EXPERIMENTAL RESULTS OF AN AIR-TO-WATER HEAT PUMP WITH A VARIABLE SPEED COMPRESSOR

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Abstract: This paper presents results of a few months monitoring of an air-to-water residential heat pump used for space heating in a single-family house. The refrigerant used is R410A. The heat pump is equipped with a variable speed compressor. The scope of this project is to show how the COP of the heat pump is influenced by the variable-speed. When low variable speed is used, the phase change temperatures are close to the source and sink temperatures leading to an improvement of the COP. The thermal heat may be adapted to the thermal demand, avoiding the short-cycle working.

Key Words: ***heat pump, variable-speed***

1 INTRODUCTION

Depending on the Seasonal Performance Factor, heat pumps can be seen as a good alternative to classical heaters using gas or oil from the point of view of primary energy consumption and CO_2 releases. For example, considering domestic heating, the calculation method used in Belgium to determine the primary energy consumption of buildings shows that the use of a heat pump in a single-family house leads to a reduction of the primary energy consumption of 20 to 40% compared to oil high efficiency boilers.

For those reasons, the heat pump market is expanding in the whole world. For example, the heat pumps sales for a dozen of European countries have increased of 52% between the years 2005 and 2006 (EHPA, 2008).

The main problem when using an air-source heat pump for space heating is the correct choice of the balance point. A low balance point will able the heat pump to heat the house even under severe climate conditions but when the outside air temperature is higher, the heat pump will appear over-sized; the heating cycles will be short and the condensation temperature will increase rapidly. Such conditions will lower the energy performance of the system. A high balance point may require the use of a back up heating system.

The use of variable-speed compressor make possible the adaptation of the thermal power of the heat pump according to the operating conditions: the heat pump will deliver the needed capacity. As a consequence, the short-cycle phenomenon will not take place and, furthermore, in case of mild climate conditions, the heat exchangers will appear as over-sized compared to the compressor capacity, leading to high efficiencies and interesting COP values.

Although it is simple to understand, on the thermodynamic point of view, the likely advantages of using a variable-speed compressor heat pump, the real gain in terms of yearly electricity consumption is difficult to evaluate. This gain will depend on:

- the control strategy which is used;
- the effect of variable speed on the compressor efficiency;

- the design procedure.

Such information are difficult to obtain for commercial material. A better understanding of the influence of the control and design strategies on the real performance of a variable-speed compressor heat pump over a heating period is necessary to validate the use of such a technology and to develop the calculation methods of the primary energy consumption of buildings as required by the European regulation.

Among the last scientific works dealing with this topic, Karlsson and his co-authors (Karlsson, 2007) studied the energy-saving potential brought by the use of variable-speed capacity control instead of conventional intermittent operation mode for domestic ground source heat pumps. The originality of this study consists in the application of the variable-speed capacity to ground source heat pumps which is quite rare. Sakellari and his co-authors (Sakellari, 2006) deals with an investigation of the control strategies used in domestic low-temperature heat pump heating system. The authors have investigated, among other things, the ventilation rate and a prognostic climatic control.

The work presented in this paper is the first step of a complete project aiming at:
- quantifying the real energy impact of this technology;
- proposing a control strategy adapted to both the climate and the building.

This first step concerns the instrumentation of an air-to-water heat pump used for space heating in a well-insulated single-family house, the acquisition of data and their analysis.

Those points are the subjects of this paper. The measurements began in March 2007 and will be finished by March 2009.

2 STUDIED HEAT PUMP

2.1 Heat pump description

The studied system is a common-type vapor compression heat pump. It uses air as cold source, water as hot sink and R410A as refrigerant.

According to the manufacturer, the heat pump performances are : 16 kW of heating capacity, 4.73 kW of compressor power and COP of 3.38 for an indoor temperature of 20°C and an outdoor temperature of 6°C The working indoor temperature range is 17°C-28°C. The working oudoor temperature range is -20°C to 21°C.

Air is pulsed on a fin-and-tube heat exchanger in which evaporation takes place. This outdoor unit is 0.95 m width, 0.33 m depth and 1.35 m high. The condenser is a shell-and-tubes heat exchanger in which water is heated. The house is then heated by circulation of this hot water under the house floor.

