being configured so as to exchange thermal energy with said third partial flow (F3) for heating the same, said return manifold (82) comprising a third inlet for said third partial flow (F3) heated by means of the thermal exchange with said compression means (21) of said first circuit (10).

3. Device (1) according to claim 1 or 2, wherein said delivery manifold comprises a fourth outlet for a fourth partial flow (F4) of said supply flow (F) independent from said first partial flow (F1) and from said second partial flow (F2), said second compression means (121) of said second circuit (100) being configured so as to exchange thermal energy with said fourth partial flow (F4) for heating the same, said return manifold (82) comprising a fourth inlet for said fourth partial flow (F4) heated by means of the thermal exchange with said compression means (121) of said second circuit (100).

4. Device according to any one of the claims 1 to 3, wherein said delivery manifold (81) and said return manifold (82) are hydraulically connected by means of a compensating hydraulic line (L).

5. Device (1) according to any one of the claims 1 to 4, wherein said second operating fluid (R2) has a density lower than that of said first operating fluid (R1).

6. Device (1) according to any one of the claims 1 to 5, wherein said second operating fluid (R2) circulating in said second operating circuit (100) is a water solution or even water alone.

7. Device (1) according to any one of the claims 1 to 4, wherein said first operating fluid is R600 and wherein said second operating fluid is (Z)-2-Butene.

8. Device (1) according to one or more of the claims 1 to 7, wherein said evaporating means (111) of said second fluid (R2) and said condensation means (31) of said first fluid (R1) are integrated in a same heat exchanger (50) so that the thermal energy deriving from the condensation of the first fluid (R1) is directly transferred to the second fluid (R2) without intermediate passages.

9. A heating installation, **characterized by** comprising a heating device (1) according to one or more of the claims from 1 to 8.

Patentansprüche

1. Heizvorrichtung (1) mit irreversiblem thermodynamischem Kreisprozess, wobei die Heizvorrichtung umfasst:

- einen ersten Kreislauf (10) für den Umlauf eines ersten Betriebsfluids (R1), wobei der genannte erste Kreislauf (10) umfasst:

  - Verdampfungsmittel (11) des genannten ersten Betriebsfluids (R1), wobei die genannten Verdampfungsmittel (11) aus einer Speisewasserströmung (F) Wärmeenergie entnehmen, um das genannte erste Betriebsfluid (R1) zu verdampfen;

- Verdichtungsmittel (21) des genannten ersten Betriebsfluids (R1), die das genannte erste Fluid (R1) nach seiner Verdampfung verdichten;
- Kondensationsmittel (31) des genannten ersten Betriebsfluids (R1), die das genannte erste Fluid (R1) nach seiner Verdichtung kondensieren;
- Ausdehnungsmittel (41) des genannten Betriebsfluids (R1);

- einen zweiten Kreislauf (100) für die Umwälzung eines zweiten Betriebsfluids (R2), wobei der genannte zweite Kreislauf (100) umfasst:

- Verdampfungsmittel (111) des genannten zweiten Fluids (R2), die das genannte zweite Fluid (R2) mittels der Wärmeenergie, die von der Kondensation des genannten ersten Fluids (R1) des genannten ersten Kreislaufs (10) abgeleitet wird, verdampfen;
- Verdichtungsmittel (121) des genannten zweiten Fluids (R2), die das genannte zweite Betriebsfluid (R2) nach seiner Verdampfung verdichten;
- Kondensationsmittel (131) des genannten zweiten Fluids (R2), die das genannte zweite Betriebsfluid (R2) nach dessen Verdichtung kondensieren, wobei die genannten Kondensationsmittel (131) des genannten zweiten Fluids (R2) eine Lieferwasserströmung (Hman) mittels der von der genannten Kondensation des genannten zweiten Fluids (R2) abgeleiteten Wärmeenergie erwärmen;
- Ausdehnungsmittel (141) des genannten zweiten Betriebsfluids (R2); wobei:

