A TWO-PHASE THERMOSIPHON DEFROSTING TECHNIQUE FOR AIR-SOURCE HEAT PUMPS

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Abstract: Air-source heat pumps loose performance during winter because of low ambient temperatures and frosting. The most common defrosting technique is to reverse the cycle. Unfortunately it provokes a break in the heat production and the defrosting energy is drawn from the heat stock previously constituted. This article presents the design of a heat pump prototype for simultaneous heating and cooling (named HPS). Its refrigeration circuit involves a piece of refrigeration circuit that could be modified and implemented to standard air-source heat pumps during retrofit or maintenance in order to carry out defrosting with better performance. It uses a water tank that recovers the subcooling energy of the refrigerant at first and is subsequently used as a cold source for evaporation. The second part of the sequence liberates the air evaporator for defrosting. Between two evaporators (air-to-refrigerant and water-to-refrigerant) at different temperatures, a thermosiphon forms. A supplementary amount of vapour flows out of the water evaporator and migrates towards the colder inside surface of the air evaporator tubes in thermal contact with the frost layer. The vapour, while condensing, brings the defrosting energy. The liquid returns back to the water evaporator by gravity. This defrosting system was observed by means of infrared thermography and tested experimentally on a heat pump prototype. It proved very efficient as defrosting time was short. Using this defrosting technique ensures:

- a continuous heat production with even better performance while defrosting thanks to the higher evaporating temperature,
- more frequent defrosting sequences because more easily activated, impacting on lower frost thickness and higher mean heat transfer coefficients.

A numerical study was carried out to assess the performance improvement achieved on a heating sequence. Simulations show a COP and an exergetic efficiency improvement respectively of 12 % and 18 %.

Key Words: defrosting, two-phase thermosiphon, air-source heat pumps

1 INTRODUCTION

Air-source heat pumps are energy-saving heating devices compared to electric convectors or radiators. They are easy to install and offer good comfort when coupled to low temperature heating systems. Moreover some heat pumps can be reversible and provide space cooling during the summer. The main drawback of this type of heat pump is frost formation at the air evaporator (Xia et al. 2006) (Huang et al. 2009) (Shao et al. 2009) during the winter. When the ambient air humidity is higher than the saturation humidity at the evaporator surface temperature, condensation forms on the fins of the air coil. Under low ambient temperatures, condensation turns to frost. Frost increases the thermal resistance and the pressure drop on the air side flow. Because classic defrosting techniques provoke a break in heat production, the system performance strongly decreases when heating demands are at their highest. The defrosting sequence described in this article is carried out without stopping the heat production. A thermosiphon defrosting technique is proposed through the design of a heat pump for simultaneous heating and cooling (HPS) (Byrne et al. 2009). The HPS can carry out space heating, space cooling and domestic hot water (DHW) production for hotels and small office or residential buildings. On the low pressure side of refrigeration plants using several evaporators, thermosiphons are usually cancelled out by non-return valves because they represent risks of refrigerant trapping in the circuit. The HPS takes advantage of this phenomenon to run an “automatic” defrosting. Thermosiphons are known to be efficient means of heat transfer (Lee et al. 2009) (Hakeem et al. 2008) (Dobson 1998). This article first presents the HPS concept and its defrosting technique. Then the thermosiphon is validated and observed using infrared thermography. Finally a sequence involving defrosting during winter was modelled using TRNSYS software (Solar Energy Laboratory 2000). A simulated comparison between the performance of the HPS and a standard reversible heat pump is presented.

2 THE CONCEPT OF THE HPS

2.1 General specifications

The HPS applications are space heating, space cooling and domestic hot water production for hotels or glass fronted buildings in which the proportion of simultaneous demands in heating and cooling is high. This situation occurs in mid-season (spring and autumn) for north-south oriented buildings in which rooms facing north need heating and rooms facing south need cooling. Another situation is encountered in summer when cooling and domestic hot water demands are simultaneous.

The HPS prototype (figure 1) produces hot and chilled water using plate heat exchangers. A balancing air coil works either as a condenser for heat rejection in a cooling mode or as an evaporator for heat suction in a heating mode. The air evaporator and the air condenser are never used at the same time. These functions have been assembled in the same three-fluid air coil (air, high pressure refrigerant and low pressure refrigerant) in order to decrease the finned surface area compared to separate air condenser and evaporator. When the tubes of the air evaporator are used the surface of the fins near the tubes of the air condenser are also used and vice versa. A subcooler is connected to the cold water loop to carry out a short-time heat storage during winter sequences. Depending on the mode of operation, the electric components (compressor, fan and electronic valves named Evr) are managed automatically by a programmable controller or manually by the operator. The thermostatic expansion valves are named TEV1 (connected to the water evaporator) and TEV2 (connected to the air evaporator). Non-return valves named Nrv1 and Nrv2 are placed at the outlets of the air and water condensers to avoid refrigerant trapping in the condensers.

