PERFORMANCE ANALYSIS OF A CASCADE HIGH-TEMPERATURE HEAT PUMP USED IN A SINGLE-FAMILY DWELLING IN BELGIUM

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Abstract: This paper is devoted to the performance analysis of a very high-temperature air-source heat pump which uses a newly-developed cascade cycle. The heat pump has been installed in a non-insulated house in Belgium in August 2013 and is used for space heating in combination with high-temperature radiators, or for domestic hot water production. The heat pump has been equipped for detailed performance monitoring. Firstly, the results of the monitoring campaign are presented. They show a complex behavior of the heat pump but are very promising. Secondly, the behavior of the heat pump used for space heating is modeled to determine the best mode to use depending on the source and sink temperatures. The results show good agreement with the manufacturer's data.

Key Words: cascade cycle, high-temperature heat pump, model, monitoring

1 INTRODUCTION

Heat pumps are nowadays recognized as efficient renewable energy systems for space heating and/or domestic hot water production from the point of view of primary energy consumption and CO_2 emissions. A 2009 European Directive states that heat pumps can be considered as renewable energy systems if they save 13% primary energy (EP 2009). A recent European Decision (EP 2013) fixes the standard power plant efficiency at 0.455, leading to a minimum Seasonal Performance Factor (SPF) of 2.5. However, a minimum value of 2.88 is still used in Belgium, which is considered as the reference value in this paper. A SPF higher than 2.88 is easy to obtain with low temperature heating systems like heating floors (Dumont et al. 2005a and 2005b, Dumont et al. 2008a and 2008b, Duprez et al. 2008). For air source heat pumps, the SPF can reach 3.2 (Dumont et al. 2008b) and for ground source heat pumps, it is usually higher than 4.0. Unfortunately, these low-temperature heating systems are mostly installed in new houses, which are a very small part of the total building stock in Belgium (20% of the buildings are less than 20 years old (BFG 2005)). The introduction of high-SPF heat pumps in old or renovated buildings, usually using high-temperature heating systems, is of high concern. In these low-insulated dwellings, a mid- or high-temperature heating system is necessary to balance the high thermal losses of the envelope. Several types of air-source high-temperature heat pumps exist on the market, mostly with injection cycle and variable capacity compressors. Combined with mid-temperature convector heaters (water temperature regime of 40-45°C), they exhibit a SPF of 2.8 to 3.0 (Dumont et al. 2011). Cascade cycle heat pumps are thermodynamically more efficient systems due to the use of two Rankine cycles in cascade. This kind of machines should exhibit interesting SPF if combined with high-temperature radiators. The purpose of this study is to monitor, analyze and model the performance of such a cascade cycle heat pump installed in a non-insulated house in Belgium.

In this paper, we first present the system monitored and the first results recorded during Autumn 2013. We then present a model of a cascade cycle heat pump which is used to compute the best operating conditions of the machine.

2 SYSTEM OPERATION AND MONITORING

2.1 System description and operation

The house is located near the city of Tournai (Wallonia, South of Belgium) and is a four-side, single-family house, built in the 80's. The heat distribution system is composed of 12 radiators. The heat pump (from the French manufacturer AJ-Tech) is a high-temperature cascade air-to-water heat pump which is used for space heating and Domestic Hot Water (DHW) production. It has been installed in August 2013 and uses a cascade cycle with refrigerants R410A in the low pressure cycle and R134a in the high pressure cycle. The system is sketched in Figure 1. The backup heater is a gas boiler which previously heated up the house before the installation of the heat pump.

