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(54) **Appliance for treating articles**

(57) An appliance for drying laundry items (**1**) having a heat pump system (**5**), the heat pump system having a refrigerant loop, the appliance comprising: a laundry treatment chamber (**2**) for drying laundry using a drying medium; a first heat exchanger (**6**) for heating a refrigerant and cooling the medium; a second heat exchanger (**7**) for cooling the refrigerant and heating the drying medium; a refrigerant expansion device (**14**) arranged in the refrigerant loop between the second heat exchanger (**7**) and the first heat exchanger (**6**), and a compressor (**8**) arranged in the refrigerant loop between the first heat exchanger (**6**) and the second heat exchanger (**7**). The refrigerant is a zeotropic blend having a glide level higher than about 7 °C at a bubble temperature of about 20 °C, and a glide level higher than about 5 °C at a bubble temperature of about 60 °C.

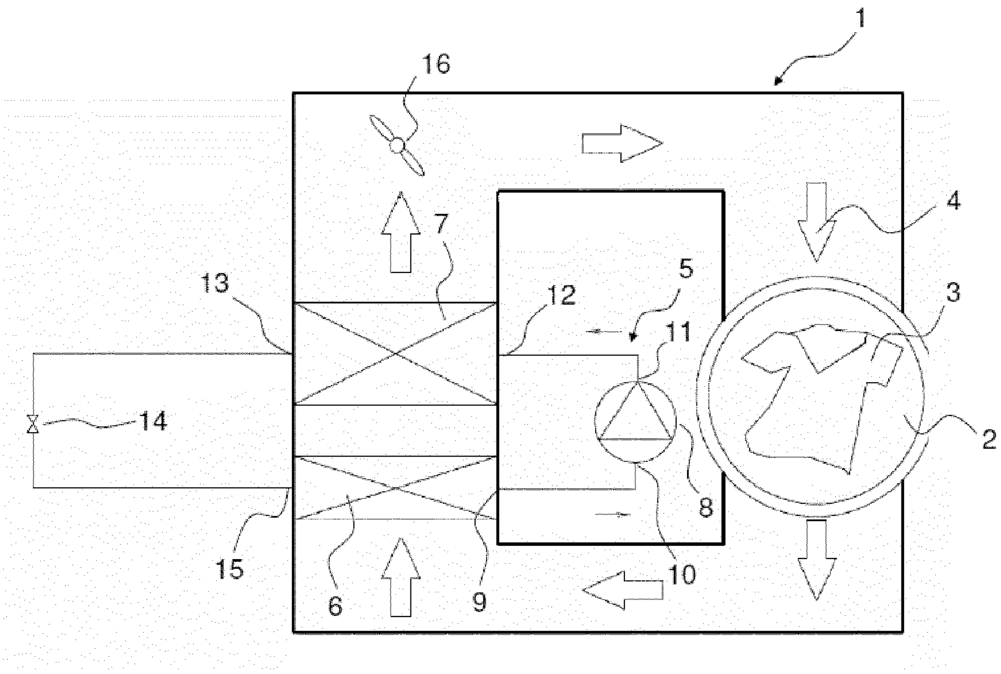

FIG. 1

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Description

Background of the invention

Field of the invention

[0001] The present invention relates to appliances for treating articles, and is in particular directed to appliances for treating laundry, like laundry washers and laundry washers/dryers.

Overview of the related art

[0002] Appliances for drying laundry, like laundry dryers (tumble dryers) and laundry washers/dryers, generally comprise a drying chamber for accommodating therein the laundry to be dried. A heated and dehumidified drying medium, typically air, is guided through the drying chamber. Upon passing through the drying chamber and laundry, the heated and dehumidified drying medium takes up humidity and at the same time cools down. The drying medium then exits the drying chamber, thereby discharging humidity from the drying chamber and the laundry.

[0003] In order to improve the energy efficiency of such appliances, it is known to use heat pumps. In this way, residual heat from the drying medium exiting the drying chamber can be extracted therefrom and transferred again to the drying medium before it re-enters the drying chamber.

[0004] Heat pumps generally operate with refrigerants as working fluids.

[0005] Known fluids that are currently used as refrigerants in heat pump tumble dryers belong to different chemical families, such as HydroChloroFluoroCarbons (HCFC), HydroFluoroCarbons (HFC), hydrocarbons, and can be pure fluids or blends of fluids.

Summary of the invention

[0006] The choice of the fluid to be used as refrigerant in a heat pump appliance for treating articles drastically affects the performance of the appliance, because of the thermal properties of the refrigerant.

[0007] The Applicant has found that the performance of a heat pump appliance for treating articles, in particular laundry or tableware, like laundry washers, laundry washers/dryers, laundry dryers, dishwashers, are greatly enhanced if the fluid used as the refrigerant in the heat pump has a high level of glide.

[0008] The glide is the difference between the temperature of the saturated vapor (the "dew point temperature" or "dew temperature") and the temperature of the saturated liquid (the "bubble point temperature" or "bubble temperature"), at a fixed pressure level, or at the dew temperature of the refrigerant fluid, or at the bubble temperature of the refrigerant fluid.

[0009] Pure fluids or azeotropic blends are characterized by the fact that the temperature stays at a constant level during the isobaric change of phases (evaporation and condensation). Thus, in pure fluids or azeotropic blends the temperature of the saturated vapor (dew temperature) and the temperature of the saturated liquid (bubble temperature) coincide, so these fluids have an essentially zero glide level.

[0010] Zeotropic blends are instead characterized by the fact that the temperature does not remain constant during the isobaric change of phases: the temperature increases during the evaporation and decreases during the condensation. Thus, the bubble temperature is different from, being lower than, the dew temperature. Zeotropic blends have a non-zero level of glide.

[0011] In a heat pump, the presence of a certain level of glide allows a better matching between the drying air and the refrigerant temperature profiles at the evaporator and at the condenser. The temperature difference between the drying air and the refrigerant decreases in case of zeotropic refrigerants, then the efficiency of the heat pump system increases. A more efficient heat pump systems translates into a lower energy consumption and less time needed to dry the objects.

[0012] The glide level varies depending on the fluid used as refrigerant, and, for the same refrigerant fluid, it varies depending on the fluid temperature and/or pressure. For the purpose of the present invention, by the expression "high level of glide" it is meant that the refrigerant fluid has a high level of glide at least in the range of temperatures that are typical for the evaporation and condensation of a refrigerant in a heat pump laundry dryer or washer/dryer. For example, the refrigerant should have a glide level higher than about 7 °C at a bubble temperature of about 20 °C (typical of the refrigerant evaporation) and a glide level higher than about 5 °C at a bubble temperature of about 60 °C (typical of the refrigerant condensation).

[0013] According to the present invention, there is provided an appliance for drying laundry items having a heat pump system, the heat pump system having a refrigerant loop. The appliance comprises:

- a laundry treatment chamber for drying laundry using a drying medium;
- a first heat exchanger for heating a refrigerant and cooling the medium;
- a second heat exchanger for cooling the refrigerant and heating the drying medium;
- a refrigerant expansion device arranged in the refrigerant loop between the second heat exchanger and the first heat exchanger, and
- a compressor arranged in the refrigerant loop between the first heat exchanger and the second heat exchanger.

[0014] The refrigerant is a zeotropic blend having a glide level higher than about 7 °C at a bubble temperature of about 20 °C, and a glide level higher than about 5 °C at a bubble temperature of about 60 °C.

[0015] Preferably, the refrigerant is selected in the group comprising: HFC, HCFC, HFO (Hydro-Fluoro-Olefines), blends of hydrocarbons, particularly blends of alkanes type molecules.

[0016] Preferably, said alkanes type molecules have general formula C_nH_{2n+2} with $1 \leq n \leq 8$, preferably $2 \leq n \leq 4$. A suitable refrigerant is R441a, which is a blend composed by a mix of Ethane (C_2H_6), Propane (C_3H_8), Buthane (C_4H_{10}) and Isobuthane (2-metil Propane) (C_4H_{10}).

