PERFORMANCE ANALYSIS AND MODELING OF A STATIC AIR-TO-WATER HEAT PUMP INTEGRATED IN A SINGLE-FAMILY DWELLING

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Abstract: This paper presents the results of the modeling of a static air-to-water heat pump installed in a single-family dwelling in Belgium. We present a model combining the heat pump and the house. Firstly, we develop sub-models for the heat pump itself, for the heating floor and for the thermal behavior of the house. These sub-models have been validated separately. The results of the heat pump model show good agreement with the experimental results. For the model combining the heat pump and the heating floor, the simulation results are as good as those obtained with the heat pump alone, thus validating the heating floor model. The use of the whole system model shows higher discrepancies with experience because the thermal model of the house is a single zone model. However, simulation results over one year give good total values only 4 to 5% lower than experimental ones.

Key Words: ***air source heat pumps, model, building thermal model***

1 INTRODUCTION

Heat pumps are interesting heating systems because they pump heat from a costless source to a useful sink, using less energy for pumping than the amount of heat delivered. Their efficiency, called coefficient of performance (COP) is always greater than 1. The energetic (non renewable primary energy consumption) and environmental (CO_2 production) gains resulting from the use of heat pumps (compared to high efficiency boilers using natural gas) depend on their average SPF value and on the average efficiency of the electrical power plants as well as on the kind of primary energy they use. In Belgium, the average gain in terms of non renewable primary energy consumption is about 40%. From an economical point of view (energy costs), the high electricity cost appeals for high SPF values (up to 3.4) or for special regulation systems which promote the heat pump running during the night. Anyway, the correct determination of the SPF value is of prime importance when the use of heat pump is chosen as heating system. SPF values can be determined either by monitoring real systems, what can be performed only on few installations, or by simulation. But simulation results have first to be validated by experimental measurements.
The scope of this project is firstly the measurement of the performance of an experimental air-source heat pump installed in a single-family dwelling in Belgium, and secondly, the development of a thermal model for the system composed of the heat pump and the house, to be validated by experimental measurements. The project is a sequel of a larger study devoted to a 2-year monitoring of nine heat pumps used for space heating and two heat pumps used for sanitary water heating (Frère et al. 2004).

2 EXPERIMENTAL

2.1 Heat pump and house

The heat pump investigated is a vapor compression static air-to-water heat pump which uses R404A as refrigerant. A static finned tube heat exchanger (54 m^2 of finned tubes) placed against the south outside fence of the backyard is fed with a glycol-water blend which takes heat from the outdoor air. The glycol-water blend brings the heat to the evaporator (plate heat exchanger located inside the house). Condensation takes place in a plate heat exchanger and heat is supplied to the house by water circulating in the floor (figure 1). There is no electrical heater inside the rooms but for extreme weather conditions, an electrical resistance located downstream the condenser is used to assist the heat pump (it was never used during the monitoring period).
The source being the outdoor air is static and its temperature is known to vary from -5°C to 15°C during the heating period (average Belgian weather conditions). The sink is the house floor, which is also a "static" sink. Its temperature varies as a function of the duration of the heating cycle. The house is a two storey well insulated detached house covering a surface area of 177 m^2 (for both floors).

2.2 Experimental results

The most important value to be monitored is the COP, which is defined as the ratio of the heat flow released by the condenser and the electrical power used by the installation. The measurements performed on the heat pump, the monitoring method and detailed results have been described elsewhere (Dumont et al. 2007). Here, we just present some results obtained for a whole year.
Instantaneous COP can be calculated considering the compressor consumption only (COP_{COMP}) or considering the compressor and auxiliary pump consumptions (COP_{HP}). The results for one year are presented in Table 1 and take into account the auxiliary pump. Running costs depend on the moment of power consumption (peak or off-peak), the amount of off-peak electricity consumption is then given (off-peak percentage). The costs are based on average Belgian electricity market prices: 0.18 Eur/kWh (peak) and 0.09 Eur/kWh (off-peak). As a comparison, running costs for the same amount of released heat using electrical heaters, fuel oil and natural gas burners are also given. The last two costs are based on burner efficiencies of 0.9 and on Belgian fuel market prices (0.5 Eur/l for fuel oil, 0.05 Eur/kWh for natural gas).