- der genannte erste Kreislauf (10) erste Kühlmittel (72), die zwischen den genannten Kondensationsmitteln (31) und den genannten Ausdehnungsmitteln (41) des genannten ersten Kreislaufs (10) funktional vorgesehen sind, umfasst, wobei die genannten ersten Kühlmittel (72) das genannte erste Betriebsfluid (R1) abkühlen;
- der genannte zweite Kreislauf (100) zweite Kühlmittel (71) umfasst, die zwischen den genannten Kondensationsmitteln (131) und den genannten Ausdehnungsmitteln (141) des genannten zweiten Fluids (R2) in der Weise funktional vorgesehen sind, dass sie das genannte zweite Fluid (R2) nach dessen Kondensation abkühlen, und

dadurch gekennzeichnet, dass die genannten ersten Kühlmittel (72) eine erste Teilströmung (F1) der genannten Speisewasserströmung (F) erwärmen und dass die genannten zweiten Kühlmittel (71) eine zweite Teilströmung (F2) der genannten Speisewasserströmung (F) unabhängig von der genannten ersten Teilströmung (F1) erwärmen;
wobei die genannte Vorrichtung (1) ferner einen Lieferverteiler (81) und einen Rücklaufverteiler (82) des genannten Speisewassers umfasst:

- der genannte Lieferverteiler (81) wenigstens einen ersten Einlass für die genannte Speisewasserströmung (F), einen ersten Auslass für die genannte erste Teilströmung (F1) der genannten Speisewasserströmung (F) und wenigstens einen zweiten Auslass für die genannte zweite Teilströmung (F2) der genannten Speisewasserströmung (F) umfasst;
- der genannte Rücklaufverteiler (82) wenigstens einen Einlass für die genannte erste Teilströmung (F1) und wenigstens einen zweiten Einlass für die genannte zweite Teilströmung (F2), die von dem genannten zweiten Kühlmittel (71) kommt, umfasst, wobei der genannte Rücklaufverteiler (82) einen Hauptauslass für die genannte Speisewasserströmung (F), die für die genannten Verdampfungsmittel (11) des genannten ersten Kreislaufs (10) vorgesehen ist, umfasst.

2. Vorrichtung (1) gemäß Anspruch 1, wobei das genannte Lieferverfahren (81) einen dritten Auslass für eine dritte Teilströmung (F3) der genannten Speiseströmung unabhängig von der genannten ersten Teilströmung (F1) und von der genannten zweiten Teilströmung (F2) umfasst, wobei die genannten ersten Verdichtungsmittel (21) des genannten ersten Kreislaufs (10) so konfiguriert sind, dass sie Wärmeenergie mit der genannten dritten Teilströmung (F3) austauschen, um diese zu erwärmen, wobei der genannte Rücklaufverteiler (82) einen dritten Einlass für die genannte dritte Teilströmung (F3), die mittels des Wärmeaustauschs mit den genannten Verdichtungsmitteln (21) des genannten ersten Kreislaufs (10) erwärmt wird, umfasst.

3. Vorrichtung (1) gemäß Anspruch 1 oder 2, wobei der genannte Lieferverteiler einen vierten Auslass für eine vierte Teilströmung (F4) der genannten Speiseströmung (F) unabhängig von der genannten ersten Teilströmung (F1) und von der genannten zweiten Teilströmung (F2) umfasst, wobei die genannten zweiten Verdichtungsmittel (121) des genannten zweiten Kreislaufs (100) so konfiguriert sind, dass sie mit der genannten vierten Teilströmung (F4) Wärmeenergie austauschen, um diese zu erwärmen, wobei der genannte Rücklaufverteiler (82) einen vierten Einlass für die genannte vierte Teilströmung (F4), die mittels Wärmeaustausch mit den genannten Verdichtungsmitteln (121) des genannten zweiten Kreislaufs (100) erwärmt wird, umfasst.

4. Vorrichtung gemäß einem der Ansprüche 1 bis 3, wobei der genannte Lieferverteiler (81) und der genannte Rücklaufverteiler (82) mittels einer Kompensationshydraulikleitung (L) hydraulisch verbunden sind.

5. Vorrichtung (1) gemäß einem der Ansprüche 1 bis 4, wobei das genannte zweite Betriebsfluid (R2) eine niedrigere Dichte als das genannte erste Betriebsfluid (R1) aufweist.

6. Vorrichtung (1) gemäß einem der Ansprüche 1 bis 5, wobei das genannte zweite Betriebsfluid (R2), das in dem genannten zweiten Betriebskreislauf (100) umläuft, eine Wasserlösung oder sogar Wasser allein ist.