As pressures and temperatures are linked during condensation, a high pressure control system ensures that condensation is completed in the condenser (and does not finish in the subcooler). Moreover it is able to control the condensation temperature and thus the heating capacity. A special liquid receiver is placed on the liquid line. It is connected to the compressor discharge line and the inlet of the air evaporator by copper tubes of smaller diameter on which electronic on-off valves are placed (EvrHP and EvrLP in figure 1). The high pressure control system indirectly controls the volume of liquid in the receiver. The volume of liquid in the different condensers depends on the mode. If the high pressure is below the set point, the electronic valve EvrHP is opened by the controller. The receiver is filled up with gas coming from the compressor discharge line at a pressure higher than the pressure in the receiver until the set pressure is reached. The gas entering the receiver drives the liquid towards the evaporator. The non-return valve closes because pressure becomes higher at the outlet than at the inlet. The subcooling heat exchanger and the bottom part of the water condenser are filled up with more liquid until the appropriate level of liquid is reached. If however the chosen mode is the cooling mode, the condenser becomes the air heat exchanger. The set point for pressure is then the lowest possible. The pressure is reduced by driving the vapour out of the top part of the receiver towards the inlet of the air evaporator. The refrigerant in a liquid phase is sucked out of the water condenser and the subcooler and enters the receiver. The operation of the control system depends upon a special liquid receiver being designed sufficiently high and narrow with the main objective to enhance temperature stratification and to limit as far as possible thermal transfer between the gas and the liquid. When the gas is injected, part of it condenses. When gas is rejected to the low pressure, part of the liquid evaporates. Although these phenomena can reduce the efficiency of the liquid variation in the receiver, it stabilizes the control system. The receiver is also thermally insulated to reduce the heat transfer towards the ambience.

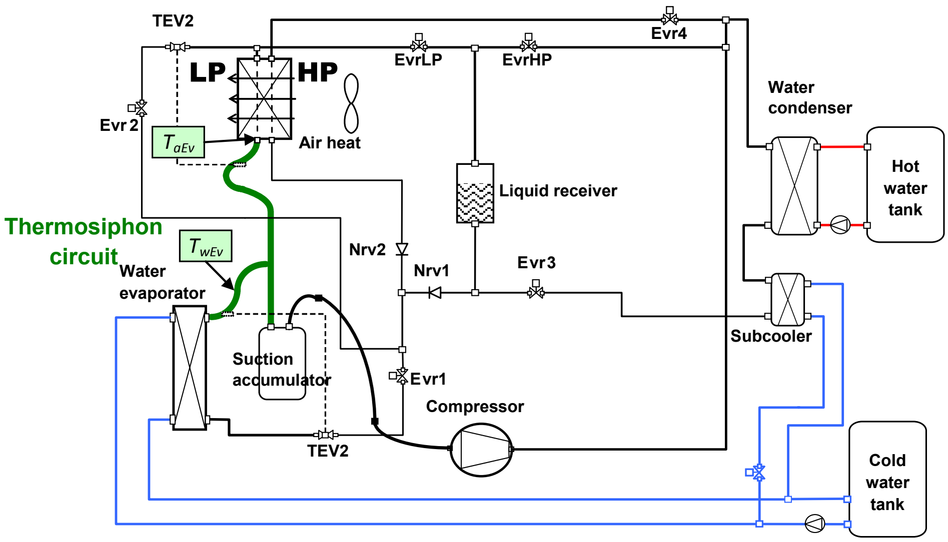


Figure 1: HPS Circuit

Table 1 shows the general specifications of the components. The chosen refrigerant is R407C, which is widely used in heat pumping technology. The refrigerant charge is increased compared to a conventional heat pump system. The calculated minimum refrigerant charge of the HPS is 9.8 kg and the effective charge is 16.3 kg for safety of usage during experiments. As a comparison, Poggi et al. (2008) gives statistical data concerning specific charge of air conditioning systems leading to a maximum of 1 kg per kW of cooling capacity for split systems. Meunier et al. (2005) gives an average ratio of 0.2 kg per kW for conventional heat pumps. Therefore the refrigerant charge of the HPS will be higher than conventional systems and its operation should be restricted with unsecure fluids.