Table 1: Measured annual performance

Period	Dec 2005 – Nov 2006
E_{YEAR} (kWh)	3103
Q_{YEAR} (kWh)	8744
SPF (-)	2.82
Off-peak perc. (%)	52.0
Cost HP (Eur)	413
Cost Gas (Eur)	486
Cost Fuel oil (Eur)	523
Cost Elec (Eur)	1163

3 MODELS

The model of the system combining the heat pump and the house involves three sub-models: the heat pump itself, the heating floor in the house and finally the house. We have developed two models, the first one based on the commercial software TRNSYS (Solar

Energy laboratory 2004), the other one based on our own code developed in Matlab®, in order to evaluate both methods. Each solution has its advantages and drawbacks, e.g. the Matlab® solution being able to master every parameter of the model but imposing the complete development of the code from scratch. Only the Matlab® model is presented here. The TRNSYS model will be presented elsewhere.

3.1 Model of the heat pump

The model of the heat pump is a steady-state model and is based on the Fripac model presented elsewhere (Dumont et al. 2002, Dumont et al. 2003). It is based on a reverse Rankine cycle presented in figure 1 for a pure fluid.
The following assumptions are made: the superheating process ΔT_{SH} (1-2) takes place in the evaporator, the compression (2-3) is characterized by the isentropic efficiency (η_{ISOS}) and a pseudo-electrical efficiency (η_{ELEC}), the de-superheating process (3-4) as well as the subcooling process ΔT_{SC} (5-6) take place in the condenser, the expansion through the thermostatic expansion valve (6-7) is an isenthalpic process and pressure drops in the heat exchangers (3-6) and (7-2) are neglected.

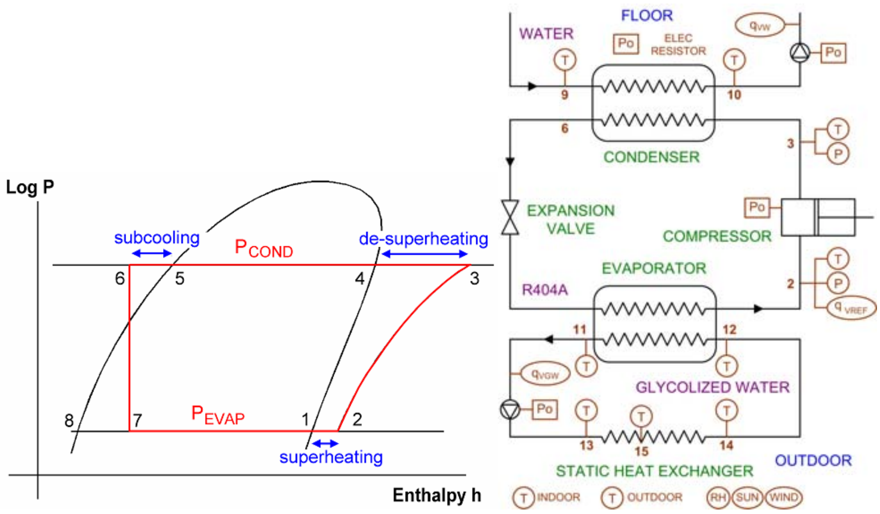

Figure 1: a) Thermodynamic cycle b) measurements performed on the heat pump

The parameters of the model are: nature of refrigerant and secondary fluids, mass flow rate of the secondary fluid at the evaporator and at the condenser q_{SEVAP} and q_{SCOND}, inlet temperature of the secondary fluid at the evaporator and at the condenser $T_{SEVAPIN}$ and $T_{SCONDIN}$, swept volumetric flow rate of the compressor q_V, dead volume ratio of the compressor ε, compressor isentropic and pseudo-electrical efficiencies η_{ISOS} and η_{ELEC} and superheating and subcooling temperature changes ΔT_{SH} and ΔT_{SC}. The model requires the overall heat transfer coefficients U_{COND} (condenser) and U_{EVAP} (evaporator) and the heat transfer surface areas A_{COND} (condenser) and A_{EVAP} (evaporator).
The thermodynamic and the transport properties of the refrigerant are calculated by the REFPROP 7.0 routines (NIST) while those for the secondary fluids are calculated by specific routines.
The equations of the heat pump model are:
- energy conservation for the different processes at the evaporator (evaporation and superheating) and at the condenser (de-superheating, condensation and subcooling):