7. Vorrichtung (1) gemäß einem der Ansprüche 1 bis 4, wobei das genannte erste Betriebsfluid R600 ist und wobei das genannte zweite Betriebsfluid (Z)-2-Buten ist.

8. Vorrichtung (1) gemäß einem oder mehreren der Ansprüche 1 bis 7, wobei die genannten Verdampfungsmittel (111) des genannten zweiten Fluids (R2) und die genannten Kondensationsmittel (31) des genannten ersten Fluids (R1) in einem selben Wärmetauscher (50) integriert sind, so dass die von der Kondensation des ersten Fluids (R1) abgeleitete Wärmeenergie ohne Zwischen-übergänge direkt an das zweite Fluid (R2) übertragen wird.

9. Heizungsinstallation, **dadurch gekennzeichnet, dass** sie eine Heizvorrichtung (1) gemäß einem oder mehreren der Ansprüche 1 bis 8 umfasst.

Revendications

1. Dispositif de chauffage (1) avec un cycle thermodynamique irréversible, comprenant :

- un premier circuit (10) pour la circulation d'un premier fluide de fonctionnement (R1), ledit premier circuit (10) comprenant :
    - un moyen d'évaporation (11) dudit premier fluide de fonctionnement (R1), ledit moyen d'évaporation (11) éliminant l'énergie thermique d'un flux d'eau d'alimentation (F) pour évaporer ledit premier fluide de fonctionnement (R1) ;
    - un moyen de compression (21) dudit premier fluide de fonctionnement (R1) qui comprime ledit premier fluide (R1) après son évaporation ;
    - un moyen de condensation (31) dudit premier fluide de fonctionnement (R1) qui condense ledit premier fluide (R1) après sa compression ;
    - un moyen d'expansion (41) dudit fluide de fonctionnement (R1) ;

- un second circuit (100) pour la circulation d'un second fluide de fonctionnement (R2), ledit second circuit (100) comprenant :

    - un moyen d'évaporation (111) dudit second fluide (R2) qui fait évaporer ledit second fluide (R2) au moyen de l'énergie thermique provenant de la condensation dudit premier fluide (R1) dudit premier circuit (10) ;
    - un moyen de compression (121) dudit second fluide (R2) qui comprime ledit second fluide de fonctionnement (R2) après son évaporation ;
    - un moyen de condensation (131) dudit second fluide (R2) qui condense ledit second fluide de fonctionnement (R2) après sa compression, ledit moyen de condensation (131) dudit second fluide (R2) chauffant un flux d'eau de distribution (Hman) au moyen de l'énergie thermique provenant de ladite condensation dudit second fluide (R2) ;
    - un moyen d'expansion (141) dudit second fluide de fonctionnement (R2) ;

dans lequel :

- ledit premier circuit (10) comprend un premier moyen de refroidissement (72) fourni fonctionnellement entre ledit moyen de condensation (31) et ledit moyen d'expansion (41) dudit premier circuit (10), ledit premier moyen de refroidissement (72) refroidissant ledit premier fluide de fonctionnement (R1) ;
- ledit second circuit (100) comprend ledit second moyen de refroidissement (71) fourni fonctionnellement entre ledit moyen de condensation (131) et ledit moyen d'expansion (141) dudit second fluide (R2), de façon à refroidir ledit second fluide (R2) après sa condensation et **caractérisé en ce que**
- ledit premier moyen de refroidissement (72) chauffe un premier flux partiel (F1) dudit flux d'eau d'alimentation (F) ;
- ledit second moyen de refroidissement (71) chauffe un deuxième flux partiel (F2) dudit flux d'eau d'alimentation (F) indépendant dudit premier flux partiel (F1) ;
- ledit dispositif (1) comprend en outre un col-

lecteur de distribution (81) et un collecteur de retour (82) de ladite eau d'alimentation ;
- ledit collecteur de distribution (81) comprenant au moins un premier orifice d'admission pour ledit flux d'eau d'alimentation (F), un premier orifice de refoulement pour ledit premier flux partiel (F1) dudit flux d'eau d'alimentation (F) et au moins un deuxième orifice de refoulement pour ledit deuxième flux partiel (F2) dudit flux d'eau d'alimentation (F) ;
- ledit collecteur de retour (82) comprenant au moins un orifice d'admission pour ledit premier flux partiel (F1) et au moins un deuxième orifice d'admission pour ledit deuxième flux partiel (F2) provenant dudit second moyen de refroidissement (71), ledit collecteur de retour (82) comprenant un orifice de refoulement principal pour ledit flux d'eau d'alimentation (F) prévu pour ledit moyen d'évaporation (11) dudit premier circuit (10).