[0017] Preferably, the weight ratio composition of the hydrocarbons in the mixture is in the range of: from 2 to 4 of Ethane; from 40 to 70 of Propane; from 25 to 45 of Butane; from 4 to 8 of Isobutane, preferably 3.1 : 54.8 : 36.1 : 6.0 of Ethane : Propane : Butane : Isobutane, respectively.

[0018] Preferably, a critical temperature of the refrigerant is in a range from about 105 °C to about 130 °C, preferably 117.3 °C.

[0019] Preferably, a critical pressure of the refrigerant is in a range from about 39 bar 25 to about 49 bar, preferably 44.04 bar.

[0020] Preferably, the nominal heat of vaporization of the refrigerant is in a range from about 430 kJ/kg to about 470 kJ/kg, preferably 451.5 kJ/kg.

[0021] Preferably, the global warming potential index of the refrigerant in a range from 1 to 10, preferably less than 5.

[0022] Advantageously, the refrigerant has a nominal glide in a range from about 18 °C to about 22 °C, preferably about 20 °C at an absolute pressure of 1 bar.

[0023] Preferably, the refrigerant has a glide of about 16 °C to 20 °C, preferably 18.2 °C at a pressure of 5 bar.

[0024] Preferably, the refrigerant has a glide of about 14.4 °C at a pressure of 15 bar.

[0025] In embodiments of the present invention, a nominal first heat exchanger inlet temperature of the medium is about 30° C at the least.

[0026] In embodiments of the present invention, the second heat exchanger comprises a liquefier heat exchanger; a nominal liquefier heat exchanger outlet temperature of the medium may be 100 °C at the most.

[0027] The appliance may comprise a closed-loop circuit wherein the medium circulates.

[0028] Preferably, the heat pump system has a nominal cooling power between about 500 W and 3,500 W, in particular between 1,500 W and 3,500 W.

Brief description of the drawings

[0029] These and other features and advantages of the present invention will be made clearer by reading the following detailed description of exemplary and non-limitative embodiments thereof, referring to the following drawing figures, wherein:

Figure 1 schematically shows a heat pump laundry dryer according to an embodiment of the present invention;

Figure 2 is a pressure versus enthalpy diagram of R407c;

Figures 3, 4 and **5** are temperature-entropy diagrams of different refrigerants and of a refrigerant according to an embodiment of the present invention;

Figure 6 is a comparative diagram showing the different glide level versus bubble point twemperature of R407c and R441a;

Figure 7 is a diagram showing the latent heat of different refrigerants and of a refrigerant according to an embodiment of the present invention;

Figure 8 is an isometric view of the heat pump laundry dryer, with one lateral wall of an appliance cabinet removed, and

Figure 9 shows in exploded view a basement of the laundry dryer of

Figure 8 configured for accommodating the heat pump.

Detailed description of exemplary embodiments of the invention

[0030] Exemplary embodiments of the invention will be described in connection with the annexed figures.

[0031] **Figure 1** schematically shows a heat pump laundry dryer **1** according to an embodiment of the present invention. It is pointed out that although in the following description a heat pump laundry dryer is considered, this choice is merely exemplary, because the present invention applies generally to laundry washers/dryers as well equipped with a heat pump for in heat-exchange relationship with an article drying medium.

[0032] The heat pump laundry dryer **1** comprises a drying chamber **2,** preferably a rotatable drum. In operation, the drying drum **2** accommodates wet laundry **3** to be dried. For adequately drying the laundry **3,** a drying medium **4,** such as air, in particular comprising ambient air, is circulated through the drying drum **2** via a drying medium circuit, which preferably forms a closed-loop circuit.

[0033] For drying the laundry **3,** the drying medium **4** heated to a temperature of 100°C at the most and thereby having a comparatively low relative humidity is fed into the drying drum **2** and impinges the wet laundry **3.** As a consequence, humidity of the wet laundry **3** is absorbed by the drying medium **4** thereby drying the laundry **3.** As the laundry **3** in the drying drum **2** generally has a temperature below that of the drying medium **4** entering the drying drum **2,** the drying medium **4** also cools down, for example to temperatures of about 30°C.

[0034] After having passed through the drying drum **2,** the drying medium **4,** having a comparatively high relative humidity, exits the drying drum **2** and is further cooled down to condense excess humidity therefrom. After that, the drying medium **4** is recirculated through the drying drum **2.** Before re-entering the drying drum **2,** the drying medium **4** is heated up again, thereby reducing its relative humidity.

[0035] For dehumidifying and reheating the drying medium **4,** the heat pump tumble dryer **1** comprises a heat pump unit **5.** The heat pump unit **5** exemplarily comprises a heat exchange medium evaporator **6** and a heat exchange medium liquefier **7.** Hereinafter, the term "refrigerant" will also be used as a synonym of the term "heat exchange medium".

[0036] The heat pump unit **5** further comprises a compressor **8** interconnected between the refrigerant evaporator **6** and the refrigerant liquefier **7.** A refrigerant evaporator outlet **9** is connected to a compressor inlet **10** and a compressor outlet **11** is connected to a refrigerant liquefier inlet **12.**

[0037] A refrigerant liquefier outlet **13** is connected via a throttling element **14,** a capillary for example, to a refrigerant evaporator inlet **15.**

[0038] By the heat pump unit **5,** heat is transferred from the refrigerant evaporator **6** to the refrigerant liquefier **7.**

[0039] A refrigerant, i.e. a heat exchange medium, circulated in the heat pump unit closed circuit, is heated up at the refrigerant evaporator **6** and cooled down at the refrigerant liquefier **7.**

[0040] The relatively low temperature at the refrigerant evaporator **6** is used to cool down the drying medium **4** so as to condensate humidity, i.e. to dehumidify the drying medium **4** exiting the drying drum **2.**

[0041] The elevated temperature at the refrigerant liquefier **7** is used to reheat the drying medium **4** which in turn is then fed to the drying drum **2** for drying the laundry **3.**

[0042] The heat pump unit **5** may further comprise auxiliary heat exchangers for further optimizing energy efficiency. For example, an auxiliary refrigerant evaporator and an auxiliary refrigerant liquefier may be provided. The number of auxiliary refrigerant heat exchangers can be varied from one to nearly any arbitrary number. An auxiliary refrigerant evaporator may be used to speed up the heat-up phase of the heat pump dryer and a refrigerant liquefier may be used to balance the excess of energy of the heat pump dryer.

[0043] The direction of refrigerant flow is indicated in **Figure 1** by small arrows, whilst the flow of the drying medium **4** is indicated by larger and broader arrows. The heat pump tumble dryer **1** may comprise a fan **16** adapted to and designed for circulating the drying medium **4** within the heat pump tumble dryer circuit.

[0044] In connection with the heat pump tumble dryer **1** described so far, the Applicant has found that is advantageous to use a refrigerant, i.e. heat exchange medium, that has a high level of glide.

[0045] The glide is the difference between the temperature of the saturated vapor (the "dew temperature") and the temperature of the saturated liquid (the "bubble temperature"), at a fixed pressure level, or at the dew point temperature of the refrigerant fluid, or at the bubble temperature of the refrigerant fluid.

[0046] **Figure 2** is the pressure over enthalpy diagram of a refrigerant fluid, particularly R407c. The saturated liquid curve and the saturated vapor curve distinguish three different zones, corresponding to three different states of the refrigerant, that are typical for all refrigerants (the shape and the values of the curves are peculiar for each specific refrigerant; those shown refer to R407c).

[0047] The points that belong to the saturated liquid curve represent the saturated liquid condition: the refrigerant is all in liquid phase at the saturation temperature related to the actual pressure. The saturation temperature of the liquid state is also called bubble temperature.

[0048] The points that belong to the saturated vapor curve represent the saturated vapor condition: the refrigerant is all in vapor phase at the saturation temperature related to the actual pressure. The saturation temperature of the vapor state is also called dew temperature.

[0049] The zone at the left of the saturated liquid zone is the sub-cooled liquid zone: the refrigerant is in liquid phase at a temperature lower than saturation temperature (at a certain pressure).

[0050] The zone between the two curves is the two phase zone: in this zone liquid and vapor phase coexist at the same time. The relative amount between liquid and vapor phase change according to the position in the two-phase zone.