$$q_{REF}\left(h_1 - h_7\right) = q_{SEVAP}\ c_{PSEVAP}\ \left(T_{SEVAPMID} - T_{SEVAPOUT}\right) = U_{EVAP}\ A_{EVAP}\ F_{EV}\ \ LMTD_{EV} \qquad (1) - (2)$$

$$q_{REF}\left(h_2 - h_1\right) = q_{SEVAP}\ c_{PSEVAP}\ \left(T_{SEVAPIN} - T_{SEVAPMID}\right) = U_{EVAP}\ A_{EVAP}\ (1 - F_{EV})\ LMTD_{SH} \qquad (3) - (4)$$

$$q_{REF}\left(h_3-h_4\right)=q_{SCOND}\ c_{PSCOND}\ \left(T_{SCONDOUT}-T_{SCONDMID2}\right)=U_{COND}\ A_{COND}\ F_{DSH}\ LMTD_{DSH} \tag{5 - 6}$$

$$q_{REF}\left(h_4-h_5\right)=q_{SCOND}\ c_{PSCOND}\ \left(T_{SCONDMID2}-T_{SCONDMID1}\right)=U_{COND}\ A_{COND}\ F_{CO}\ LMTD_{CO} \tag{7 - 8}$$

$$q_{REF}\left(h_5-h_6\right)=q_{SCOND}\ c_{PSCOND}\ \left(T_{SCONDMID1}-T_{SCONDIN}\right)=U_{COND}\ A_{COND}\ (1-F_{CO}-F_{DSH})\ LMTD_{SC} \tag{9 - 10}$$

- model for the volumetric efficiency of the compressor:

$$q_{REF}=q_V\ \rho_2\ \eta_V\left(\varepsilon,\frac{P_{COND}}{P_{EVAP}},c_{P2},c_{V2}\right) \tag{11}$$

From equations (1) to (11), the values of q_{REF}, T_{EVAP}, T_{COND}, $T_{SEVAPOUT}$, $T_{SEVAPMID}$, $T_{SCONDOUT}$, $T_{SCONDMID1}$, $T_{SCONDMID2}$, F_{EV}, F_{CO} and F_{DSH} can be calculated. Then, P_{EVAP}, P_{COND}, T_1 to T_8 as well as the heating capacity (Φ_{COND}), the compressor electrical power (P_{ELEC}), and the coefficient of performance (COP) can be estimated:

$$\Phi_{COND}=q_{REF}\left(h_3-h_6\right) \tag{12}$$

$$P_{ELEC}=\frac{q_{REF}\left(h_3-h_2\right)}{\eta_{ELEC}} \tag{13}$$

$$COP=\frac{\Phi_{COND}}{P_{ELEC}} \tag{14}$$

The electrical consumption of the two auxiliary pumps are not taken into account in the model.

3.2 Model of the thermal behavior of the house

The model for the building is a dynamic model representing its thermal behavior. It is based on a resistor-capacity network and has been derived from a more detailed model described elsewhere (Anciaux et al. 2006). In this network, 10 nodes are defined: 1 for the outdoor air (temperature $T_{OUTDOOR}$), 4 for the walls (temperatures T_{WALL}, 3 layers), 1 for the indoor air (temperature T_{INDOOR}), 3 for the heating floor (temperatures T_{FLOOR}, 2 layers), and 1 for the ground below the building (T_{GROUND}) (figure 2). This model describes what is called a single zone model.

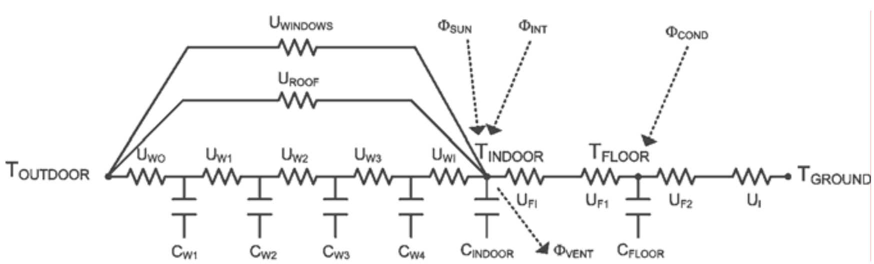

Figure 2: Network model for the building

The heat flow rate for the heating of the building (Φ_{COND}) is connected to the heating floor node, heat coming from the heat pump. At the indoor node, other heat flow rates are also