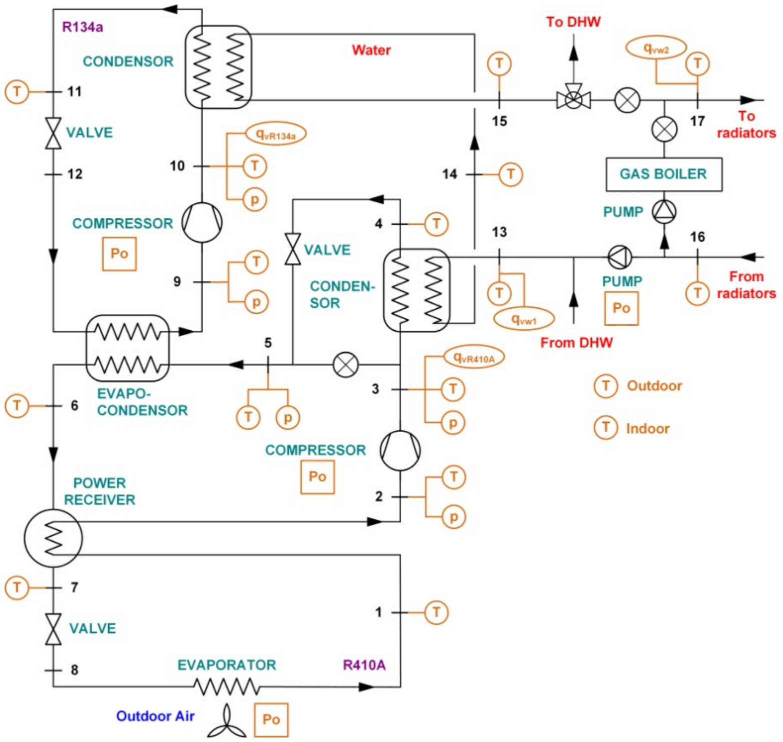

Figure 1: Heat pump configuration and sensors location for monitoring

The R410A cycle is composed of a variable-capacity rotary compressor, a plate condenser, a plate evapo-condenser, an internal heat exchanger (power receiver) and a finned tubes evaporator. The internal heat exchanger is used to prevent liquid R410A entering the compressor during startup. The R134a cycle is composed of a fixed-capacity scroll compressor, a plate condenser, and an evapo-condenser common with the R410A cycle.
The R410A vapor at low pressure is subsequently superheated (1-2), compressed (2-3), condensed (3-6), condensed again (6-7), expanded (7-8) and finally evaporated (8-1). The R134a vapor at low pressure is subsequently compressed (9-10), condensed (10-11), expanded (11-12) and finally evaporated and superheated (12-9).
Three different operation modes are used: mode#1 uses the R410A cycle only (1-2-3-4-5-6-7-8); mode#2 uses both cycles (R410A: 1-2-3-4-5-6-7-8; R134a: 9-10-11-12) and heat is released at the R410A and R134a condensers; mode#3 is the standard cascade cycle (R410A: 1-2-3-5-6-7-8; R134a: 9-10-11-12) where heat is released only at the R134a

condenser. The use and the control of the 3 modes are patented. These modes are presented in Figure 2.

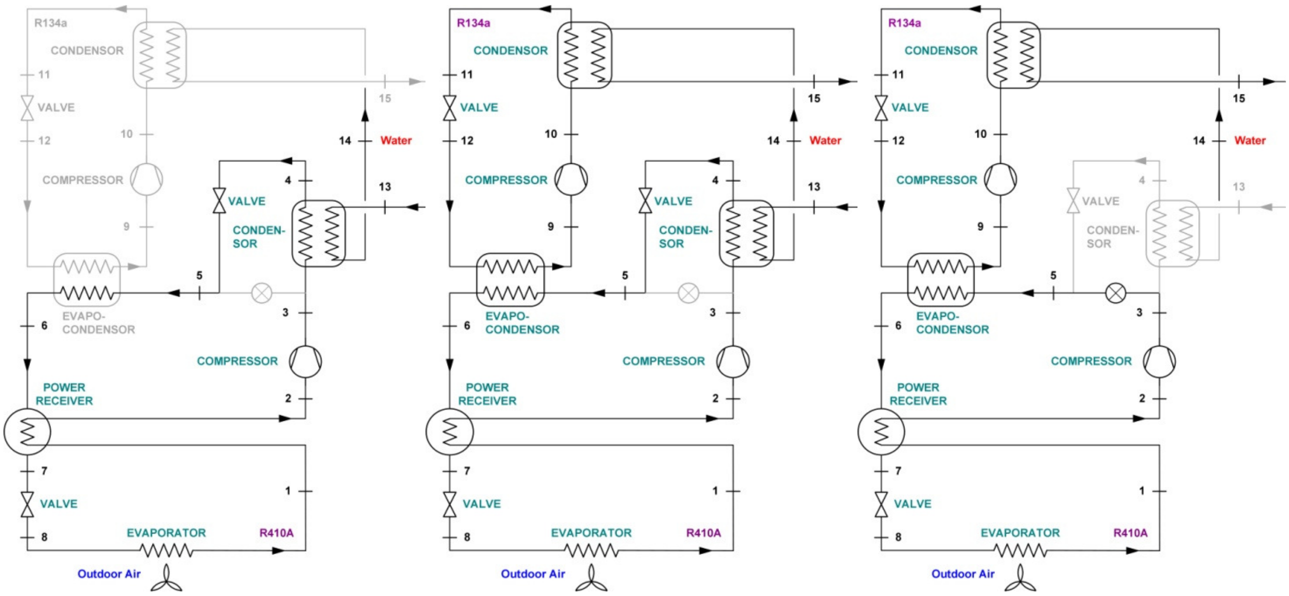

Figure 2: Operation modes: mode#1 (left), mode#2 (center) and mode#3 (right)

2.2 System monitoring

In order to achieve performance monitoring, the system has been equipped with sensors which measure temperature (T), pressure (p), volume flow rates (q_{VR410A}, q_{VR134a}, q_{VW1}, q_{VW2}) and electric power of the outdoor unit (evaporator fan and R410A compressor), of the R134a compressor and of the circulation pump (Po). Indoor and outdoor temperatures are also monitored (Figure 1). Each sensor delivers a signal every second, which is averaged every minute and stored in a data logger, then uploaded every week on a remote computer. Table 1 presents the different sensors.