Table 1: Specifications of HPS components

Component	Specification
Compressor Brand: Copeland Type: Scroll Ref: ZB38KCE-TFD	Swept volume: 14.5 m³/h Nominal cooling capacity (T_{ev} = 0 °C / T_{cd} = 40 °C): 11.5 kW
Water heat exchangers - condenser - evaporator - subcooler	Type: plate heat exchanger 50 plates, 2.45 m² 34 plates, 0.8 m² 14 plates, 0.16 m²
Air heat exchangers	Type: finned tubes, 68 m² 6 rows of 30 tubes (l = 750 mm , Ø = 10 mm)
Working fluid	R407C

2.2 Operating modes

Three operating modes can be run.

- The simultaneous mode produces hot and chilled water using the water condenser and the water evaporator (electronic valves Evr1 and Evr3 are open).

- The heating mode produces hot water using the water condenser, the air evaporator (electronic valves Evr2 and Evr3 are open) and also the subcooler to store the subcooling energy in the cold water tank.
- The cooling mode only produces cold water using the water evaporator and the air condenser (electronic valves Evr1 and Evr4 are open).

A production ratio r_p can be defined following equation 1 using heating and cooling capacities. For a simultaneous mode this ratio is close to 1.3.

$$r_p = \frac{\dot{Q}_h}{\dot{Q}_c} \tag{1}$$

A demand ratio r_d can be defined following equation 2 using instantaneous heating and cooling demands.

$$r_d = \frac{heating\ demand}{cooling\ demand} \tag{2}$$

Depending on the building demand ratio r_d, two sequences can be run:

- if $r_d > r_p$, heating demand is higher than cooling demand, the sequence starts in the simultaneous mode and continues in the heating mode when the cooling demand is satisfied;
- if $r_d < r_p$, cooling demand is higher than heating demand, the sequence starts in the simultaneous mode and continues in the cooling mode when the heating demand is satisfied.

The variability of operating times in each mode enables the control system to adapt hot and chilled water production to heating and cooling demands.

2.3 Classic winter sequence

During winter the sequence alternates between heating and simultaneous modes. The cold water tank is used as a short-time heat storage. The sequence begins by the heating mode engaging the water condenser, the air evaporator and the subcooler. The cold water tank is heated, usually from 5 to 15 °C, by the refrigerant subcooling energy. Then the HPS switches to the simultaneous mode and uses energy stored in the cold water tank as a cold source for the water evaporator. The cold water tank temperature decreases from 15 to 5 °C.

In the simultaneous mode, the evaporating temperature is higher than in the heating mode from the moment that ambient air is colder than the short-time heat storage tank. Therefore, using the simultaneous mode for a time during the winter sequence enables to produce hot water continuously with improved average system performance compared to standard air-source heat pumps. Besides, in the simultaneous mode, the air evaporator is not used for evaporation and can be defrosted using a two-phase thermosiphon.

2.4 Winter sequence with defrosting

In the heating mode, under cold outside air temperatures, the fins of the air evaporator get frosted. Before frost thickness becomes critical, the cold tank temperature rises to 15 °C and the simultaneous mode is engaged by the controller. In this mode, the air coil is automatically defrosted by a two-phase thermosiphon formed between the two evaporators. A supplementary flow of vapour comes out of the water evaporator and migrates towards the air evaporator where the temperature, and thus the pressure, is lower. The refrigerant exchanges latent heat with the frosted fins and condenses. It finally flows back to the water evaporator by gravity.

A major advantage of this sequence is to carry out defrosting without stopping the heat production. Frost thickness can thus be minimized and mean convection heat transfer coefficients at the evaporator can be maximized. The average heat pump efficiency under frosting conditions is improved compared to the performance of standard air-source heat pumps that use hot gas or reversed cycle defrosting methods (Rajapaksha et al. 2003).

3 EXPERIMENTAL VALIDATION

3.1 Experimental setup

A HPS prototype was built and connected to a water distribution system. The water system is used to limit the temperature variation of hot and cold water. It is composed of two water tanks, a circulation pump and four fan coil units (FCU) that dissipate the heating or cooling energy produced by the heat pump. The heating and cooling nominal power of the FCU is respectively 1.85 kW (50 °C / 40 °C) and 1.5 kW (7 °C / 12 °C). Temperature and humidity of the air circulating through the AHX are controlled, namely in order to obtain frosting conditions at the air coil. Type K thermocouples were placed on every inlet and outlet of the refrigeration cycle components on refrigerant, water and air sides. They were placed in contact with the copper tube and recovered by a lagging tape. The uncertainty on these measurements is ±0.5 K. High and low pressure sensors come from the P299 series of Johnson Controls. Their accuracy is ± 1 %. The sensors were connected to an acquisition unit using a time base of 1 second and to the control computer through which were also managed the different electric components. Electric energy consumption was given by a pulse electric meter. Water mass flow rates were calculated using water meters.