connected: the ventilation heat flow rate (Φ_{VENT}), the internal heat generation (Φ_{INT}) and the solar contribution through the windows (Φ_{SUN}).
The parameters of the model are the thermal transmittances (U) and capacities (C) of the network, which are related to the composition of the walls, the windows, the roof and the heating floor. Geometrical parameters like surface areas (A) for walls, windows, roof and heating floor are also needed. Operating parameters are the ground temperature $T_{GROUND,}$ the outdoor temperature $T_{OUTDOOR}$, $\Phi_{VENT,}$ Φ_{INT} and Φ_{SUN}.
The equations of the model are described by 6 differential equations:

$$C_{W1}\frac{dT_{WALL1}}{dt} = U_{W1}A_{WALL}\left(T_{WALL2}-T_{WALL1}\right)+U_{WO}A_{WALL}\left(T_{OUTDOOR}-T_{WALL1}\right) \tag{15}$$

$$C_{W2}\frac{dT_{WALL2}}{dt} = U_{W2}A_{WALL}\left(T_{WALL3}-T_{WALL2}\right)+U_{W1}A_{WALL}\left(T_{WALL1}-T_{WALL2}\right) \tag{16}$$

$$C_{W3}\frac{dT_{WALL3}}{dt} = U_{W3}A_{WALL}\left(T_{WALL4}-T_{WALL3}\right)+U_{W2}A_{WALL}\left(T_{WALL2}-T_{WALL3}\right) \tag{17}$$

$$C_{W4}\frac{dT_{WALL4}}{dt} = U_{WI}A_{WALL}\left(T_{INDOOR}-T_{WALL4}\right)+U_{W3}A_{WALL}\left(T_{WALL3}-T_{WALL4}\right) \tag{18}$$

$$\begin{aligned}C_{INDOOR}\frac{dT_{INDOOR}}{dt} &= U_{WI}A_{WALL}\left(T_{WALL4}-T_{INDOOR}\right)+(U_{FI}+U_{F1})A_{FLOOR}\left(T_{FLOOR}-T_{INDOOR}\right)\\ &+\left(U_{ROOF}A_{ROOF}+U_{WINDOWS}A_{WINDOWS}\right)\left(T_{OUTDOOR}-T_{INDOOR}\right)-\phi_{VENT}+\phi_{INT}+\phi_{SUN}\end{aligned} \tag{19}$$

$$\begin{aligned}C_{FLOOR}\frac{dT_{FLOOR}}{dt} &= (U_{FI}+U_{F1})A_{FLOOR}\left(T_{INDOOR}-T_{FLOOR}\right)\\ &+(U_{F2}+U_{I})A_{FLOOR}\left(T_{GROUND}-T_{FLOOR}\right)+\phi_{COND}\end{aligned} \tag{20}$$

T_{FLOOR}, T_{INDOOR} and the different T_{WALL} can be computed from equations (15) to (20).

3.3 Model of the heating floor

The model for the heating floor is a steady-state model. On a physical basis, this floor is a concrete slab with tubes where hot water from the heat pump circulates. An insulation layer is placed under the slab in order to isolate the concrete from the ground. The heat transfer equations between the water in the tubes and the slab are fundamentally two-dimension equations. This problem can be shown to be equivalent to a one-dimension equation, reduced to a lumped-parameter equation, if particular geometrical assumptions are made, which is the case for the heating floor in the house being monitored. The detailed model is described in (Solar Energy laboratory 2004).
The parameter U_{HF} is related to the composition and geometrical characteristics of the tubes and of the concrete, and to the heat transfer mode between water and concrete.
The model equation is:

$$T_{SCONDIN} = T_{FLOOR} + (T_{SCONDOUT} - T_{FLOOR})\exp\left(-\frac{U_{HF}A_{FLOOR}}{q_{SCOND}c_{P\,SCOND}}\right) \tag{21}$$

It relates the water temperature at the inlet and the outlet of the condenser ($T_{SCONDIN}$ and $T_{SCONDOUT}$) of the heat pump to the heating floor temperature at the tubes level T_{FLOOR}.