2. Dispositif (1) selon la revendication 1, dans lequel ledit collecteur de distribution (81) comprend un troisième orifice de refoulement pour un troisième flux partiel (F3) dudit flux d'alimentation indépendant dudit premier flux partiel (F1) et dudit deuxième flux partiel (F2), ledit premier moyen de compression (21) dudit premier circuit (10) étant configuré de manière à échanger de l'énergie thermique avec ledit troisième flux partiel (F3) pour chauffer ce dernier, ledit collecteur de retour (82) comprenant un troisième orifice d'admission pour ledit troisième flux partiel (F3) chauffé au moyen de l'échange thermique avec ledit moyen de compression (21) dudit premier circuit (10).

3. Dispositif (1) selon la revendication 1 ou 2, dans lequel ledit collecteur de distribution comprend un quatrième orifice de refoulement pour un quatrième flux partiel (F4) dudit flux d'eau d'alimentation (F) indépendant dudit premier flux partiel (F1) et dudit deuxième flux partiel (F2), ledit second moyen de compression (121) dudit second circuit (100) étant configuré de manière à échanger de l'énergie thermique avec ledit quatrième flux partiel (F4) pour chauffer ce dernier, ledit collecteur de retour (82) comprenant un quatrième orifice d'admission pour ledit quatrième flux partiel (F4) chauffé au moyen de l'échange thermique avec ledit moyen de compression (121) dudit second circuit (100).

4. Dispositif selon l'une quelconque des revendications 1 à 3, dans lequel ledit collecteur de distribution (81) et ledit collecteur de retour (82) sont raccordés hydrauliquement au moyen d'une ligne hydraulique de compensation (L).

5. Dispositif (1) selon l'une quelconque des revendications 1 à 4, dans lequel ledit second fluide de fonctionnement (R2) a une densité inférieure à celle dudit premier fluide de fonctionnement (R1).

6. Dispositif (1) selon l'une quelconque des revendications 1 à 5, dans lequel ledit second fluide de fonctionnement (R2) circulant dans ledit second circuit de fonctionnement (100) est une solution d'eau ou même de l'eau uniquement.

7. Dispositif (1) selon l'une quelconque des revendications 1 à 4, dans lequel ledit premier fluide de fonctionnement est du R600 et dans lequel ledit second fluide de fonctionnement est du (Z)-2-butène.

8. Dispositif (1) selon une ou plusieurs des revendications 1 à 7, dans lequel ledit moyen d'évaporation (111) dudit second fluide (R2) et ledit moyen de condensation (31) dudit premier fluide (R1) sont intégrés dans un même échangeur de chaleur (50), de sorte que l'énergie thermique provenant de la condensation du premier fluide (R1) est directement transférée au second fluide (R2) sans passages intermédiaires.

9. Installation de chauffage, **caractérisée en ce qu'**elle comprend un dispositif de chauffage (1) selon une ou plusieurs des revendications 1 à 8.