Table 1: Sensors used for monitoring

Device	Range	Measurement point
Capacity pressure transducer	0 - 40 bar	2, 3, 5, 9,10
RTD (Class A) temperature sensor	-100 - 600°C	16, 17
RTD (Class A) temperature sensor	-50 - +50°C	Indoor, outdoor
Thermocouple T temperature sensor	-270 - 400°C	1-7, 9-11, 13-15
Vortex volume flow meter (R410A)	0 - 1.38 dm^3/s	3
Vortex volume flow meter (R134a)	0 - 2.0 dm^3/s	10
Vortex volume flow meter (water)	0 - 85 l/min	13, 17
Power analyzer	0 - 10 kW	Outdoor unit, R134a compressor
Power analyzer	0 - 1000 W	Evaporator fan, water pump

3 MONITORING RESULTS

3.1 Space heating

3.1.1. Typical mid-season day

The results presented in Figure 3 were recorded on October 16, 2013 and are typical of a mild weather (autumn and spring). The outdoor temperature is about 10°C. There are 3 space heating cycles (and 2 cycles for DHW production).

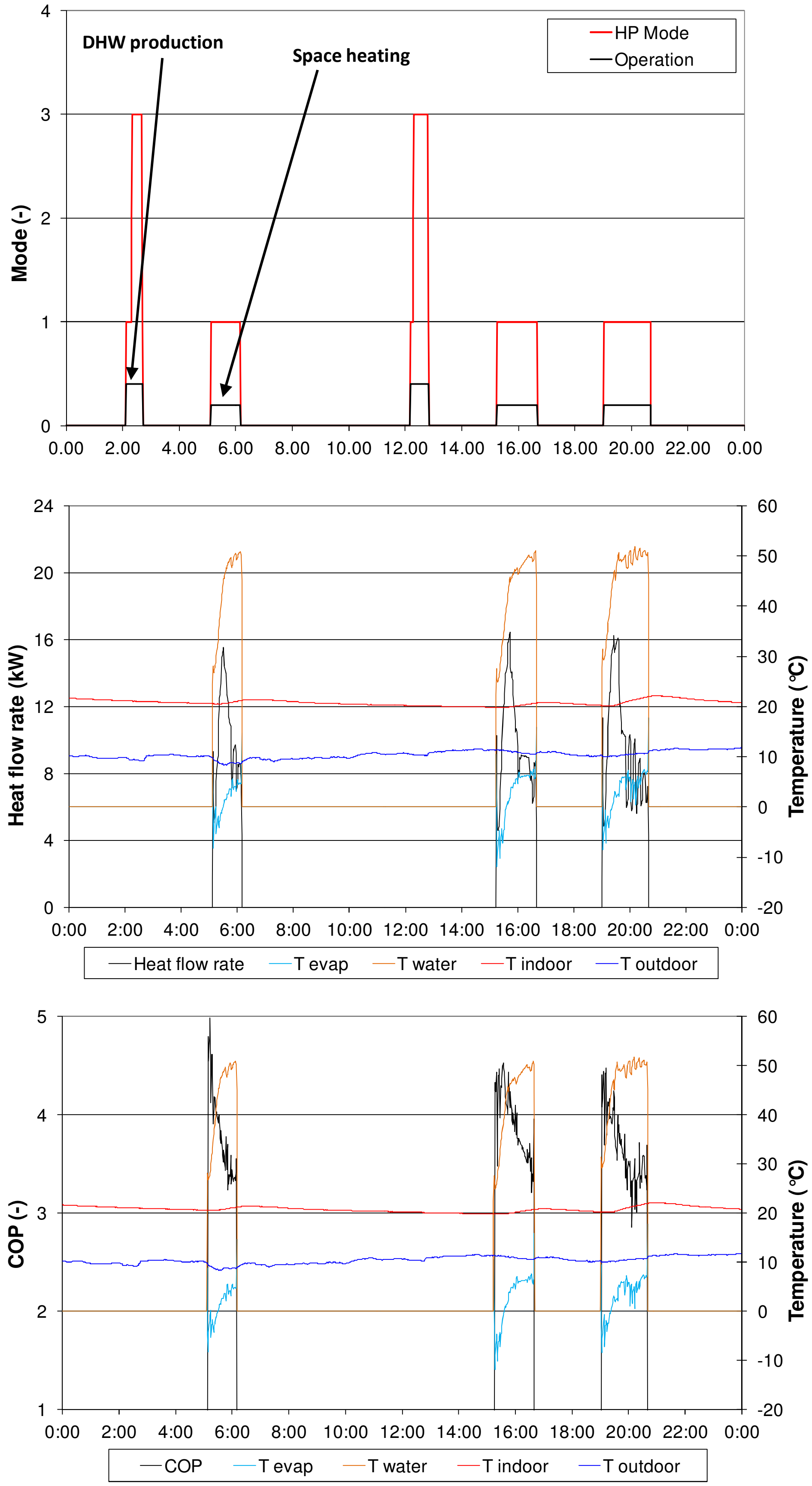

Figure 3: Operation mode (top), Heat flow (middle) and COP (bottom) for October 16, 2013

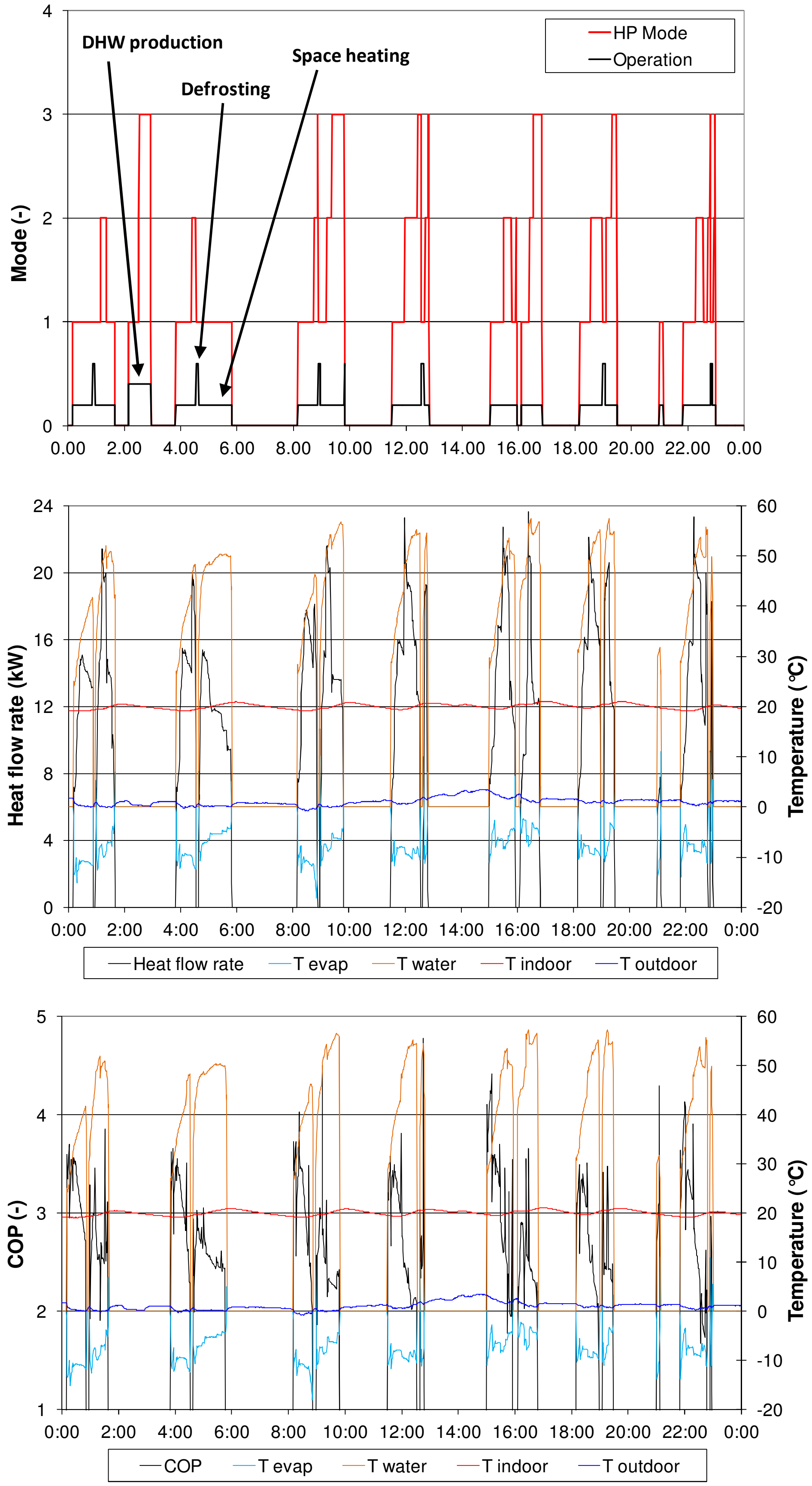

Figure 4: Operation mode (top), Heat flow (middle) and COP (bottom) for December 12, 2013