[0051] The zone at the right of the saturated vapor zone is the superheated vapor zone: the refrigerant is in vapor phase at a temperature higher than saturation (at a certain pressure).

[0052] The saturated condition (liquid and vapor) can be seen as a limit condition for the maintenance of the complete liquid and vapor condition respectively. A little movement in the diagram toward right and left respectively would bring the refrigerant into two phase status (liquid plus vapor).

[0053] Pure fluids or azeotropic blends are characterized by the fact that the temperature stays at a constant level during the isobaric change of phases (evaporation and condensation). Thus, in pure fluids or azeotropic blends the temperature of the saturated vapor (dew temperature) and the temperature of the saturated liquid (bubble temperature) coincide, so these fluids have an essentially zero glide level.

[0054] Zeotropic blends are instead characterized by the fact that the temperature does not remain constant during the isobaric change of phases: the temperature increases during the evaporation phase and decreases during the condensation phase. Thus, the bubble temperature is different from, being lower than, the dew temperature. Zeotropic blends have a non-zero level of glide.

[0055] In a heat pump, the presence of a certain level of glide allows a better matching between the drying air and the refrigerant temperature profiles at the evaporator and at the condenser. The temperature difference between the drying air and the refrigerant decreases in case of zeotropic refrigerant, then the efficiency of the heat pump system increases. A more efficient heat pump system translates into a lower energy consumption and less time needed to dry the objects.

[0056] The glide level varies depending on the fluid used as the refrigerant, and, for the same refrigerant fluid, it varies depending on the fluid temperature and/or pressure. By the expression "high level of glide" it is meant that the refrigerant fluid has a high level of glide at least in the range of temperatures that are typical for the evaporation and condensation of a refrigerant in a heat pump dryer.

[0057] For example, the refrigerant should have a glide level higher than about 7 °C at a bubble temperature of about 20 °C (typical of the refrigerant evaporation phase) and a glide level higher than about 5 °C at a bubble temperature of about 60 °C (typical of the refrigerant condensation phase).

[0058] The Applicant has found that suitable refrigerant fluids are HFC, HCFC, HFO, blends of hydrocarbons, particularly blends composed by a mix of simple hydrocarbon alkane compounds, each compound having general formula C_nH_{2n+2}.

[0059] The Applicant has found that a suitable mix of simple hydrocarbon alkane compounds that can advantageously be used as heat exchange medium is R441a. R441a is a blend composed by a mixture that is characterized by: Ethane (C_2H_6), Propane (C_3H_8), Buthane (C_4H_{10}) and Isobuthane (2-metil Propane) (C_4H_{10}).

[0060] The Applicant has found that a suitable weight ratio composition of the hydrocarbons in the mixture is in the range of: from 2 to 4 of Ethane; from 40 to 70 of Propane; from 25 to 45 of Butane; from 4 to 8 of Isobutane. A particularly preferable weight ratio is 3.1 : 54.8 : 36.1 : 6.0 of Ethane : Propane : Butane : Isobutane, respectively.

[0061] R441a is composed by natural fluids, so it is more eco-friendly than both HFCs and HFOs, which are compounds normally used as refrigerant fluids in refrigeration systems. Compared to HFC and HFO, R441a does not contain Halogen atoms such as Cl (Chlorine) and F (Fluorine), containing only C (carbon) and H (hydrogen) atoms.

[0062] HFO molecules are characterized by an unsaturation (olefin-based molecules). The unsaturation (double C bonds) could be easily degraded by oxidants present in the system. Temperature, pressure and metals could catalyse the reaction. The secondary product of this reaction could be acidic molecules that provide some corrosion phenomena in the system. Differently, the degradation of R441a does not provide this kind of unwanted corrosive effect.

[0063] Additionally, the GWP (Global Warming Potential) of R441 a is close to zero.

[0064] Another refrigerant fluid that exhibits glide is R407c, but its glide level if significantly lower compared to R441a.

[0065] The temperature-entropy diagrams of **Figures 3, 4** and **5** relate to the the refrigerant thermodynamic cycle in the case of R134a **(Figures 3A** and **3B),** R407c **(Figures 4A** and **4B)** and R441a **(Figures 5A** and **5B);** the entropy (s, in [kJ/kg K]) is on the abscissa, while the temperature **(T,** in [°C]) is on the ordinate. **SLC** denotes the saturated liquid curve, **SVC** denotes the saturated vapor curve. On the same diagrams the temperature profile of the air (dashed line) is also shown (only the temperature value is relevant; the entropy values refer only to the refrigerant).

[0066] The cycle depicted in solid line (1 -> 2 -> 3-> 4) represents the thermodynamic cycle of the refrigerant; in particular, state 1 is the state of the refrigerant at the liquefier inlet **12,** state 2 is the state of the refrigerant at the liquefier outlet **13,** state 3 is the state of the refrigerant at the evaporator inlet **15,** and state 4 is the state of the refrigerant at the evaporator outlet **9.** In respect of the drying air, A is the temperature of the drying air upon leaving the drum **2** and entering the evaporator **6,** B is the temperature of the drying air after having passed through the evaporator **6** and before passing through the liquefier 7, and C is the temperature of the drying air after having passed through the liquefier **7** before re-entering the drum **2.**

[0067] Looking at **Figures 3A** and **3B,** it can be appreciated that R134a has no glide: the temperature remains constant during the isobaric changes of phase (evaporation and condensation); the difference in temperature between the refrigerant and the drying air increases during the refrigerant transitions of phase.

[0068] Looking at **Figures 4A** and **4B,** it can be appreciated that in the case R407c, another HFC but having a certain level of glide, the temperature does not remain constant during the refrigerant transitions of phase, so that there is a better matching between the temperature of the refrigerant and that of the drying air. "Better matching" means that the temperature difference between refrigerant and drying air is lower along the heat exchanger. Therefore, the drying air can be heated up at a higher level, or the average condensation temperature can be lower in the liquefier, and the air can be cooled down at a lower level, or the average evaporation temperature can be higher in the evaporator. This leads to a higher heating and cooling capacity, or to a lower power consumption keeping the same thermal capacity (the closer the condensation and evaporation temperature levels, the higher the efficiency of the heat pump system).

[0069] The situation improves in the case of R441a, which has a higher level of glide compared to R407c, as shown in **Figure 6** (wherein the glide, in ordinate, [°C] is shown as a function of the bubble temperature, in abscissa, [°C]). As visible in **Figures 5A** and **5B,** the matching between the refrigerant temperature during the change of phase and the temperature of the drying air is quite good, and the difference between these temperatures is lower than in the case of the other refrigerants.

[0070] **Figure 7** is a diagram showing the latent heat (in ordinate, [kJ/kg]) as a function of the bubble temperature (in abscissa, [°C]) of R134a, R407c and R441a. It can be seen that R441a has a higher latent heat compared to base hydrocarbons, especially at high temperatures (the liquefying temperature is usually around 55-70 °C in a heat pump dryer, to allow the drying air to be warmed up at desired levels, while the evaporation temperature is around 15-30 °C). Refrigerants having a higher latent heat level are able to exchange a great amount of heat with a relative low refrigerant mass flow rate, or they are able to exchange the same heat amount with a lower refrigerant mass flow rate. This means that the compressor has to move lower refrigerant mass, with less electrical power consumption.

[0071] The Applicant has found to be of particular advantage for the heat exchange processes that the refrigerant has at least one of the following properties:

- belongs to the family of the hydrocarbons
- has a critical temperature of about 105 °C to 130 °C, preferably about 117.3 °C;
- has a critical pressure of about 39 bar to 49 bar, preferably about 44.04 bar;
- has a nominal heat of vaporization of about 430 kJ/kg to 470 kJ/kg, preferably about 451.5 kJ/kg ("nominal" means at an absolute pressure equal to 1 bar);
- has a nominal glide of 18 °C to 22 °C, preferably about 20°C ("nominal" means at a absolute pressure equal to 1 bar);
- has a heat of vaporization of about 357 kJ/kg to 437 kJ/kg, preferably about 397.6 kJ/kg at a pressure equal to 5 bar;
- has a glide of about 16 °C to 20 °C, preferably 18.2 °C at a pressure equal to 5 bar;
- has a global warming potential index of about 1 to 10 (preferably less than 5).