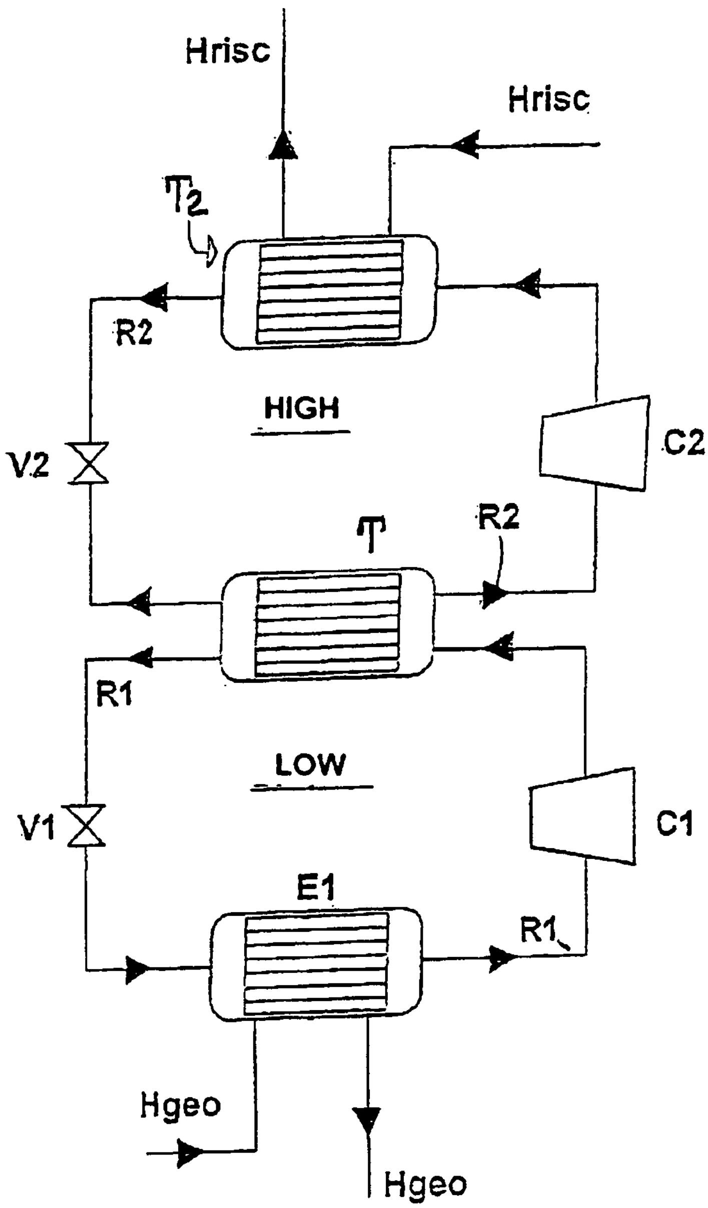


Figure 1