3.4 Model of the whole system

The set of equations (1) to (11) and (15) to (21) describes a single model of the system heat pump-heating floor-building. The connection between the heat pump model and the heating floor model is done through the 3 following variables/parameters: $T_{SCONDIN}$, $T_{SCONDOUT}$ and q_{SCOND}. The heat pump model and the building model are connected through the derived-variable Φ_{COND} while the heating floor model and the building model are connected through variable T_{FLOOR}.
The parameter set for the model is:

1. Device specific parameters:
   For the building and the heating floor
   - Composition dependent: transmittances and capacities U and C;
   - Geometry dependent: surface areas A;

   For the heat pump
   - Thermodynamic cycle dependent: superheating and subcooling ΔT_{SH}, ΔT_{SC};
   - Compressor: q_V, ε, η_{ISOS}, η_{ELEC};
   - Condenser: U_{COND}, A_{COND};
   - Evaporator: U_{EVAP}, A_{EVAP}.
2. Operating parameters:
   For the building
   - Heat gains and losses: Φ_{INT}, Φ_{VENT};
   - Climate dependent: Φ_{SUN}, T_{GROUND} and $T_{OUTDOOR}$;
   - T_{INDOOR}.

   For the heat pump
   - Secondary fluid flow rates: q_{SEVAP}, q_{SCOND};
   - $T_{SEVAPIN}$.

For this set, we have to make two remarks: firstly, $T_{SEVAPIN}$ is an operating parameter because no model for the static air heat exchanger has been developed. Secondly, T_{INDOOR} is also an operating parameter if its value is predefined to a setpoint value. Indeed, the simulation of the behavior of the whole system over one year can be performed following two approaches:
- the heat pump is controlled so that T_{INDOOR} cannot be lower than a predefined setpoint, allowing the heat pump to switch on and off when heat is needed or not;
- the heat pump is controlled following an on/off pattern (e.g. the experimental pattern) so that T_{INDOOR} is calculated for the whole year and is not an operating parameter anymore.

4 RESULTS

The model has been compared to experimental results obtained thanks to the monitoring of the heat pump. Each sub-model has been investigated separately. Therefore, the simulation uses the experimental on/off pattern of the heat pump. Each simulation uses constant device specific parameters but also some time-variable operating parameters like the heat gain and losses, the climate dependent parameters and $T_{SEVAPIN}$. The integration time-step for the differential equations (15) to (20) is 6 min (0.1 h). The fsolve subroutine of Matlab® has been used for solving the whole set of equations.

4.1 Results for the heat pump model

The device specific parameters of the heat pump have been obtained by analyzing the experimental results. As already shown previously (Dumont and Frère 2005, Dumont et al. 2007), they can be assumed constant over one year: ΔT_{SH}=3.0°C, ΔT_{SC}=6.0°C, q_V=1.865 10^{-3}

m^3/s, ε=0.0434, η_{ISOS}=0.58, η_{ELEC}=0.97, $U_{EVAP}A_{EVAP}$=1200W/K, $U_{COND}A_{COND}$=800W/K. Two operating parameters are also constant: q_{SEVAP}=0.36kg/s and q_{SCOND}=0.27kg/s while the third one $T_{SEVAPIN}$ is equal to the experimental values. As we want to assess the heat pump model only, we used the experimental water inlet temperature $T_{SCONDIN}$.
The simulation has been performed for a period of time of several months, using the experimental on/off pattern of the heat pump. It gives good agreement with the experimental values as shown on figure 3 for 11th and 12th December 2005.