[0072] More preferably, the heat of vaporization is of about 322.6 kJ/kg at a pressure equal to 15 bar, and the glide of about 14.4°C at a pressure equal to 15 bar.

[0073] Preferably, the heat pump system has a nominal cooling power between about 500 W and 3,500 W, in particular between 1,500 W and 3,500 W.

[0074] **Figure 8** is an isometric view of an exemplary laundry dryer. The laundry dryer comprises a cabinet **800,** having lateral walls (one of which has been removed in the drawing) and housing the drying drum **2** and the heat pump unit **5.**

[0075] The heat pump unit **5** is for example housed in an appliance basement **805,** which is shown per se in **Figure 9.** The basement **900** is, in the shown example, a shell composed of two half-shells **805** and **810** designed to match each other so that, when matched, they define inside them a space for accommodating the heat pump system parts, like the evaporator **6,** the liquefier **7,** the compressor **8,** and passageways for the drying air.

Claims

1. An appliance for drying laundry items (1) having a heat pump system **(5),** the heat pump system having a refrigerant loop, the appliance comprising:

a laundry treatment chamber **(2)** for drying laundry using a drying medium;
a first heat exchanger **(6)** for heating a refrigerant and cooling the medium;
a second heat exchanger **(7)** for cooling the refrigerant and heating the drying medium;
a refrigerant expansion device **(14)** arranged in the refrigerant loop between the second heat exchanger **(7)** and the first heat exchanger **(6),** and
a compressor **(8)** arranged in the refrigerant loop between the first heat exchanger **(6)** and the second heat exchanger **(7),**
characterized in that
the refrigerant is a zeotropic blend having a glide level higher than about 7 °C at a bubble temperature of about 20 °C, and a glide level higher than about 5 °C at a bubble temperature of about 60 °C.

2. The appliance of claim 1, wherein the refrigerant is selected in the group comprising: HFC, HCFC, HFO, blends of hydrocarbons, particularly blends of alkanes type molecules.

3. The appliance of claim 2, wherein said alkanes type molecules have general formula C_nH_{2n+2} with $1 \leq n \leq 8$, preferably $2 \leq n \leq 4$.

4. The appliance of claim 3, wherein the refrigerant is R441a.

5. The apparatus of claim 4, wherein the weight ratio composition of the hydrocarbons in the mixture is in the range of from 2 to 4 of Ethane; from 40 to 70 of Propane; from 25 to 45 of Butane; from 4 to 8 of Isobutane, preferably 3.1 : 54.8 : 36.1 : 6.0 of Ethane : Propane : Butane : Isobutane, respectively.

6. The appliance of any one of the preceding claims, wherein a critical temperature of the refrigerant is in a range from about 105 °C to about 130 °C, preferably 117.3 °C.

7. The appliance of any one of the preceding claims, wherein a critical pressure of the refrigerant is in a range from about 39 bar 25 to about 49 bar, preferably 44.04 bar.

8. The appliance of any one of the preceding claims, wherein the nominal heat of vaporization of the refrigerant is in a range from about 430 kJ/kg to about 470 kJ/kg, preferably 451.5 kJ/kg.

9. The appliance of any one of the preceding claims, wherein the global warming potential index of the refrigerant in a range from 1 to 10, preferably less than 5.

10. The appliance of any one of the preceding claims, wherein the refrigerant has a nominal glide in a range from about 18 °C to about 22 °C, preferably about 20 °C at an absolute pressure of 1 bar.

11. The appliance of any one of the preceding claims, wherein the refrigerant has a glide of about 16 °C to 20 °C, preferably 18.2 °C at a pressure of 5 bar.

12. The appliance of any one of the preceding claims, wherein the refrigerant has a glide of about 14.4 °C at a pressure of 15 bar.

13. The appliance of any one of the preceding claims, wherein a nominal first heat exchanger inlet temperature of the medium is about 30° C at the least.

14. The appliance of any one of the preceding claims, wherein the second heat exchanger comprises a liquefier heat exchanger, and wherein a nominal liquefier heat exchanger outlet temperature of the medium is 100 °C at the most.