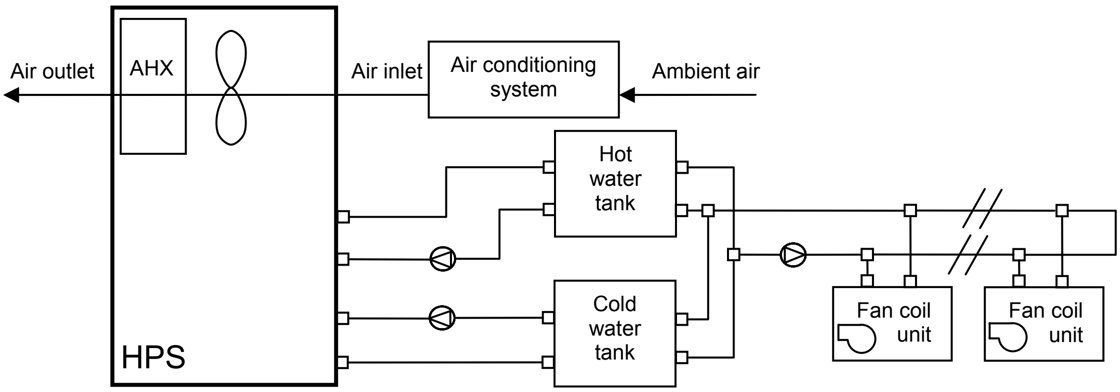


Figure 2: Scheme of the connections between the HPS and the air and water systems

3.2 Two-phase thermosiphon observation

The two-phase thermosiphon circulates in tubes that have large diameters (represented in green in figures 1 and 3). Vapour migrates from the water evaporator towards the air evaporators. Condensed refrigerant returns back by gravity to the water evaporator and the suction accumulator.

Thermographic pictures (figure 4) have been taken before and during the defrosting phase at the position of the white square in figure 3. Before the defrosting sequence, temperature is homogenous. The tube contains vapour flowing from the air evaporator towards the compressor. When the two-phase thermosiphon defrosting sequence is launched, the thermostatic expansion valve TEV2 does not supply refrigerant to the air evaporator anymore because the electronic valve Evr2 is closed. In figure 4 it can be seen that heterogeneity of temperature appears in the tube. The hotter vapour phase flows up the tube in the top part of the section and the colder liquid phase flows down in the bottom part.

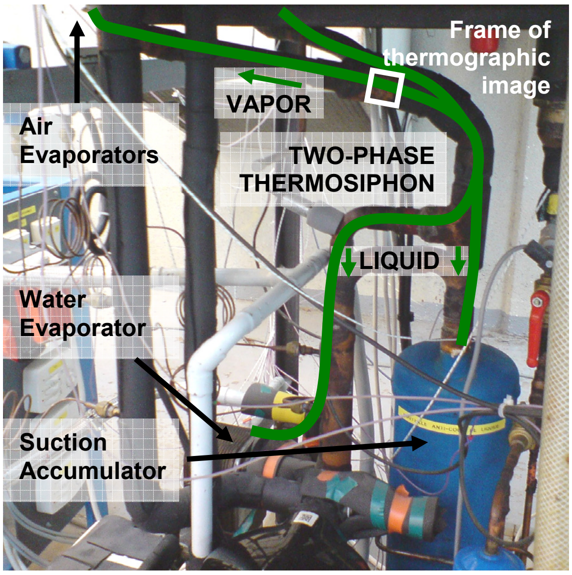
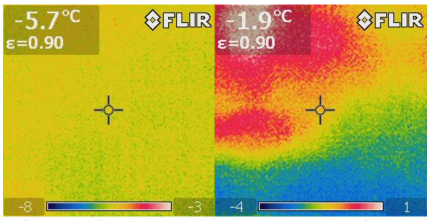


Figure 3: Photograph of the thermosiphon setup



Before defrosting During defrosting

Figure 4: Thermographic pictures of the tube between the two evaporators

3.3 Defrosting technique validation

The validation is based on the observation of the frost layer disappearance during a defrosting sequence using the two-phase thermosiphon. The defrosting time is compared to a frost layer disappearance time by thermal exchange with the ambient air surrounding the test bench. In this second case compressor and fans are stopped.