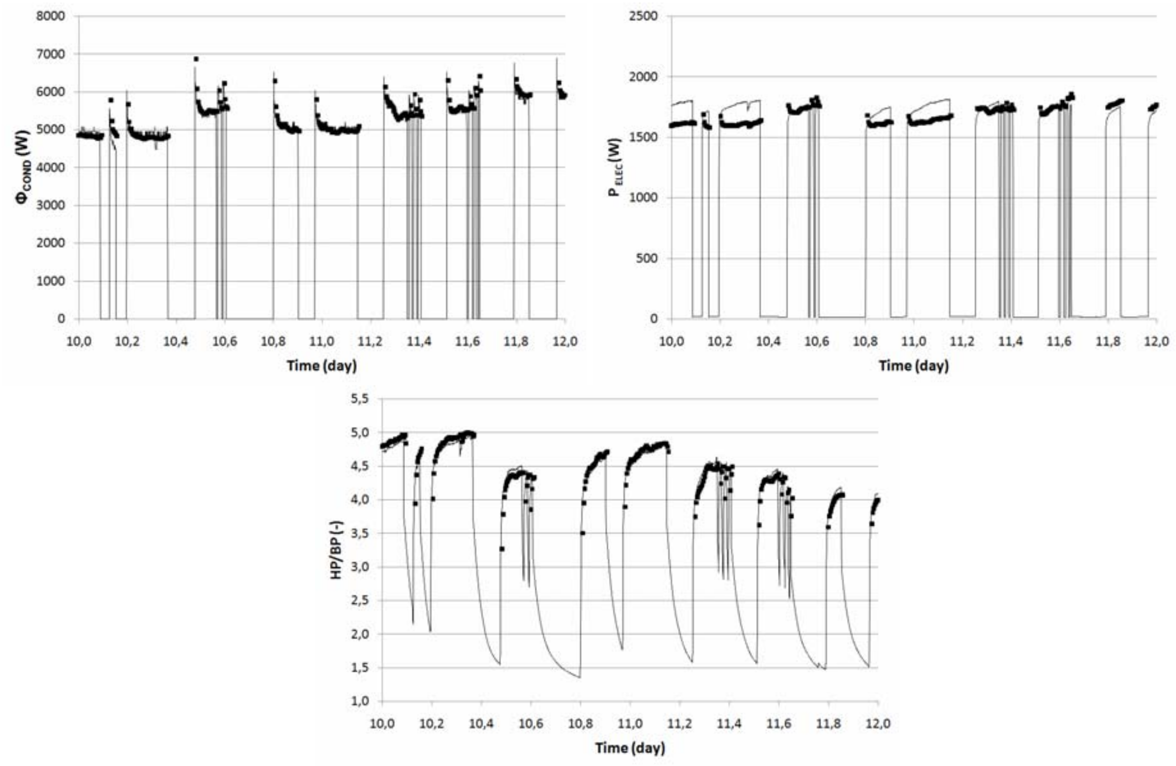

Figure 3: Comparison between experimental data (–) and simulation (■) for 11th and 12th December 2005 a) Φ_{COND} and b) P_{ELEC} c) compression ratio

4.2 Results for model combining the heat pump and the heating floor

The device specific and operating parameters of the heat pump are the same as in §4.1 except that in this case $T_{SCONDIN}$ is computed thanks to the heating floor model.
The device specific parameters for the heating floor (transmittances U_{F1}, U_{F2}, U_{HF}, U_I, capacity C_{FLOOR} and surface area A_{FLOOR}) are computed following the thermal properties of the material in the slab and in the insulation layer.
Here, one operating parameter is used: T_{INDOOR} equal to the experimental values.
The simulation has been performed for a period of time of several months, using the experimental on/off pattern of the heat pump. We obtain the same level of agreement as obtained for the heat pump model alone, as shown on figure 4 for 11th and 12th December 2005. We can remark that the temperature level of the water in the floor is in excellent agreement with the experimental one (figure 4), leading to the conclusion that the heating floor model is good.