15. The appliance of any one of the preceding claims, comprising a closed-loop circuit wherein the medium circulates.

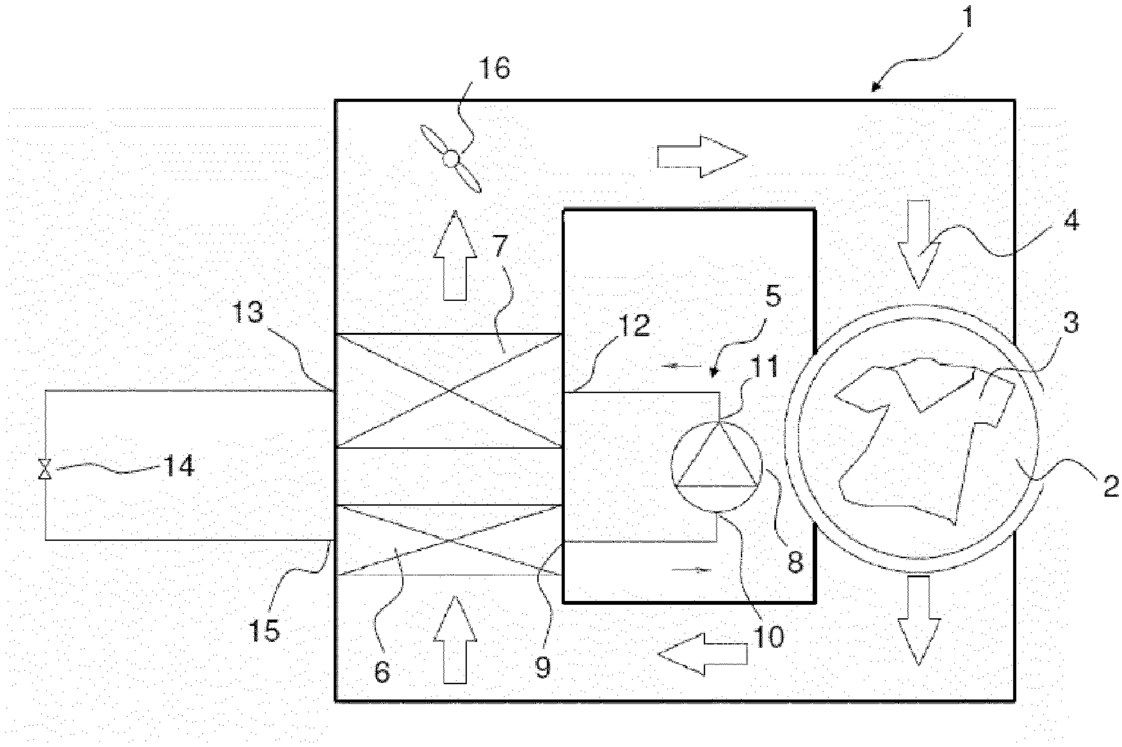


FIG. 1

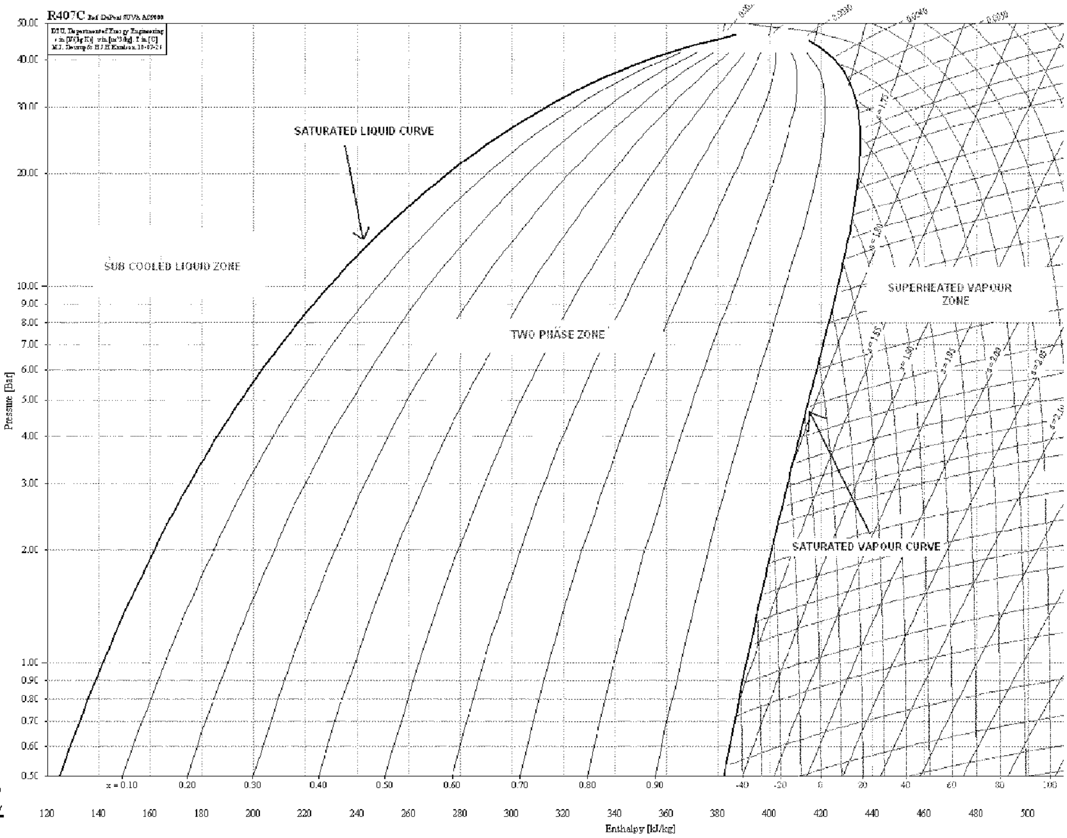

R407C
SATURATED LIQUID CURVE
SUB COOLED LIQUID ZONE
TWO PHASE ZONE
SUPERHEATED VAPOUR ZONE
SATURATED VAPOUR CURVE
Pressure [Bar]
Enthalpy [kJ/kg]

FIG. 2

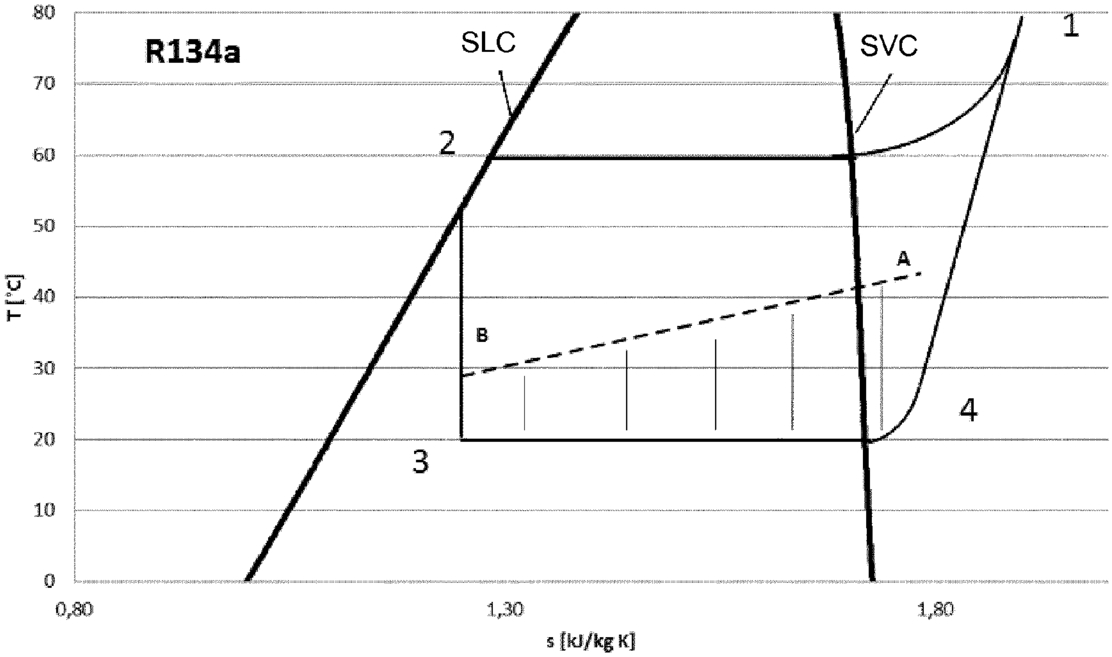

R134a
SLC
SVC
1
2
3
4
A
B
80
70
60
50
40
30
20
10
0
T [°C]
0,80
1,30
1,80
s [kJ/kg K]

FIG. 3A

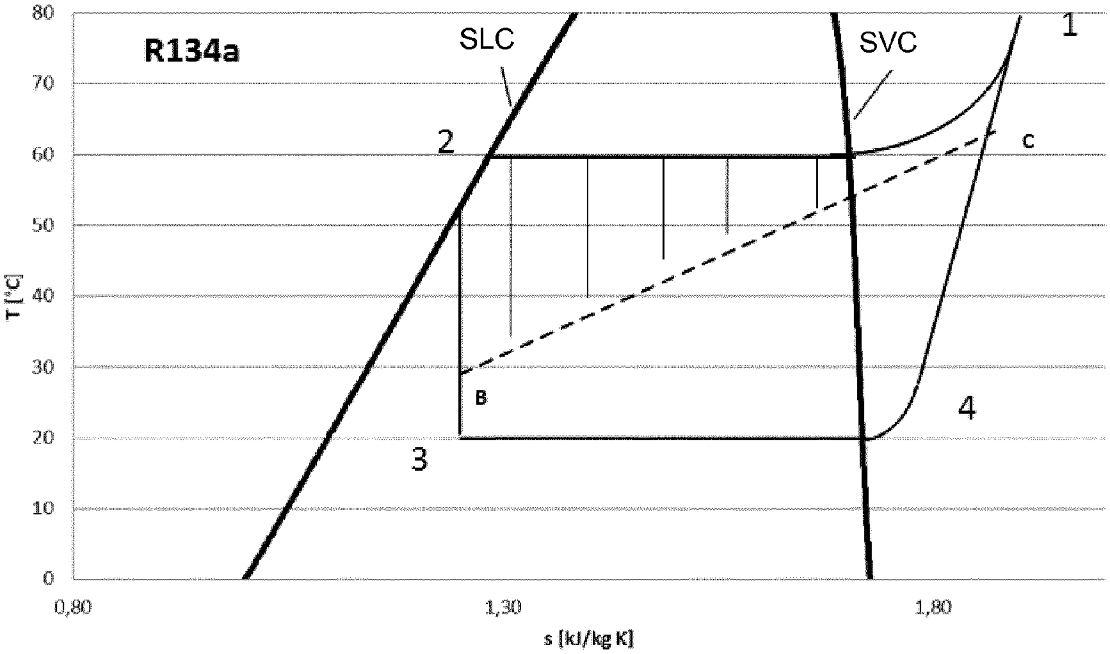

R134a
SLC
SVC
1
2
3
4
B
C
80
70
60
50
40
30
20
10
0
T [°C]
0,80
1,30
1,80
s [kJ/kg K]