In contrast to most air-to-water heat pumps, each cycle does not begin with a heating peak, because of the filtering effect of the high thermal inertia radiators. The compressor speed is progressively increased to reach the desired water temperature regime (here 50°C) then, the speed is decreased to maintain the water temperature. The evaporation temperature increases from -5°C to 5°C due to the decrease of the flow rate. The indoor temperature steadies between 20 and 22°C. The delay between the heat flow rate and indoor temperature evolutions witnesses the high inertia of the radiators (about 1 hour). During one cycle, the heat flow rate decreases from 16 kW to 8 kW while the COP decreases from 4.5 to 3.5. The use of a capacity-controlled rotary compressor allows to obtain long cycles even in mild weather conditions and good COP. The HP mode is always mode#1. There is no need for defrosting.

3.1.2. Typical winter day

The results presented in Figure 4 were recorded on December 12, 2013 and are typical of a cold weather (winter). The outdoor temperature is about 1°C. There are 3 space heating cycles (and 1 for DHW production).
As mentioned above, the compressor speed is progressively increased to reach the desired water temperature regime which is here around 55-58°C, and then decreased to maintain the water temperature. The evaporation temperature oscillates between -10°C and -5°C. The indoor temperature steadies between 19 and 21°C. During a cycle, the heat flow rate decreases from 22 kW to 12 kW while the COP decreases from 3.5 to 2.5. Due to the high water temperature, modes#2 and#3 are used: the HP begins a heating cycle in mode#1 (water temperature lower than 50°C), then switches to mode #2 to keep the heat flow high enough and when a high water temperature is needed switches to mode #3 (water temperature higher than 50°C). It is not clear how the control system chooses between mode#2 and mode#3 and if mode#2 is chosen, why it switches to mode#3. This choice is investigated below in the performance optimization section. Six defrosting cycles are also observed even if no clear sign of frosting appears on the evaporation temperature profile.
In standard operating conditions, the temperature difference between R410A and R134a in the evapo-condenser is in the range 7-10°C.

3.1.3. Day-averaged performance

The day-averaged COP_{SYS} values are presented in Figure 5 for the period October 2013 - January 2014. The figure exhibits the typical decrease with the outdoor temperature but with values quite interesting for a high-temperature heat pump.

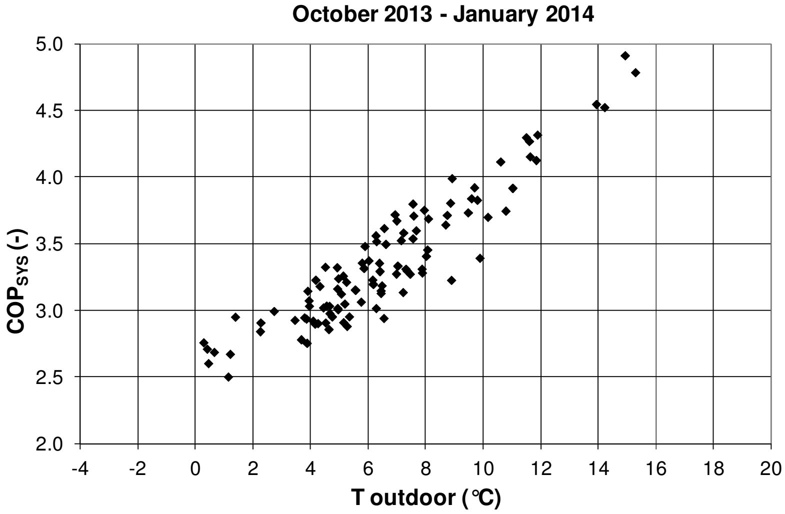

Figure 5: Day-averaged COP_{SYS} for space heating

3.1.4. Seasonal performance

Monthly performance values for space heating (from October 16, 2013 to January 31, 2014) are presented in Table 2 where several energy consumption indicators are presented: E_{HP} (compressors, evaporator fan), E_{WP} (condenser water pump), E_{DEF} (E_{HP}+E_{WP} for defrosting), E_{SB} (Heat pump stand-by energy, including condenser water pump when compressors are off). Q is the heat delivered to the water circulating in the radiators. $COPM_{SYS}$ includes E_{HP} and E_{WP} while $COPM_{TOT}$ includes all kinds of energies. We assumed that the stand-by energy is to be included in space heating performance and not in DHW production performance.