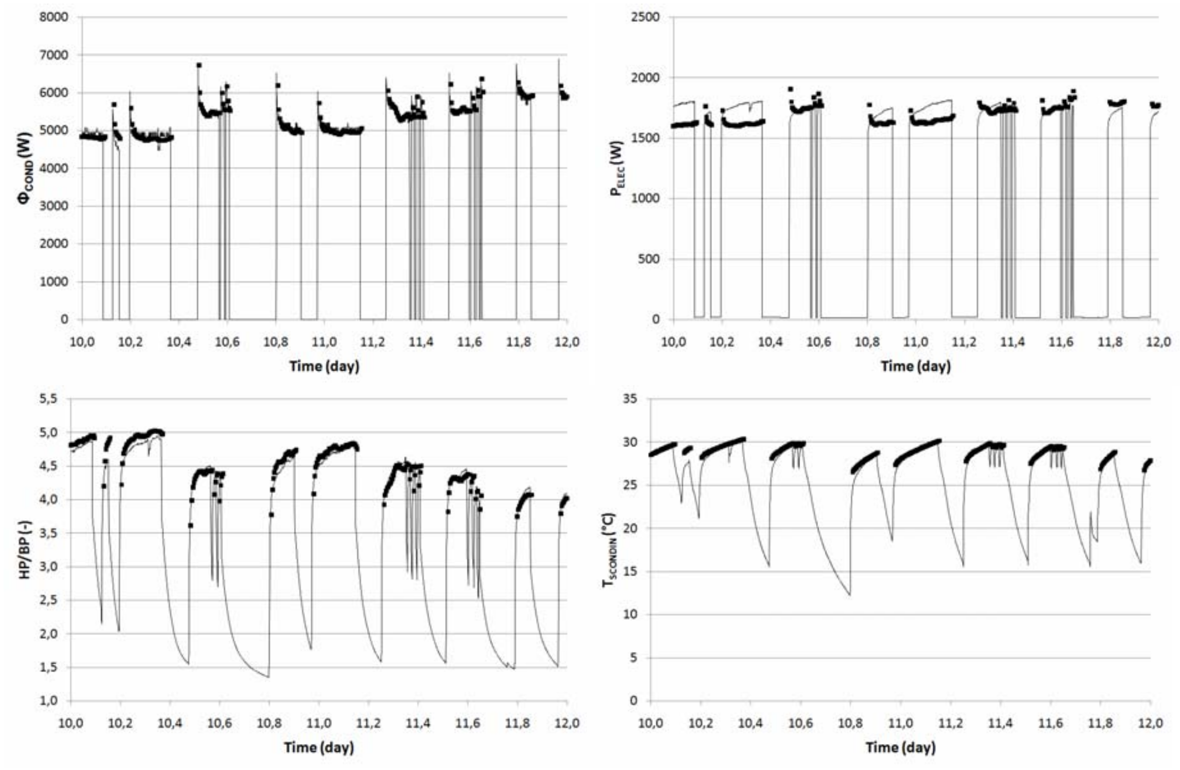

Figure 4: Comparison between experimental data (–) and simulation (■) for 11th and 12th December 2005 a) Φ_{COND} and b) P_{ELEC} c) compression ratio d) $T_{SCONDIN}$

4.3 Results for the whole system model

The device specific and operating parameters of the heat pump and of the heating floor are the same as in §4.2 except that in the present case T_{INDOOR} is computed using the building single zone model.
The device specific parameters for the building model (transmittances U, capacities C_{FLOOR} and surface areas A) are computed using the thermal properties of the materials and the geometrical data of the house.
The operating parameters are: $T_{OUTDOOR}$ equal to the experimental values, T_{GROUND} fixed at a constant value of 10°C, Φ_{SUN} computed using standard values for Belgium (Solar Energy Laboratory 2004), Φ_{INT} computed following a normative method (RT2000 2000) and Φ_{VENT} computed using experimental values.
The simulation has been performed for a period of time of one year, still using the experimental on/off pattern of the heat pump. There is a higher discrepancy between the simulated and experimental values, as shown on figure 5 for 11th and 12th December 2005. These discrepancies are due to the lower simulated indoor temperature: as the building model is a single zone model, the indoor simulated temperature is the average between the ground floor temperature (connected to the floor through the heating floor model) and the first floor temperature (experimentally, about 6°C lower than the ground floor temperature).
The lower indoor temperature implies that the condensation temperature in the heat pump is also lower, giving higher heat flow rates and lower electrical powers.