The heat pump is equipped with a variable speed compressor and with an internal heat exchanger allowing heat recovery (the liquid flow leaving the condenser is subcooled while the vapor flow leaving the evaporator is superheated).

2.2 Instrumentation

In order to achieve the heat pump performance measurements, the whole system was completely instrumented (Figure 1). The different sensors measure temperatures, pressures, volume flow rates and electrical powers of the compressor and of the auxilaries. An electrical back-up heating system is integrated downstream the condenser.

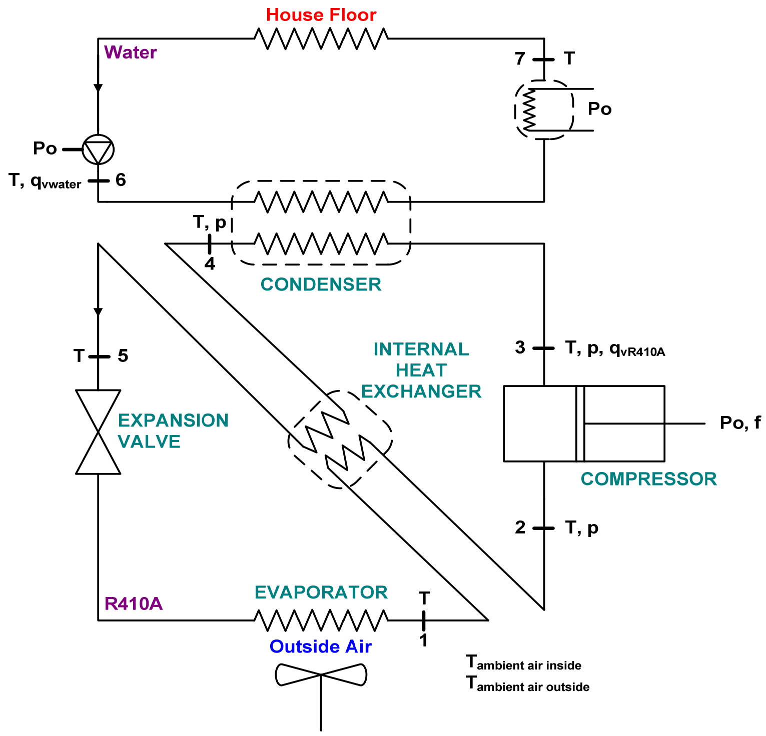

Figure 1: Instrumented heat pump

Each variable is measured every second by a data logger which calculates and stores mean values over one minute.

Table 1 presents the types and ranges of the different sensors.

Table 1: Measurement devices: location and operating range

Apparatus	Range	Location
Pressure transmitter	0 – 40 bar	2, 3, 4
Vortex flowmeter	0 – 2 dm³/s	3
Electromagnetic flowmeter	0 – 3 dm³/s	6
Temperature sensor	-100 – 600°C	1, 2, 3, 4, 5, 6, 7
	-50 – 50°C	Outside and inside air
Power analyzer	0 – 10 kW	Compressor + fan, electrical resistance
Power analyzer	0 – 200 W	Water pump
(Frequencymeter)		(Compressor)

Above all, it should be noticed that, at the time of writing this paper, it was impossible to measure the compressor frequency or its rotation speed because some electronic measurements components are missing.

Below, a brief description of the different measuring elements is presented.

2.2.1 Pressure transmitters

The pressure transmitters used can be equipped with a ceramic or metal sensor. In the first case, the system pressure acts on the ceramic diaphragm of the pressure sensor and displaces it. A capacitance variation, which is proportional to pressure, is measured by electrodes on the ceramic substrate and diaphragm. The measuring range depends on the thickness of the ceramic diaphragm. In the second case, the process pressure displaces the separating diaphragm and a filling liquid transmits the pressure to a resistance measuring bridge. The output voltage of the bridge (proportional to pressure) is then measured.

2.2.2 Temperature sensors

The used thermometers consist of a thin walled metal pipe containing the Pt100 Ω resistance having accuracy class A (tolerance: 0.15 + 0.002 I t I). The output signal is either the resistance signal or 4-20 mA.