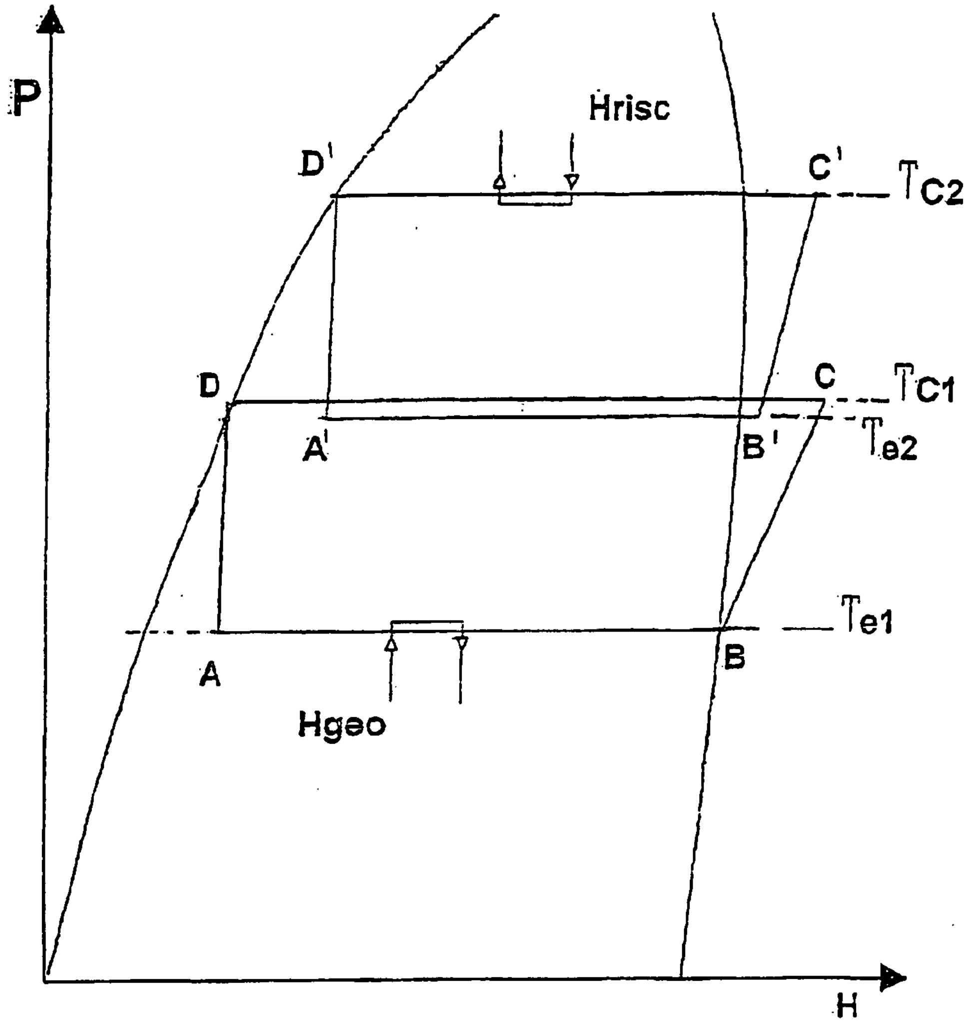
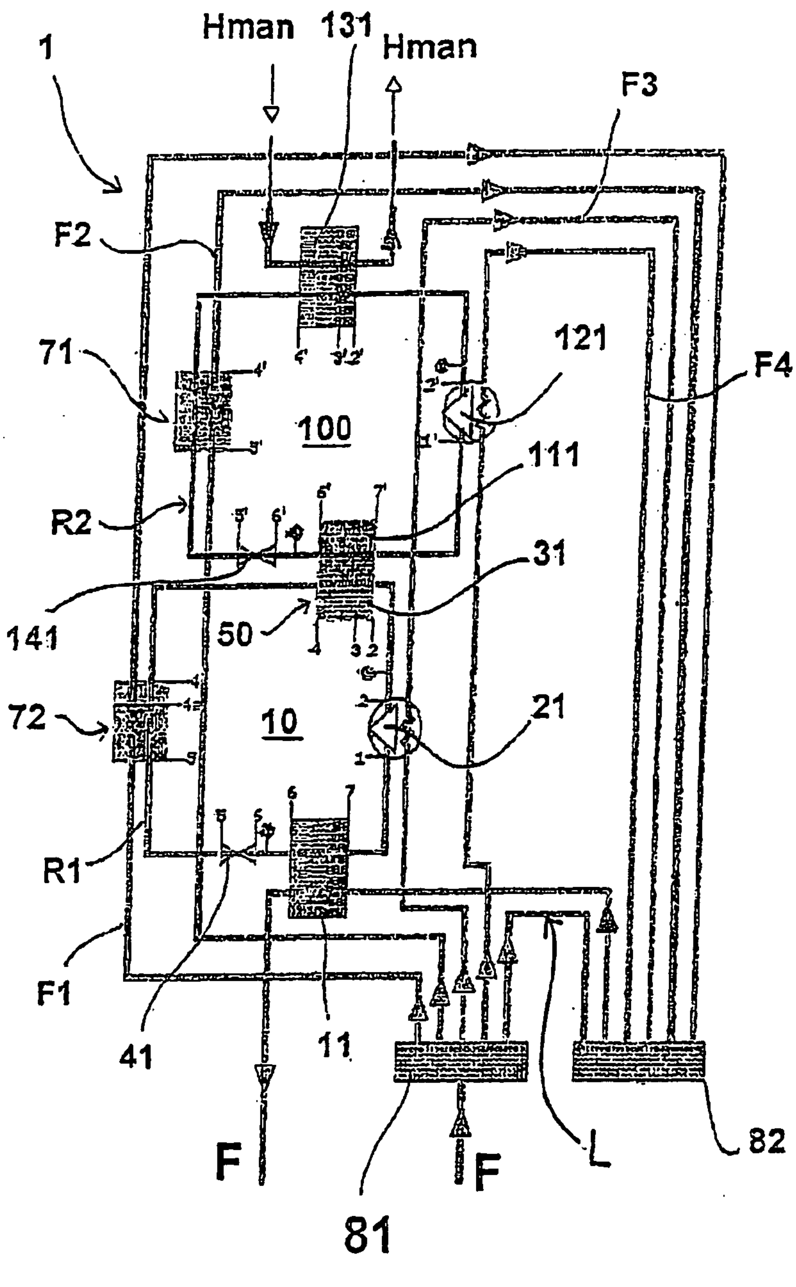