Table 2: Month performance for space heating

Month	E_{HP} (kWh)	E_{WP} (kWh)	E_{DEF} (kWh)	E_{SB} (kWh)	Q (kWh)	$COPM_{SYS}$ (-)	$COPM_{TOT}$ (-)
October	135.1	4.0	0.0	7.0	521.5	3.75	3.57
November	650.9	15.5	1.2	15.9	2229.2	3.35	3.26
December	963.2	18.5	2.0	17.3	2953.3	3.01	2.95
January	942.4	19.9	1.1	16.2	3039.4	3.16	3.10

Table 2 shows that E_{HP}, E_{WP} and Q exhibit the usual profile over the months: higher values during winter and lower values during mid-season. E_{WP} is low for October because the results were recorded for only half a month. Circulation pump energy E_{WP} and stand-by energy E_{SB} are low: they account for 2% each of the HP energy consumption, while defrosting energy is less than 0.5% of the total energy consumed.

3.2 Domestic hot water production

3.2.1. Day-averaged performance

DHW is produced by the same heat pump as the one used for space heating but cannot occur at the same time as space heating. DHW is usually produced at fixed periods of time during the night or the evening. The behavior of the HP is similar to what is presented in Figures 3 and 4 for space heating. Most of the time, heating in a DHW cycle begins in mode#1 and ends in mode#3 (mode#2 is not allowed for DHW). Figure 6 presents day-averaged COP_{SYS} values for the period October 2013 - December 2013. During the week days, hot water at a temperature of 55°C is produced. This temperature is increased to 65°C during the weekend. This explains the scattering of the results. We can expect a SPF above 2.5, which is excellent for this water temperature. This is obviously due to the use of the cascade cycle.

3.2.2. Seasonal performance

Monthly performance values for DHW production (from October 16, 2013 to January 31, 2014) are presented in Table 3 where similar indicators as for space heating are presented. Table 3 shows that E_{HP} and E_{WP} exhibit a profile similar to space heating while Q is more constant (DHW consumption does not depend on outdoor temperature).

4 PERFORMANCE OPTIMIZATION

4.1 Model of the heat pump

As mentioned above, it is not easy to understand how the HP switches from one mode to another one. Therefore, a model of the heat pump has been developed (Boivin 2013) to understand the thermal and performance behavior of the HP.

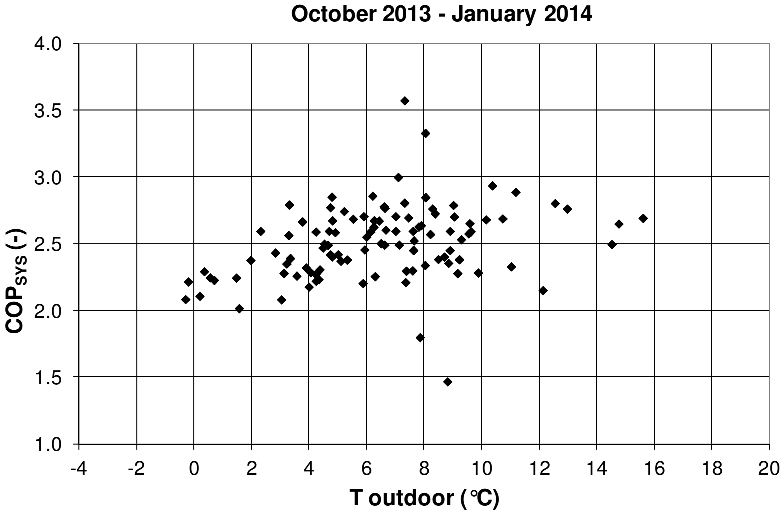

Figure 6: Day-averaged COP_{SYS} for DHW production

Table 3: Month performance for DHW production

Month	E_{HP} (kWh)	E_{WP} (kWh)	Q (kWh)	$COPM_{SYS}$ (-)
October	49.9	1.5	125.5	2.45
November	110.0	3.2	285.5	2.52
December	128.8	3.8	338.5	2.55
January	102.7	3.0	271.2	2.57

The model equations are not developed here: they will be published in another paper. The basic principles of the model are:

- compressors are modeled with a fixed (R134a compressor) or variable volumetric flow rate (R410A compressor), with for each compressor, a variable volumetric efficiency depending on the compression ratio and a constant isentropic efficiency;
- heat exchangers are modeled with a constant UA value and heat flow rate is computed with the standard ε-NUT method. It must be noted that each exchanger zone is modeled separately (evaporation, condensation, superheating, desuperheating and subcooling) and that the energy balance is performed on both sides of the heat exchanger (Dumont et al. 2008a);
- expansion valves are considered isenthalpic;
- we use a fixed degree of superheating for each evaporator and a fixed degree of subcooling for each condenser;
- the equation set is solved with standard Matlab functions (Fsolve).