2.2.3 Vortex flow measuring system

When the fluid (refrigerant in our case) flows past an obstacle in the measuring tube, vortices are formed on each side of this body with opposite rotation directions. When the vortices are detached or shed by the flow, they create a local low pressure area. The pressure changes are recorded by the sensor and converted to electrical pulses. As the development of vortices is regular, the frequency of the vortex shedding is proportional to the volume flow.

2.2.4 Electromagnetic flow measuring system

The measuring principle of this electromagnetic flowmeter is based on the Faraday's law of induction (a conductor moving in a magnetic field induces an electrical voltage). In this electromagnetic flowmeter, the moving conductor is the flowing fluid (in liquid state; in our case, water flow at the condenser).

Two field coils, placed at both sides of the measuring tube, generate a constant strength magnetic field by a pulsed direct current with alternating polarity. Two measuring electrodes are placed perpendicularly to the coils on the inside wall of the tube. They measure the voltage induced by the fluid flowing through the magnetic field which is proportional to the flow velocity and therefore to the volume flow which is calculated using the tube section.

2.2.5 Power analyzer

Power analyzers are placed for measuring the power of the compressor with the air fan (at the evaporator), the water pump and the electrical resistance.

They measure the voltage and current of those elements and then they are converted to an electric power.

2.2.6 Data recorder

A data manager Memograph is used for the measurements acquisition. It is both a videographic recorder and a compact measured values acquisition system.

There are 16 input channels (voltage, current or thermometer resistance signals). It plots signals and stores the data internally (then they can be archived on computer, memory cards, etc).

In our case, the measurements are done very second and the mean values calculated for one minute are stocked in the recorder. Regularly, these stocked values are downloaded on a computer.

2.3 Results treatment

The dew and bubble evaporation and condensation temperatures are calculated using respectively the measured low and high pressures.

Temperature T_3 and the high pressure at the outlet of the compressor permit to calculate the refrigerant density ρ_3.

$$\rho_3 = f(T_3, HP) \tag{1}$$

The mass flow rate of refrigerant, q_{mR410A}, is then calculated:

$$q_{mR410A} = \rho_3 q_{vR410A} \tag{2}$$

Temperatures at the inlet and outlet (T_3 and T_4) of the condenser and the high pressure permit to calculate the fluid enthalpies, h_3 and h_4.

$$h_3 = f(T_3, HP) \tag{3}$$

$$h_4 = f(T_4, HP) \tag{4}$$

The heat flow at the condenser, Φ_{CR410A}, is then calculated:

$$\Phi_{CR410A} = q_{mR410A}(h_4 - h_3) \tag{5}$$

The density of water at the condenser inlet, ρ_6 is calculated knowing the temperature T_6:

$$\rho_6 = f(T_6) \tag{6}$$

The mass flowrate of water, q_{mwater}, is then calculated:

$$q_{mwater} = \rho_6 q_{vwater} \tag{7}$$

The mean temperature of water in the condenser, T_{mwater}, is calculated using the inlet, T_6, and outlet, T_7, temperatures.

$$T_{mwater} = \frac{T_6 + T_7}{2} \tag{8}$$

The specific heat at constant pressure of water, c_{pwater}, is calculated at the mean temperature of water in the condenser.

$$c_{pwater} = f(T_{mwater}) \tag{9}$$

The heat flow at the condenser considering the water flow, Φ_{Cwater}, is then calculated:

$$\Phi_{Cwater} = q_{mwater} c_{pwater} (T_7 - T_6) \tag{10}$$

The two values of heat flow at the condenser, Φ_{CR410A} and Φ_{Cwater}, are compared. They should be identical.

As the different powers are measured (i.e. the electrical power of the compressor and fan, $P_{comp+fan}$, and the power of the water pump, P_{water}), the total power, P_{total}, is calculated (equation 11) and then the COP of the heat pump (equation 12):

$$P_{total} = P_{comp+fan} + P_{water} \tag{11}$$

$$COP = \frac{\Phi_{Cwater}}{P_{total}} \tag{12}$$

3 RESULTS AND COMMENTS

3.1 Daily results

The results presented below were obtained the 5th December 2007. That day, the mean outside air temperature was 11.6°C. It was thus a mild winter day.