3.3.1 Frosting phase

The first phase consists in running the heat pump in a heating mode under frosting conditions. Ambient air temperature is controlled around 0 °C. Frost progressively appears on the LP section of the air coil of the prototype. The frost layer thickness increases over a period of 30 minutes. Meanwhile the cold water tank temperature increases from 5 to 15 °C thanks to the amount of energy recovered by subcooling the refrigerant.

3.3.2 Defrosting phase

After 30 minutes, the frost layer is assumed to be sufficient to run the defrosting sequence. The electronic valve Evr2 is switched off and the electronic valve Evr1 is switched on. The water evaporator is used instead of the air evaporator. The simultaneous mode is engaged and the two-phase thermosiphon defrosting starts. The frost layer disappears within 2 minutes (figure 5). In order to compare, after a 30-minute frosting phase, the prototype has been stopped. The defrosting energy was then brought by thermal convective exchange with the ambient air surrounding the test bench. The time of frost disappearance reached 20 minutes. This comparison confirms that the thermosiphon defrosting technique works very efficiently.

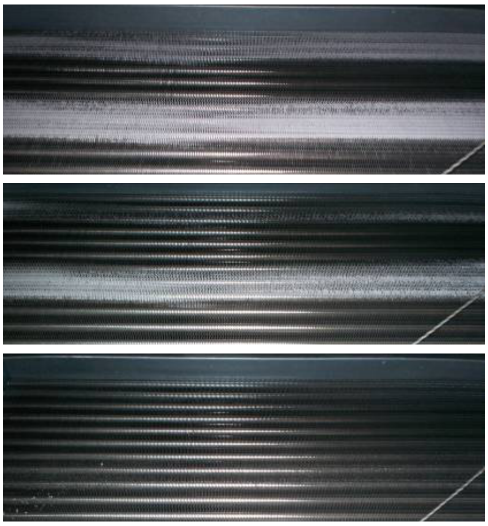


Figure 5: Photographs of the air coil during the thermosiphon defrosting phase

4 SIMULATION STUDY

A simulation study is carried out to evaluate the performance improvement brought by the use of such a defrosting technique. An energetic and exergetic comparison is made between the HPS and a standard reversible heat pump working with R407C and coupled to a hotel of 45 bedrooms located in Rennes (France).

4.1 Modelling assumptions

The simulations are run using TRNSYS software. The time step is of 5 minutes. The reversible heat pump is supposed not to be able to produce DHW because during summer, it would interfere too much with an operation in the cooling mode. The reversible heat pump model uses an electric water heater for DHW production. In the HPS model, the heating capacity devoted to DHW production is deducted from the total heating capacity of the HPS. An electric backup is also implemented to the model in case the HPS cannot satisfy the DHW production on its own. The chosen hypothesis is that the use of a desuperheater (not in place on the prototype but modelled for the simulation study) does not induce any change in the set point for high pressure compared to what it would be without DHW production. Defrosting of the HPS is supposed « automatically » carried out during a simultaneous mode by the two-phase thermosiphon. For the reversible HP defrosting is achieved by an inversion of the refrigeration cycle. Reversing the cycle provokes a drawing of heat in the hot water tank. A defrosting model was implemented to the heat pump modelling.
An electric backup for space heating was also implemented to the HPS and reversible heat pump models.
Table 2 presents the annual needs in space heating, space cooling and DHW production of the hotel of 45 bedrooms located in Rennes.

Table 2: Annual needs of a 45-bedroom hotel in Rennes

Sectors	*Annual needs*
DHW production	74870 kWh/year
Space heating	51680 kWh/year
Space cooling	39184 kWh/year

4.2 Frosting and defrosting model for the reversible heat pump

A simplified frosting model was programmed in order to launch defrosting sequences in the reversible heat pump model. The defrosting model calculates the frost mass on the air coil at time step *i*+1 by equation 3 in function of the specific humidity at the inlet and the outlet of the coil, air mass flow rate and the duration of the time step *dt*. The bypass factor f_{bp} of the air coil is taken constant at a value of 0.7.

$$m_f^{\,i+1} = \left(w_{in} - w_{out}\right) \cdot \left(1 - f_{bp}\right) \cdot \dot{m}_{air} \cdot dt + m_f^{\,i} \tag{3}$$