FIG. 3B

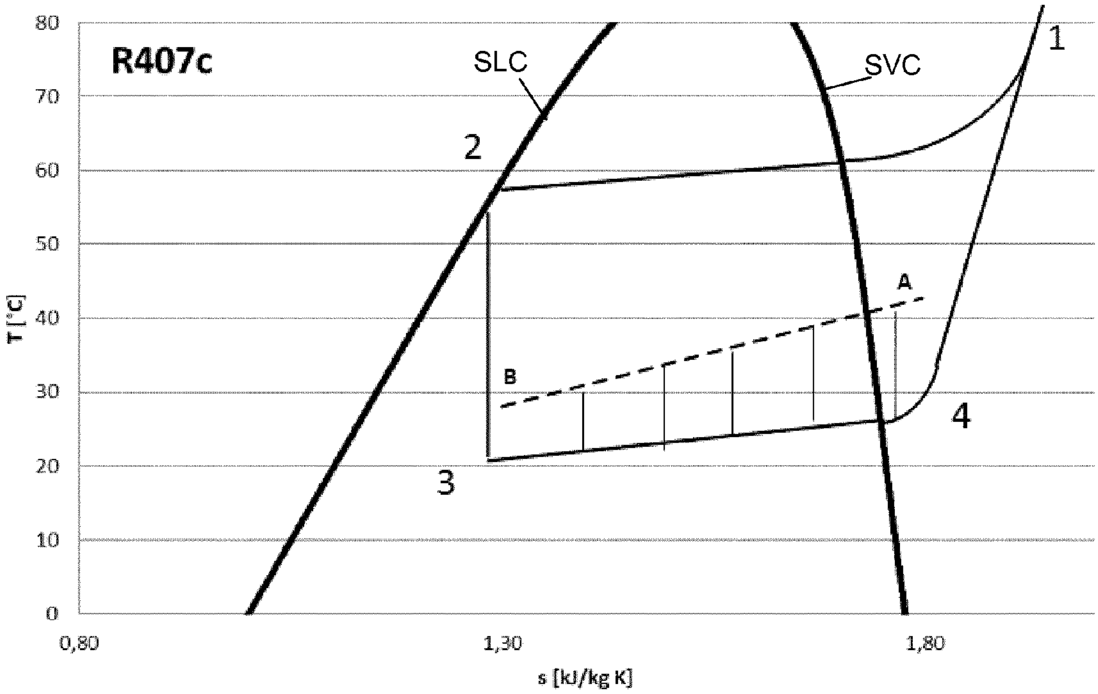

R407c
SLC
SVC
1
2
3
4
A
B
80
70
60
50
40
30
20
10
0
T [°C]
0,80
1,30
1,80
s [kJ/kg K]

FIG. 4A

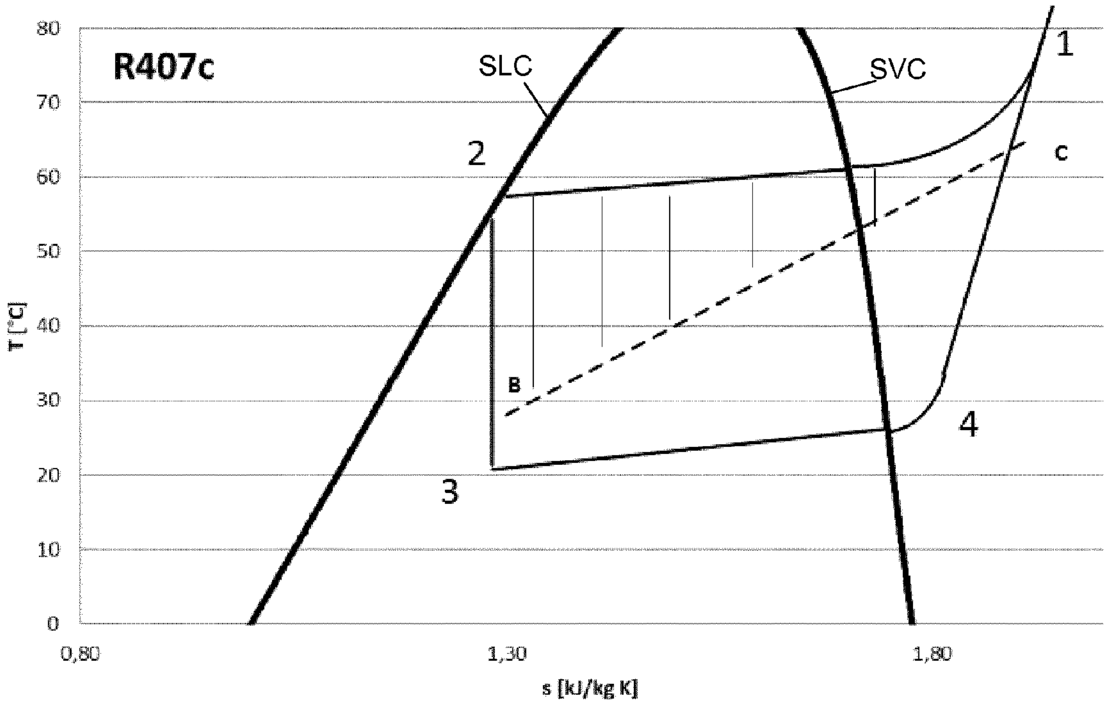

R407c
SLC
SVC
1
2
3
4
B
C
80
70
60
50
40
30
20
10
0
T [°C]
0,80
1,30
1,80
s [kJ/kg K]

FIG. 4B

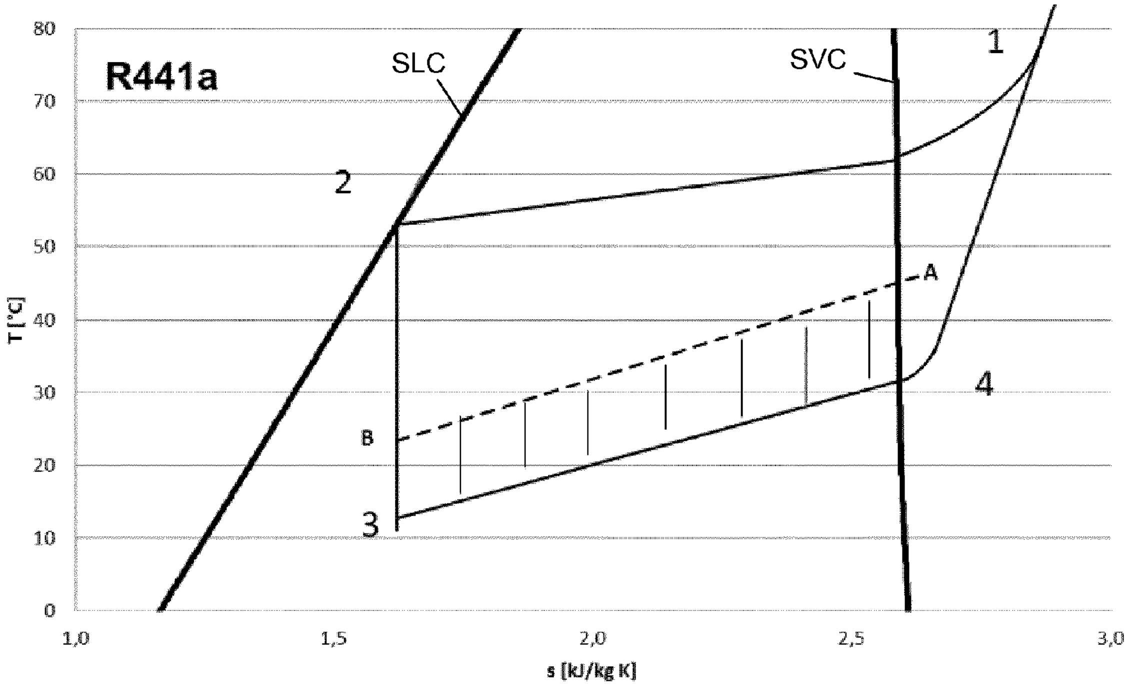

R441a
SLC
SVC
1
2
3
4
A
B
T [°C]
s [kJ/kg K]
0
10
20
30
40
50
60
70
80
1,0
1,5
2,0
2,5
3,0