The temperature inside the house was stable and pleasant during the whole day (around 21°C).

Figure 2 presents the evolution of the outside air temperature and mass flow rate of refrigerant during the whole day.

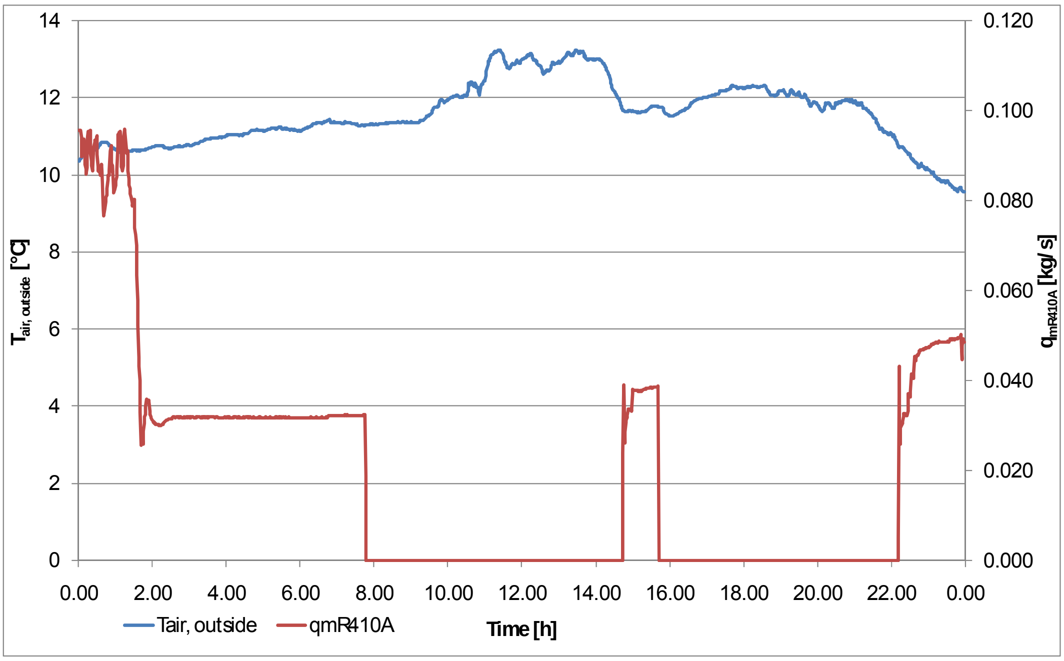

Figure 2: Evolution of $T_{air, outside}$ and q_{mR410A} with time

From midnight to nearly 2.00 AM, the mass flow rate of refrigerant is relatively constant and between 0.08 and 0.1 kg/s. There are slight a periodic variations of the rotation speed of the compressor during this period. After 2.00 AM, it is set to a low value and the mass flow rate of refrigerant is 0.032 kg/s. The installation is then stopped at 8.00 AM. During the day cycle, around 3.00 PM, the rotation speed seems also to be set to a constant value; the refrigerant mass flow rate is 0.038 kg/s. The beginning of the last cycle is characterized by a rising mass flow rate during the first hour.

Figure 3 presents the evolution of outside air temperature and evaporation temperature during the day.