The real values of the parameters have not been used due to the lack of experimental data by now. The monitored data will be used for model validation in a future work.
Therefore, most of the model parameters (UA values, volumetric efficiency functions, isentropic efficiencies, degrees of superheating and of subcooling) are standard values obtained on other heat pumps monitored by the Research Institute for Energy.
The volume flow rate of the R410A compressor has been chosen to match the datasheet performance of the HP at $T_{OUTDOOR}$=7°C and T_{WATER}=50°C. The volume flow rate of the R134a compressor has been taken to optimize the HP COP at $T_{OUTDOOR}$=0°C and T_{WATER}=50°C. For a given flow rate, the increase of the volume flow rate of the R134a compressor leads to an increase of the COP in mode #3 but to a decrease of the COP in mode#2. A balance between these two effects had to be realized (Boivin 2013).

4.2 Results of the model

The model has been used to compute the heat flow rate and the COP of the HP for the three HP modes. The results, obtained for fixed compressor speeds (50 Hz for the R410A compressor) and variable outdoor and inlet water temperature, are presented in Figure 7.
Mode#1 exhibits the typical air-source HP behavior: strong decrease of heat flow rate with decreasing outdoor temperature and weak decrease with increasing inlet water temperature, and strong decrease of COP with decreasing outdoor temperature or increasing inlet water temperature.
Mode#3 exhibits a less common behavior: there is also a decrease of the heat flow rate and of the COP with decreasing outdoor temperature but the heat flow rate increases with increasing inlet water temperature. This is useful to keep high heat flow values when high water temperature is needed (low outdoor temperature and use of radiators), but with the drawback of dramatically falling COP.

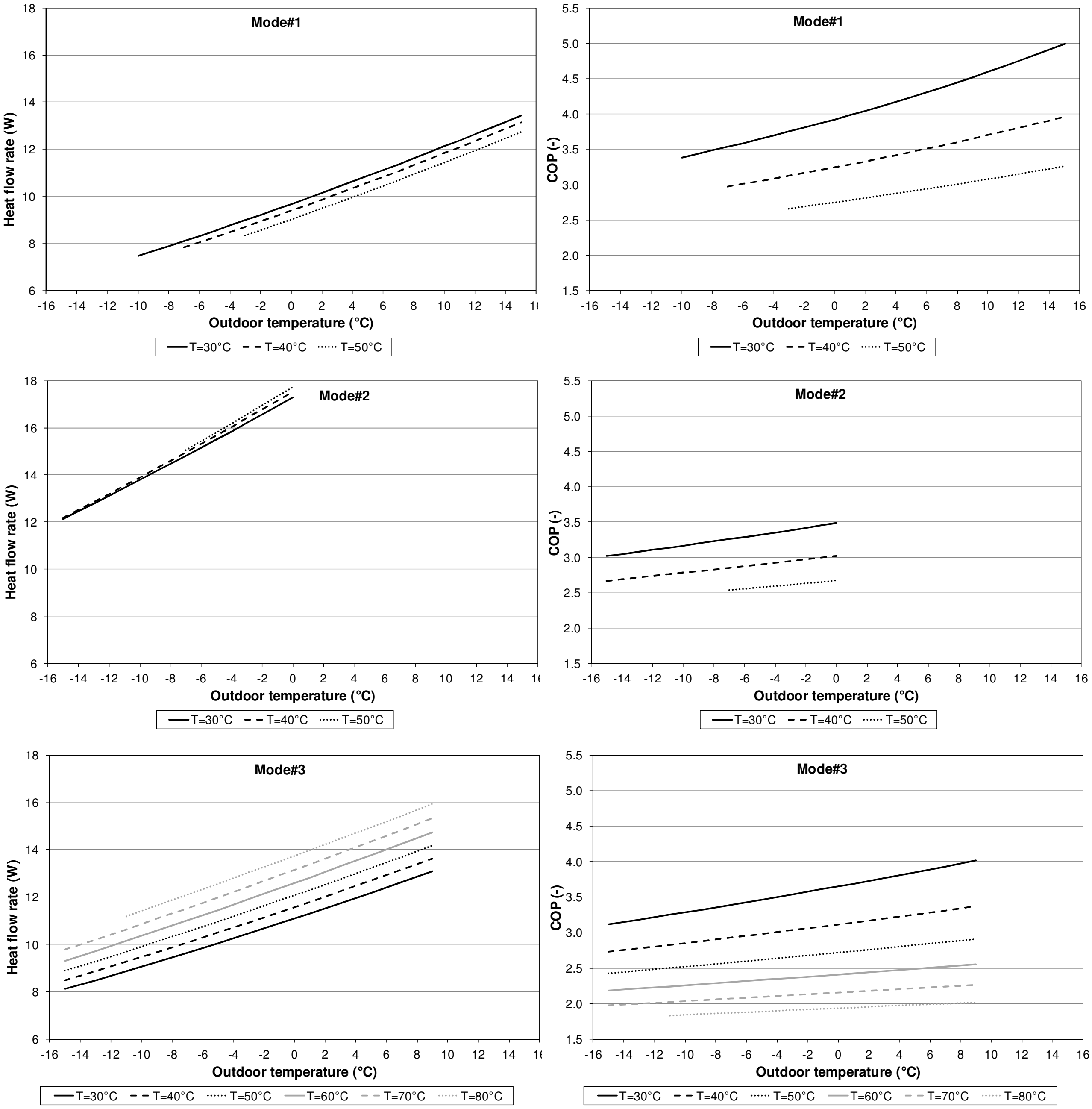

Figure 7: Heat flow rate and COP vs water and outdoor temperatures for the three HP modes