FIG. 5A

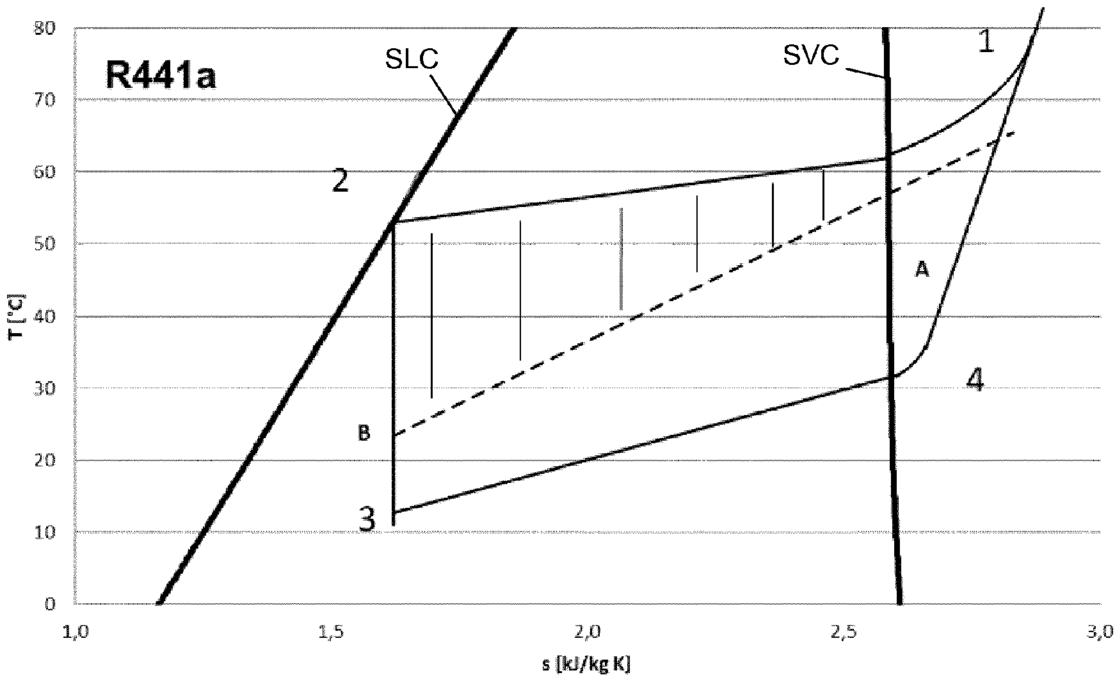

R441a
SLC
SVC
1
2
3
4
A
B
T [°C]
s [kJ/kg K]
0
10
20
30
40
50
60
70
80
1,0
1,5
2,0
2,5
3,0

FIG. 5B

FIG. 6

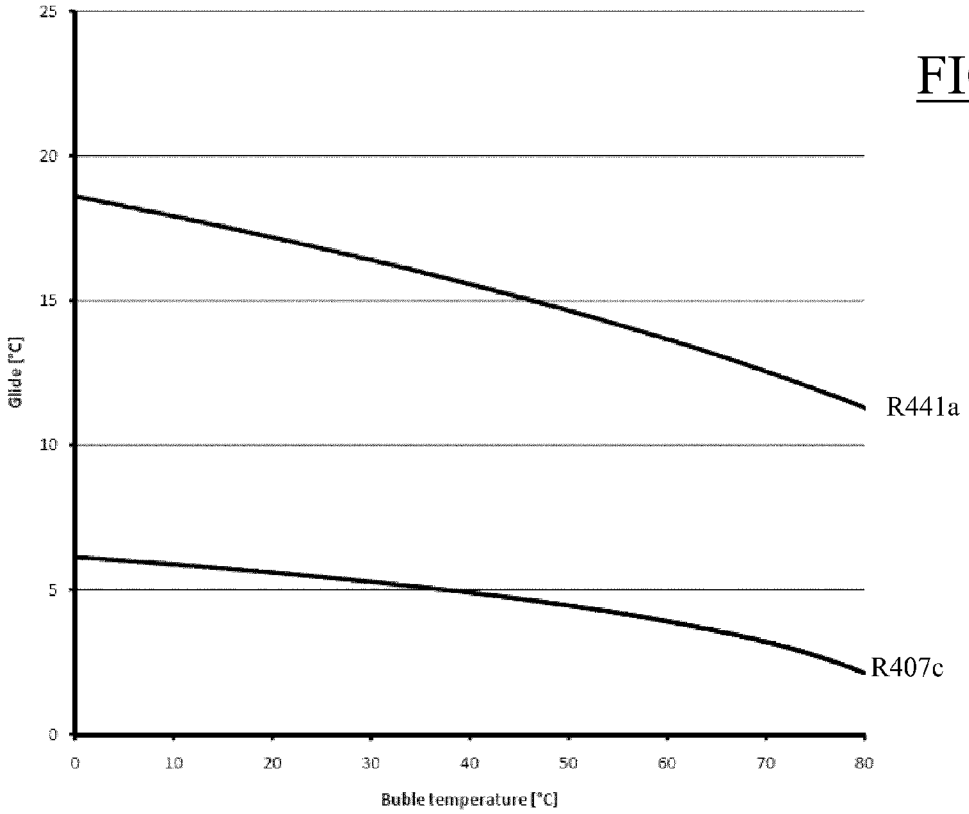
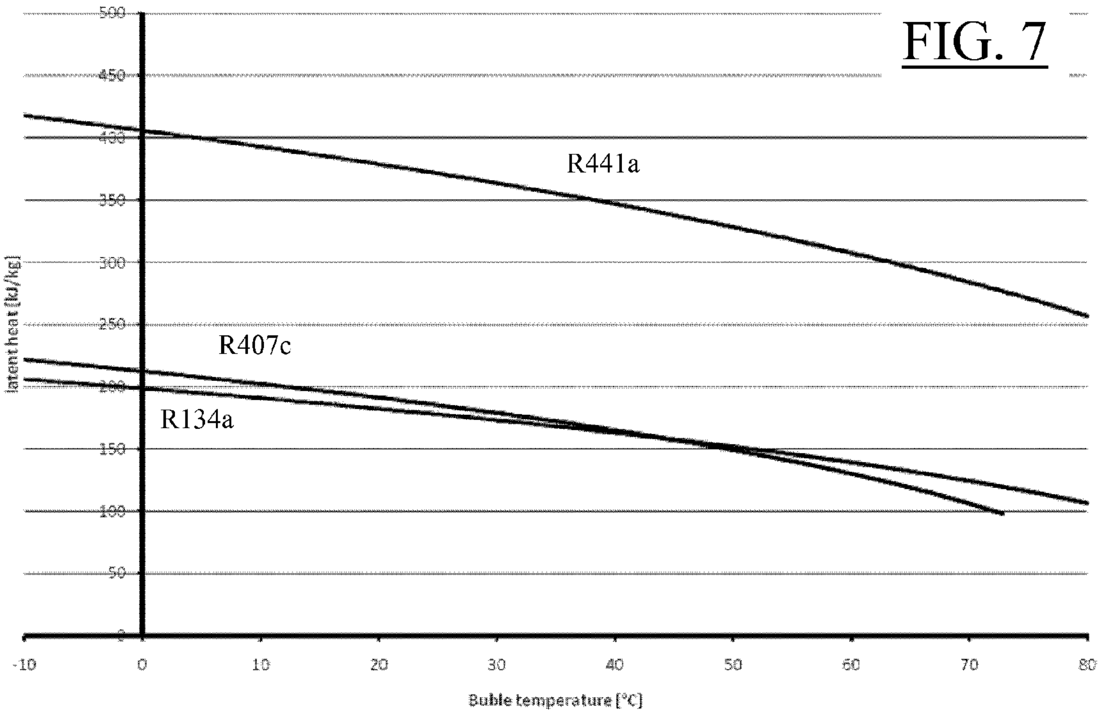

25
20
15
10
5
0
Glide [°C]
0
10
20
30
40
50
60
70
80
Buble temperature [°C]
R441a
R407c

FIG. 7


500
450
400
350
300
250
200
150
100
50
0
latent heat [kJ/kg]
-10
0
10
20
30
40
50
60
70
80
Buble temperature [°C]
R441a
R407c
R134a