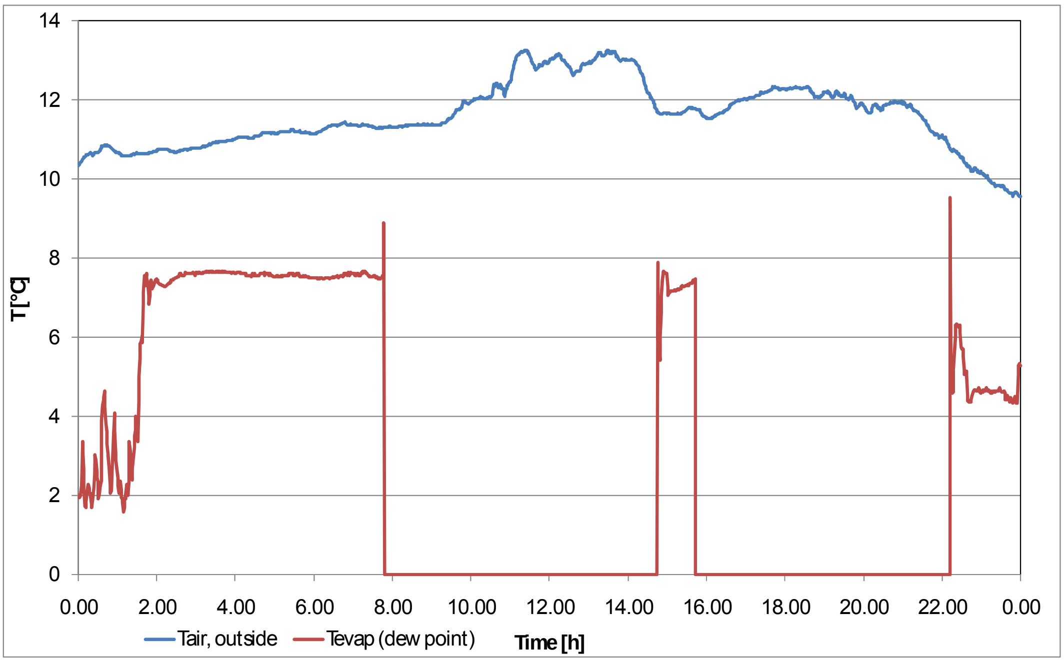

Figure 3: Evolution of $T_{air,\ outside}$ and $T_{evaporation}$ with time

From midnight to 2.00 AM, the evolution of the evaporation is directly linked to the variation of the mass flow rate. The average temperature difference between the outside air and the refrigerant is around 10°C which is a quite usual value. Each time the mass flow rate decreases the temperature difference decrease. After 2.00 AM, the rotation speed and thus the mass flow rate value of refrigerant are so low that less heat is distributed and thus the difference between the evaporation and the outside air temperature is very low (around 4°C). During the last cycle the slow increase of the compressor speed during the first hour lead to a decrease of the evaporation temperature during the same period.

A potential advantage of the use of the variable speed is pointed out. If the outside air temperature is around 7°C and if the rotation speed is low, the evaporation temperature is high and thus the frost formation is delayed.

Figure 4 presents the evolution of the condensation temperature of R410A and of water temperature at the inlet and at the outlet of the condenser during the period.

During the first part of the night cycle, the condensation temperature varies from 47 to 57°C. After 2.00 AM, when the rotation speed is lower, this condensation temperature is reduced to 38-39°C. During the day cycle, the condensation temperature is slightly lower (around 35°C).

The evolution of water temperatures is quite usual.