Figure 2



1
Hman
131
Hman
F3
F2
71
121
F4
100
111
R2
31
141
50
72
10
21
R1
F1
41
11
F
F
L
82
81

Figure 3

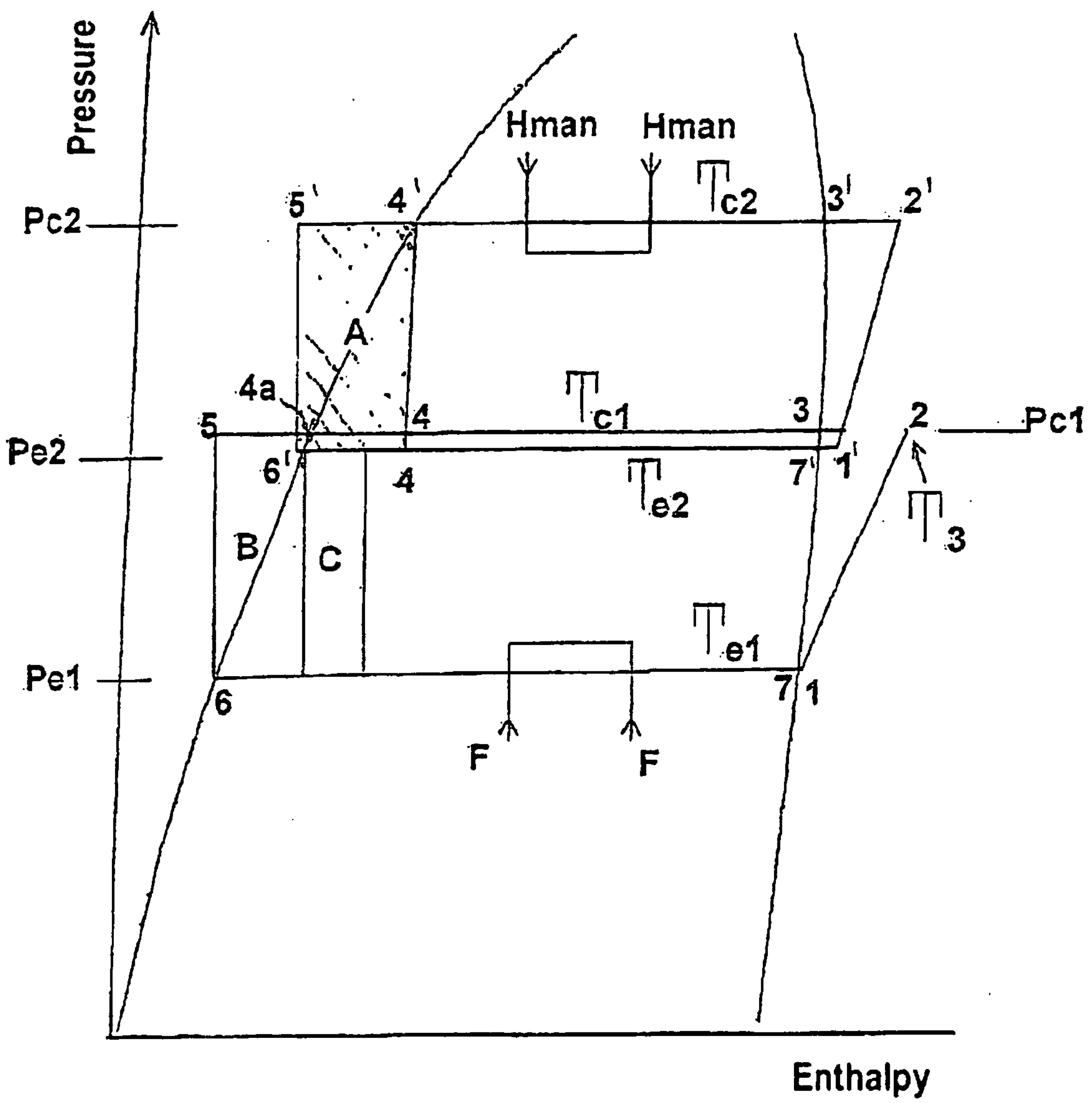

Pressure
Hman
Hman
Pc2
5'
4'
Tc2
3'
2'
A
4a
Tc1
4
3
2
Pc1
5
Pe2
6'
4
Te2
7'
1'
T3
B
C
Te1
Pe1
6
7
1
F
F
Enthalpy

Figure 4

REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

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