When the heat pump has worked during 45 minutes with an ambient temperature beneath 7 °C, the cycle is reversed. Evaporation is carried out at the water heat exchanger and condensation at the air heat exchanger. Heat is pumped out from the hot water tank to carry out defrosting. Defrosting sequences take into account the performance loss due to the break in the heat production and to the heat used at the evaporation of the reversed cycle. Therefore in the simulation, there is a break in the heat production during a complete time step but the defrosting time is shorter (equation 4). During the rest of the time step, the heat pump is stopped. It is calculated by the ratio of the defrosting energy (equation 5) over the available heating capacity in the reversed cycle. The energy drawn from the hot water tank corresponds to the defrosting energy Q_{df}. The mean electric power over the time step during which defrosting occurs is calculated using equation 6.

$$t_{df} = \frac{Q_{df}}{\dot{Q}_c + \dot{W}} \tag{4}$$

$$Q_{df} = m_f \cdot L_f \tag{5}$$

$$\dot{W}_{ave} = \frac{t_{df}}{dt} \cdot \dot{W} \tag{6}$$

4.3 Equations for energy and exergy calculations

In the winter, the heating mode or the simultaneous mode is used. The instantaneous COP is calculated over a time step by dividing the heating capacity by the electrical power. When only space cooling is needed, the instantaneous COP is given by the division of the cooling capacity by the electric power. When heating and cooling are needed, the simultaneous mode is used. The instantaneous COP is given by the sum of heating and cooling COPs (equation 7). The average coefficient of performance over a sequence is calculated in proportion to the sum of heating and cooling needs on each time step (equation 8).

$$COP_{inst} = \frac{\dot{Q}_h + \dot{Q}_c}{\dot{W}} \tag{7}$$

$$COP_{ave} = \frac{\sum COP_{inst} \cdot (q_h + q_c)}{\sum (q_h + q_c)} \tag{8}$$

Summing heating and cooling energies appears somewhat tricky since these energies are from different nature. So the scientific community prefers to work with exergy which corresponds to the amount of energy ideally convertible into mechanical work (O'Callaghan et al. 1981). Exergy evaluates the quality of the produced energy (equation 9). For instance, Sarkar et al. (2006) use exergy as a performance factor of a heating and cooling heat pump working with CO_2. Exergetic efficiency is defined as the ratio of the produced exergy over the consumed exergy at compression. As electric and mechanical energy are considered to be pure forms of exergy, the consumed exergy at the compressor corresponds directly to the mechanical work of compression and the exergy consumption of the electric backup corresponds to the electric energy consumption. Equation 10 gives the exergetic efficiency of both machines for each time step. For the reversible heat pump, exergy used for DHW production is zero. The source temperature T_{so} is calculated by the logarithmic mean temperature equation (equation 11). The reference temperature T_0 is the ambient temperature.

$$Ex = Q \cdot \left| 1 - \frac{T_0}{\overline{T}_{so}} \right| \tag{9}$$

$$\eta_{ex} = \frac{Ex_h + Ex_{DHW} + Ex_c}{W} \tag{10}$$

$$\overline{T}_{so} = \frac{T_{in} - T_{out}}{\ln \frac{T_{in}}{T_{out}}} \tag{11}$$

4.4 Simulation results

This section of the article presents the simulation results of coefficients of performance and exergetic efficiencies of the HPS and the reversible heat pump, during a winter heating sequence. The winter sequence chosen corresponds to the evening of January 13th (between 16:00 and 22:30) of the TRNSYS weather data file of Rennes. During this sequence, the ambient temperature decreases from 2.0 °C to -0.9 °C.

4.4.1 Evolution of coefficients of performance during a heating sequence

Figure 6 presents the evolution of the coefficient of performance during the chosen heating sequence.

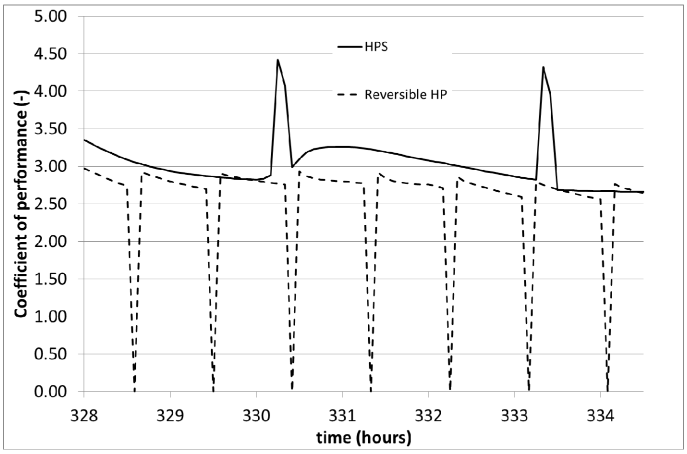


Figure 6: Evolution of coefficients of performance during a heating sequence

A time step during which the COP is zero corresponds to a defrosting sequence of the reversible heat pump. Coefficients of performance are globally better for the HPS and much better during the use of the simultaneous mode. The simultaneous mode COP is higher than the heating mode COP because the source temperature is higher and the discrepancy between the source and the evaporating temperatures is lower for water than for air heat exchangers. Therefore the evaporating temperature in the simultaneous mode is increased even more.