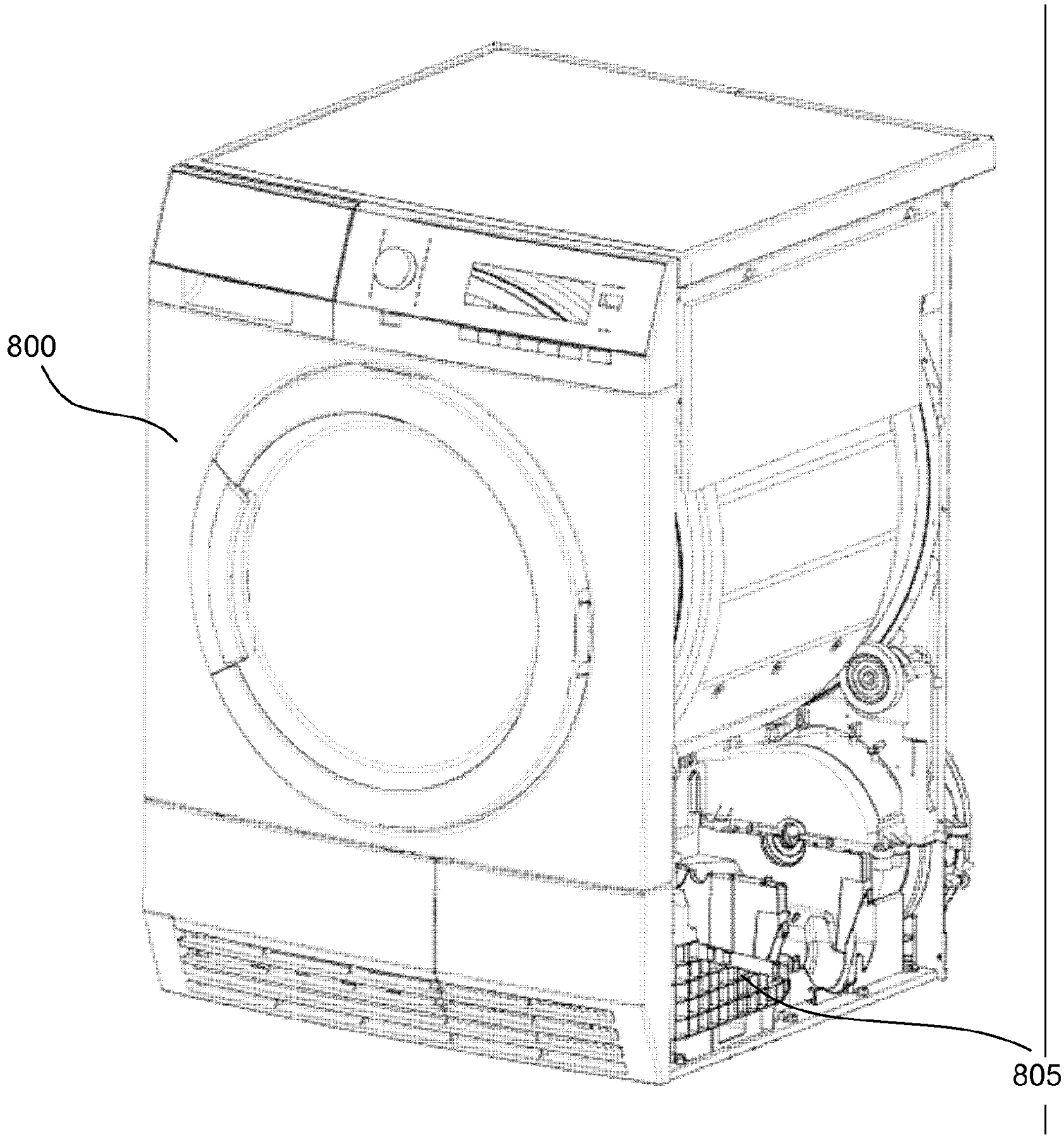


FIG. 8

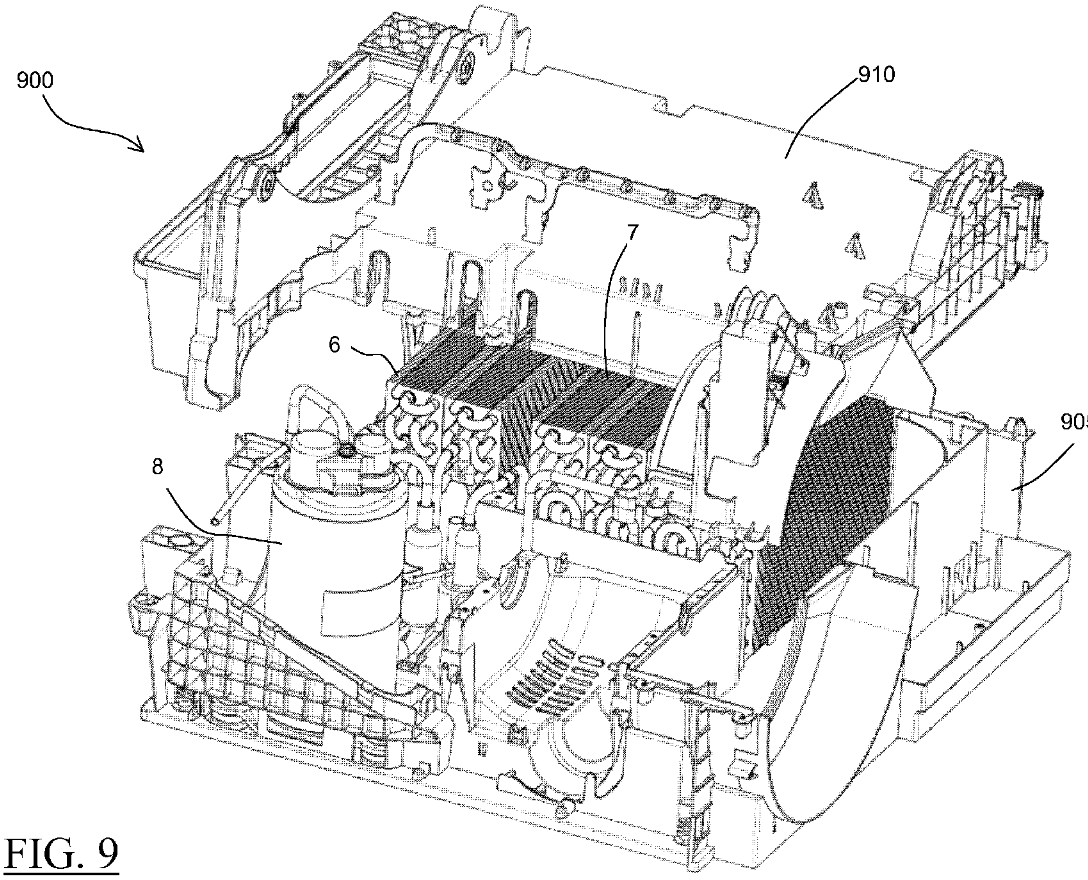


FIG. 9

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EUROPEAN SEARCH REPORT

Application Number
EP 12 17 8148

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
Y	EP 2 053 159 A1 (BSH ELECTRODOMESTICOS ESPANA [ES]) 29 April 2009 (2009-04-29) * paragraph [0008] * * paragraph [0011] * * paragraph [0018] * * paragraph [0027]; claim 1; figure 1 * -----	1-15	INV. D06F58/20 A47L15/42
Y	EP 1 964 965 A1 (BSH BOSCH SIEMENS HAUSGERAETE [DE]) 3 September 2008 (2008-09-03) * paragraph [0010]; claim 1 * -----	1-15	
Y	"Hydrocarbon refrigerant R441a gets first roll out in the UDA", ACR NEWS , 23 March 2011 (2011-03-23), XP002695016, Retrieved from the Internet: URL:http://www.acr-news.com/news/news.asp?id=2418&title=Hydrocarbon+refrigerant + [retrieved on 2013-04-09] * page 1 - page 2 * -----	1-15	TECHNICAL FIELDS SEARCHED (IPC) D06F A47L
A	ComStar: "ComStar to Offer Silekan Compressors Using New "Green" Energy-Saving Hydrocarbon Refrigerant R441a", , 21 March 2011 (2011-03-21), pages 1-2, XP002695422, Retrieved from the Internet: URL:http://www.digitalproducer.com/article/ComStar-to.Offer-Silekan-Compressors [retrieved on 2013-04-15] * page 1 - page 2 * -----	1	

The present search report has been drawn up for all claims

1

Place of search	Date of completion of the search	Examiner
Munich	15 April 2013	Fachin, Fabiano

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