Mode#2 uses the two condensers. As it is a combination of Mode#1 and Mode#3, the heat flow rate of the R410A cycle decreases while the heat flow rate of the R134a cycle increases with increasing inlet water temperature. The sum of these two flow rates is nearly independent of the water temperature. Due to the use of both condensers, the heat flow rate is higher than in the two other modes.
From Figure 7, we can see that Mode#1 is used for low source-sink temperature differences (mild weather, floor heating). Mode#3 extends the behavior of mode#1 to higher source-sink temperature differences. Figure 7 shows that mode#3 COP curves are the extrapolation of mode#1 COP curves for low outdoor temperature. Mode#2 is used to boost the heat flow delivered by the heat pump when the two other modes cannot deliver the required heat, but with a worse COP. It cannot be used for high temperature water, for which only mode#3 is possible.
The complete analysis of the three HP modes for variable source and sink temperatures is synthetized in Table 4. In this table, the best mode to be used is presented in function of outdoor temperature and water temperature range. The best mode is the one for which the COP is maximum without having vapor temperature at the outlet of the compressor higher than 110°C.
We can see that mode#1 is used with mild weather conditions ($T_{OUTDOOR}$>-4°C) and for low to middle water temperature (T_{WATER}<55°C). It delivers small heat flow rates. Mode#2 is used for cold weather ($T_{OUTDOOR}$<-5°C) and low water temperature (T_{WATER}<35°C). Mode#3 is used for all the other conditions. There is also a zone where mode#1 and mode#2 yield similar performance. In this zone, mode#1 is able to deliver small heat flow rates while mode#2 delivers higher flow rates.
Table 4 also compares the HP mode obtained with the model with the HP mode coming from the HP manufacturer data. Both data agrees well except in 5 cases. For these, the mode proposed by the manufacturer is written in black. Manufacturer data just extends a bit the range of use of mode#2 alone. It has to be noted that the values used for the parameters of the model are not those of the real HP but are "standard" values.

Table 4: HP mode model versus manufacturer (yellow: mode#1, white: mode#1 or #2, green: mode#2, blue: mode#3)

	Water temperature (°C)						
T outdoor (°C)	<25	25-35	36-45	46-55	56-65	66-75	76-85
<-15							
-15 to -10			2				
-9 to -5				2			
-4 to 0		2	2				
1 to 5				2			
6 to 10							
11 to 15							
>15							

5 CONCLUSION

The performance of a high temperature heat pump installed in a non-insulated single-family house has been monitored since October 2013. The first results presented in this paper show a complex behavior with interesting COP, mostly due to the use of three different HP modes. A model of the HP has been developed to explain this complex behavior. It has been used to analyze the performance of the system in function of the source and sink temperatures and the results agree very well with manufacturer data. In a near future, the model will be used to optimize the operation of a combination of a high temperature HP with given radiators and house envelope. Therefore, the determination of the model parameters trough the analysis of the HP monitoring will be performed. The monitoring will also continue for a period of two years.

6 NOMENCLATURE

COP	Coefficient of performance of a heat pump [-]
COP_{SYS}	Coefficient of performance including only heat pump and water pump [-]
$COPM_{SYS}$	Monthly COP including only heat pump and water pump [-]
$COPM_{TOT}$	Monthly COP including all electric consumptions [-]
E_{DEF}	Seasonal electric consumption for the defrosting process [kWh]
E_{HP}	Seasonal electric consumption of the heat pump [kWh]
E_{SB}	Seasonal electric consumption during stand-by [kWh]
E_{WP}	Seasonal electric consumption of the condenser water pump [kWh]
p	Pressure [bar]
Po	Electric power [W]
q_V	Volume flow rate [m^3/s]
Q	Seasonal heat delivered to the house [kWh]
SPF	Seasonal performance factor of a heat pump [-]
T	Temperature [°C]
UA	Product of the overall heat transfer coefficient by the surface area of a heat exchanger [W/K]
T_{INDOOR}	Indoor temperature [°C]
$T_{OUTDOOR}$	Outdoor temperature [°C]

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