The HPS works in the simultaneous mode only two times during the sequence that lasts 6 h 30 min. The frosting level of the air heat exchanger may be important and could affect the performance of the HPS. This problem is not taken into account in the modelling since it can be easily solved by a correction in the sizing of the cold water tank.

4.4.2 Evolution of exergetic efficiencies during a heating sequence

Figure 7 shows the evolution of the exergetic efficiencies during the chosen heating sequence.

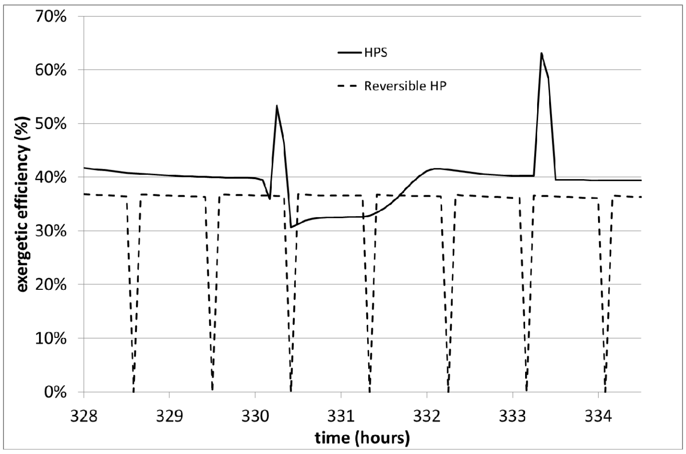


Figure 7: Evolution of exergetic efficiencies during a heating sequence

The exergetic efficiencies almost follow the evolutions of the COPs, except between the hours 330 and 332 which correspond to the interval 18:00 to 20:00 of January 13th. During this 2-hour period, a demand in DHW appears. This extra demand provokes a performance loss in terms of exergetic efficiency. The reversible heat pump is not affected by the DHW demand since DHW is produced by an electric water heater.

4.4.3 Mean performance factors during a heating sequence

Table 3 shows the mean performance factors during the heating sequence presented on figures 6 and 7. The COP and the exergetic efficiency are respectively 12 % and 18 % higher for the HPS compared to a reversible HP.

Table 3: Comparison of performance factors during a heating sequence

Type of heat pump	*HPS*	*Reversible HP*
COP (-)	3.03	2.71
η_{ex} (%)	39.3 %	33.3 %

When taking into account the energy and exergy consumptions of the electric backups for space heating and DHW production, the COPs decrease and the exergetic efficiencies increase (table 4). The exergetic efficiency is higher because electric energy consumed by the electric backups is considered as pure exergy. The COP of the reversible heat pump dramatically decreases because of the complete DHW production is carried out by the electric water heater. The exergetic efficiency of the reversible heat pump increases more because there is no conversion factor (Carnot factor) for DHW production, as there is for thermal energy produced by the HPS.

Table 4: Comparison of performance factors taking into account the backups for heating and DHW production during a heating sequence

Type of heat pump	*HPS*	*Reversible HP*
COP (-)	2.57	1.82
η_{ex} (%)	44.5 %	55.6 %

5 CONCLUSIONS AND PERSPECTIVES

This article presents the design of an air-source heat pump for simultaneous heating and cooling (HPS) that offers improved performance compared to a standard reversible heat pump. Apart from producing simultaneously hot and cold water, the system switches from the air evaporator to a water evaporator during winter sequences. When standard reversible heat pumps are penalized by defrosting (break in the heat production, decrease in COP), the HPS carries out defrosting with enhanced performance thanks to a water source at the evaporation at a higher temperature than ambient air.

In refrigeration circuits, thermosiphon phenomena are usually cancelled out by the use of non-return valves because of possible trapping of refrigerant in some parts of the circuit. On the contrary, the HPS takes advantage of the thermosiphon effect, generated between the two evaporators, to carry out a defrosting sequence. The two-phase thermosiphon was observed by thermographic pictures of a section of the tube between the two evaporators. The defrosting effect was validated by the quick disappearance of the frost layer on the air coil during a defrosting sequence (2 minutes) compared to the defrosting time when the heat pump was stopped (20 minutes). Other experiments must be carried out in real operating configurations to assess the effect of the two-phase thermosiphon defrosting technique under more realistic winter conditions of temperature and humidity.