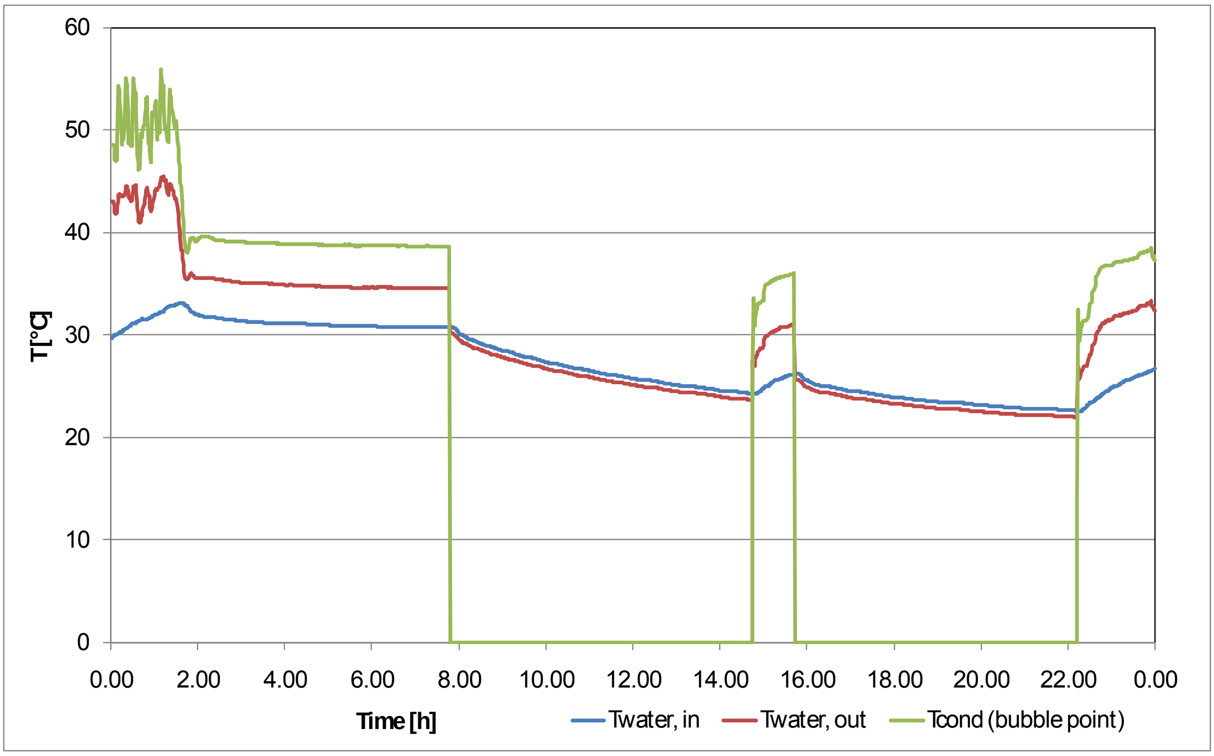

Figure 4: Evolution of $T_{condensation}$, $T_{water,\ in}$, $T_{water,\ out}$ with time

Figure 5 presents the evolution of the outside air temperature and heat flow released at the condenser during the period.