A simulation study was carried out to evaluate the performance improvement of the defrosting technique during a heating sequence. Results show that compared to a reversible heat pump the HPS works with higher COP (12 %) and higher exergetic efficiency (18 %) without taking the electric backups into account.

The two-phase thermosiphon is a non-penalizing defrosting technique that can be adapted to standard air-source heat pumps with some modifications on the refrigerant circuit. A short-time heat storage has to be connected and used as a heat source first for subcooling and subsequently for evaporation. This defrosting system was tested on a special prototype and further research will be carried out on the simplification of this concept and its adaptation on conventional heat pumps.

4 REFERENCES

Xia Y., Zhong Y., Hrnjak P.S., Jacobi A.M. 2006. “Frost, defrost, and refrost and its impact on the air-side thermal-hydraulic performance of louvered-fin, flat-tube heat exchangers”, International Journal of Refrigeration, Vol. 29, pp. 1066-1079.

Shao L.-L., Yang L., Zhang C.-L. 2010. “Comparison of heat pump performance using fin-and-tube and microchannel heat exchangers under frost conditions”, Applied Energy, Vol. 87, pp. 1187-1197.

Byrne P., Miriel J., Lenat Y. 2009. “Design and simulation of a heat pump for simultaneous heating and cooling using HFC or CO_2 as a working fluid”, International Journal of Refrigeration, Vol. 32, pp. 1711-1723.

Lee J., Ko J., Kim Y., Jeong S., Sung T, Han Y., Lee J.-P, Jung S. “Experimental study on the double-evaporator thermosiphon for cooling HTS (high temperature superconductor) system”, Cryogenics, 2009, Vol. 49, pp. 390–397.

Hakeem M.A., Kamil M., Arman I. 2008. “Prediction of temperature profiles using artificial neural networks in a vertical thermosiphon reboiler”, Applied Thermal Engineering, Vol. 28, pp. 1572-1579.

Dobson R.T. 1998. “Simulation of the two-phase flow in a thermosyphon using an inclined transparent tube with the lower-end closed and the upper-end open”, Revue Générale de Thermique, Vol. 37, pp. 968-972.

Rajapaksha L., Suen K.O. 2003. “Influence of reversing methods on the performance of a reverible water-to-water heat pump”, Applied Thermal Engineering, Vol. 23, pp. 49-64.

Huang D., Li Q., Yuan X. 2009. “Comparison between hot-gas bypass defrosting and reverse-cycle defrosting methods on an air-to-water heat pump”, Applied Energy, Vol. 86, pp. 1697-1703.

Solar Energy Laboratory, University of Wisconsin-Madison. 2000. “TRNSYS, A Transient Simulation Program”, Vol. I, Reference Manual.

Poggi F., Macchi-Tejeda H., Leducq D., Bontemps A. 2008. “Refrigerant Charge in Refrigerating Systems and Strategies of Charge Reduction”, International Journal of Refrigeration, Vol. 31, pp. 353-370.

Meunier F., Rivet P., Terrier M.-F. 2005. Froid Industriel, Editions Dunod.

O'Callaghan P. W., Probert S. D. 1981. “Exergy and economics”, Applied Energy, Vol. 8, pp. 227-243.

Sarkar J., Bhattacharyya S., Ram Gopal M. 2006. “Simulation of a transcritical CO_2 heat pump cycle for simultaneous cooling and heating applications”, International Journal of Refrigeration, Vol. 29, pp. 735-743

NOMENCLATURE

AHX	air heat exchanger
COP	coefficient of performance
DHW	domestic hot water
dt	time step duration (s)
Evr	Electronic valve of regulation
f	factor (-)
FCU	fan coil unit
HP	heat pump
HPS	heat pump for simultaneous heating and cooling
L	latent heat of fusion (J kg^{-1})
LP, HP	low pressure, high pressure (Pa)
m	mass (kg)
$\dot{m}$	mass flow rate (kg s^{-1})
Nrv	non-return valve
q	demand (J)
Q	thermal energy (J)
$\dot{Q}$	heating or cooling capacity (W)
r	ratio (-)
t	time (s)
T	temperature (°C)
TEV	thermostatic expansion valve
$\dot{W}$	electric power (W)

Greek symbol:

η	efficiency (%)

Subscripts:

aEv	air evaporator
ave	average
bp	bypass
c	cooling
d	demand
df	defrosting
ex	exergetic
f	frost
h	heating
inst	instantaneous
p	production
r	refrigerant
so	source
wEv	water evaporator