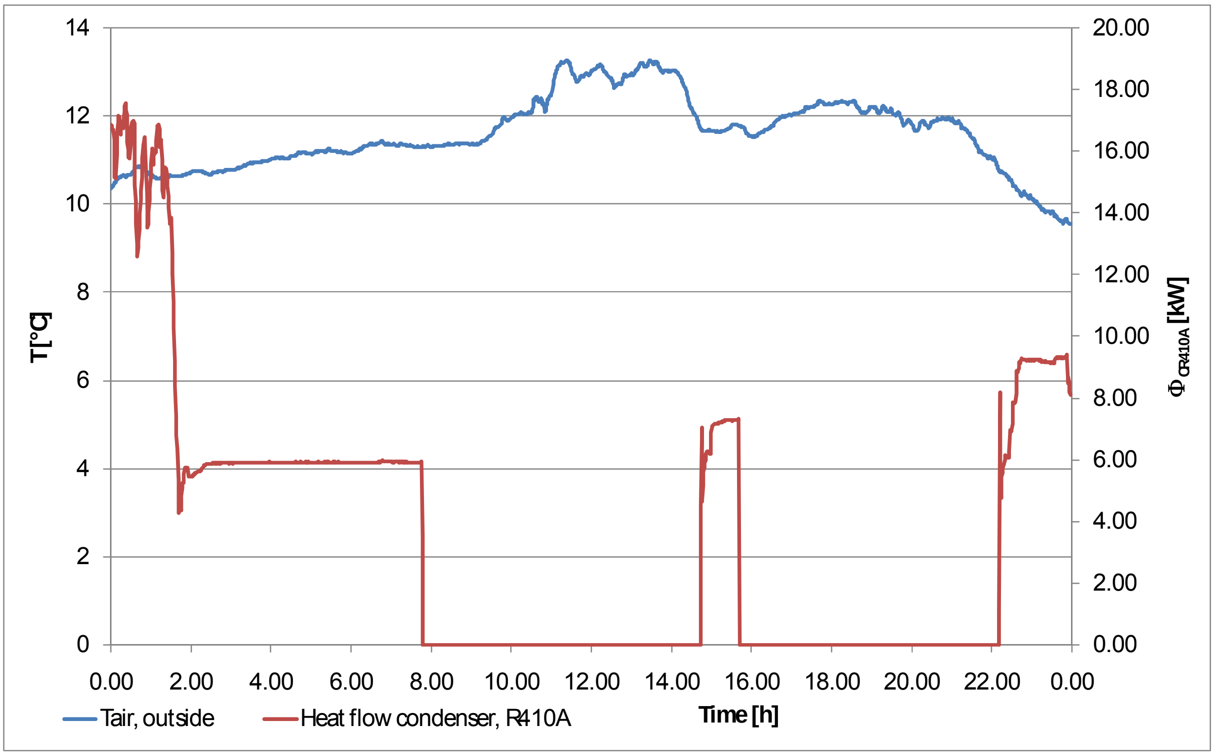

Figure 5: Evolution of $T_{air,\ outside}$ and Φ_{CR410A} with time

This Figure shows the extreme variability of the thermal power with compressor speed: it decreased from nearly 12 kW at midnight to 4 kW at 2.00 AM. This allows a good adaptation of the thermal power to the thermal demand and prevents from the short-cycle phenomenon.

Figure 6 presents the evolution of the COP during the period.

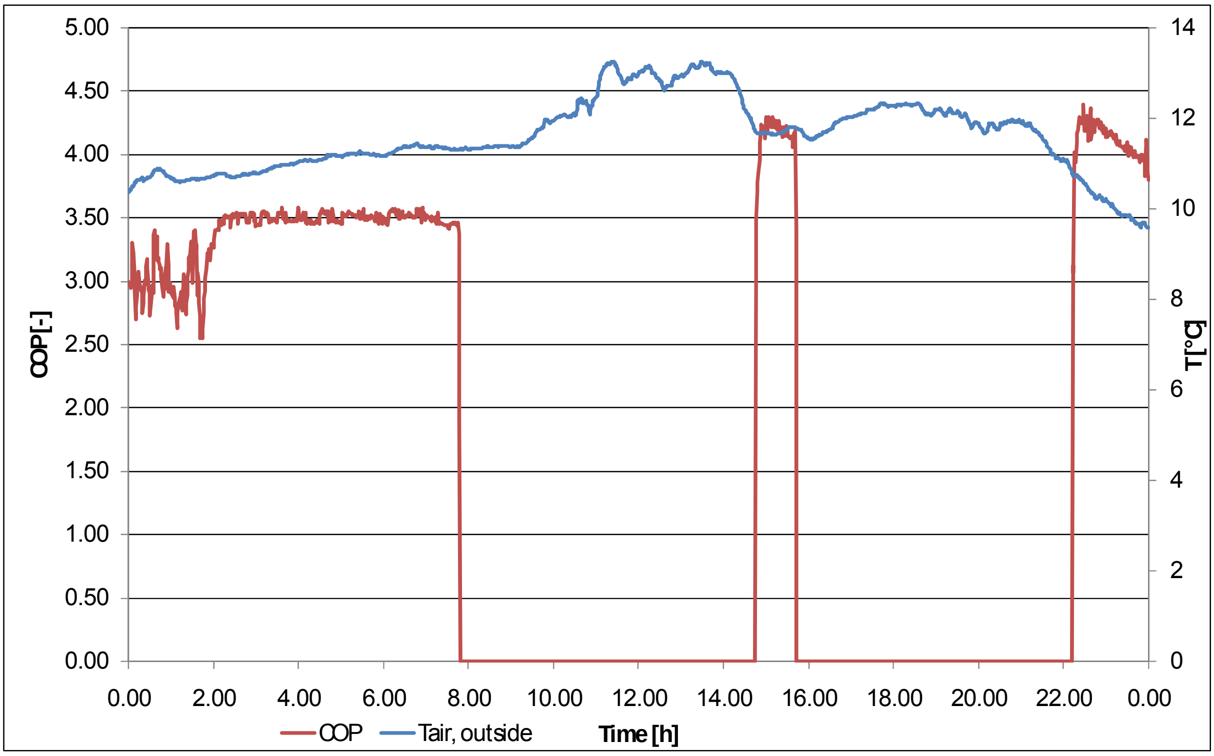

Figure 6: Evolution of the COP with time

At the beginning of the night cycle, the condensation temperature is high and the evaporation temperature is low; the COP is around 3.0. After 2.00 AM, the condensation temperature is lower and closer to the house temperature while the evaporation temperature is close to the outside temperature, the heat pump performances are thus better: COP increases up to 3.5.

During the day cycle, the combination of low speed working and high outside temperature leads to high COP values (4.5). The last cycle shows a continuous decrease of the COP due the increase of the compressor speed and to the lowering outside temperature

3.2 Global results

Table 2 presents a summary of the heat pump experimental performance for nearly one year (e.g. electrical consumption of the compressor and fan, electrical consumption of water pump, heat transferred to water, COP, night running time).

Table 2: Heat pump experimental results

	Electrical consumption Compressor + fan [kWh]	Electrical consumption Water pump [kWh]	Heat Water side [kWh]	Heat R410A side [kWh]	COP [-]	Total running time [%]	Night running time [%]
March*	331.77	26.09	1103.85	1118.34	3.08	38.49	65.30
April	201.17	30.25	675.03	758.54	2.92	14.52	63.82
May	0	0	0	0	0	0	0
June	0	0	0	0	0	0	0
July	0	0	0	0	0	0	0
August	0	0	0	0	0	0	0
September	64.80	9.98	278.41	278.41	3.6	5.06	63.10
October	444.04	50.82	1624.01	1624.01	3.18	28.68	81.16
November	845.49	52.16	2615.46	2615.46	2.88	42.72	83.41
December*	686.57	36.4	2026.94	2026.94	2.78	51.83	71.74
Total	**2573.84**	**205.7**	**8421.69**	**8421.69**	**2.96**	**28.18**	**74.76**

The results for the months of March and December are not complete: the measurements have begun the 16th of March and those obtained after the 20th of December have not been treated yet.

The COP is good and varies from 2.78 to 3.6 for nearly a year.

The heat delivered to the house is coherent with the insulation level of the house. It can also be noticed that the heat calculated at both sides (water and refrigerant) are equivalent.

For cold months, when the outside air temperature decreases, it can be seen that the total running time of the heat pump increases. The heat pump works essentially during the night in order to benefit of the electricity lower price.

4 CONCLUSIONS

This work is the first part of a study which aims at studying the effects of the variable speed on heat pump performances used for space heating

In this paper, the first experimental results obtained on a real installation are presented.

Globally the likely behavior expected from a thermodynamic analysis is pointed. When low variable speed is used, the phase change temperatures are close to the source and sink temperatures leading to an improvement of the COP. The thermal heat may be adapted to the thermal demand, avoiding the short-cycle working.

As a consequence, quite high average COP values may be observed for the beginning and the end of the heating period (mild conditions) as expected. The observed values in winter are more normal (close to what is expected from a heat pump without heat capacity modulation). The average COP observed for the whole period is just good without being exceptional.

It has been shown that the heat flow released at the condenser of the heat pump in the beginning of the night cycle was huge (three times higher than the value obtained with the low rotation speed). During this period, the COP was around 3.0 while it increases up to 3.5 when using correctly the variable speed. This shows that the speed modulation (capacity control) does not seem to be used in a very efficient way: the purpose is to maximize the night working for electricity price reasons.

In the future works, the variable speed has to be measured in order to verify the accuracy of our conclusions. This measurement will also allow us to determine the real control strategy which is used on the commercial heat pump. Modeling developments will be conducted based the experimental results in order to propose more efficient control strategies.

5 NOMENCLATURE

c_p	Specific heat at constant pressure [J kg^{-1} K^{-1}]
h	Enthalpy [J kg^{-1}]
HP	High pressure [Pa]
P	Power [W]
q_m	Mass flow rate [kg s^{-1}]
q_v	Volume flow rate [m^3 s^{-1}]
T	Temperature [K]
Φ_C	Heat flow released at the condenser of the heat pump [W]
ρ	Density [kg m^{-3}]

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