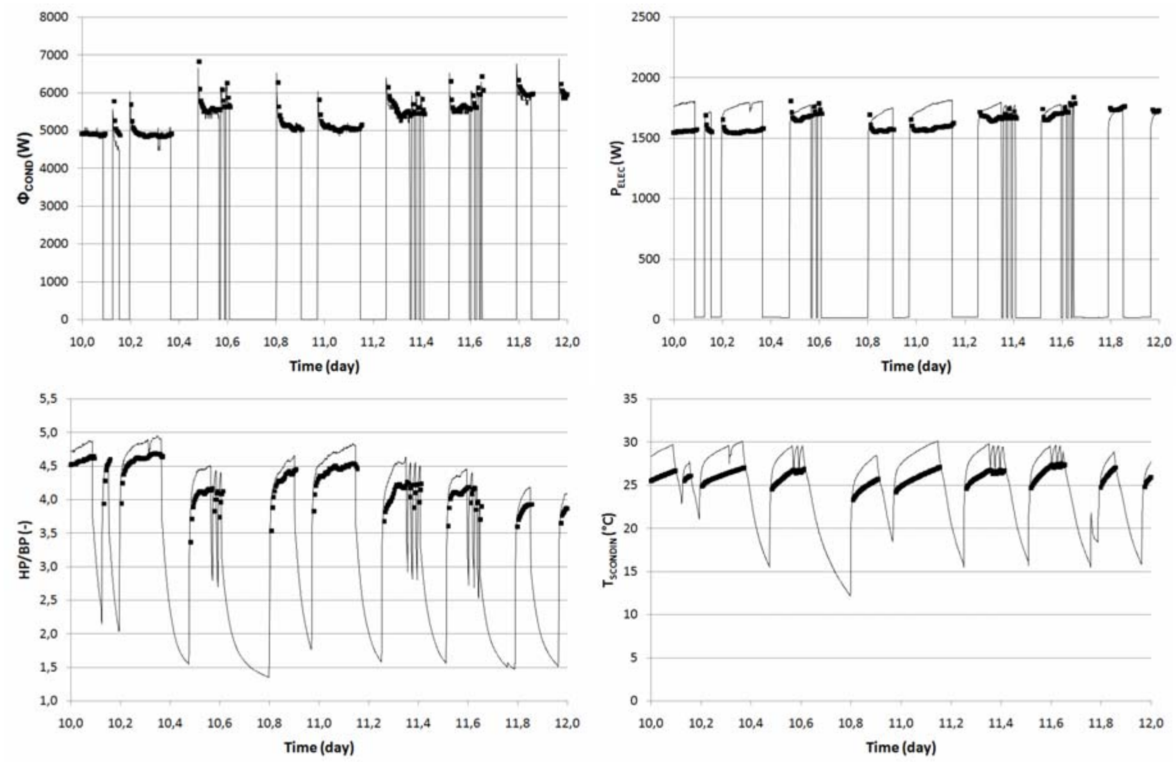

Figure 5: Comparison between experimental data (–) and simulation (■) for 11th and 12th December 2005 a) Φ_{COND} and b) P_{ELEC} c) compression ratio d) $T_{SCONDIN}$

The total discrepancy over one year is surprisingly quite low as shown in Table 2: heat released to the house during the whole year is 4.0% lower for the simulation compared to the experiment while electrical consumption is 4.5% lower for the same period.

Table 2: Heat delivered, electrical consumption and COP for one year, comparison between simulated and experimental values

Period	Q_{SIM} (kWh)	E_{SIM} (kWh)	COP_{SIM} (-)	Q_{EXP} (kWh)	E_{EXP} (kWh)	COP_{EXP} (-)
Dec 2005	1651	523	3.16	1715	553	3.10
Jan 2006	1754	570	3.08	1789	608	2.94
Feb 2006	1539	501	3.07	1561	519	3.01
Mar 2006	1539	439	3.10	1371	450	3.05
Apr 2006	713	217	3.29	757	220	3.44
May 2006	254	74	3.43	306	79	3.88
Jun 2006	65	21	3.14	75	21	3.63
Jul 2006	0	0	-	0	0	-
Aug 2006	60	16	3.65	74	17	4.26
Sep 2006	5	1	3.68	6	1	4.44
Oct 2006	242	64	3.82	279	66	4.23
Nov 2006	734	202	3.63	810	219	3.70
Year	8377	2628	3.19	8744	2753	3.18

5 CONCLUSION

The models developed in this project for the heat pump and for the heating floor give excellent simulation results compared to experimental ones. The thermal model of the house is not so good, because it is a single zone model, but the overall model combining the three

sub-models predicts correctly the annual heat delivered, electricity consumption and SPF of the heat pump. In the near future, several improvements are planned:
- development of a model for the static air heat exchanger;
- development of a two zone model for the thermal behavior of the building in order to obtain correct ground floor temperature values;
- development of a model for the control of the heat pump, following the needs of the house.

The simulations will also be compared with the simulations obtained with the TRNSYS models, for which we already have single zone and two zone building models (Diricq 2006, Ducobu 2007).

6 NOMENCLATURE

A_{COND}	condenser surface area	(m^2)
A_{EVAP}	evaporator surface area	(m^2)
A_{FLOOR}	heating floor surface area	(m^2)
A_{ROOF}	roof surface area	(m^2)
A_{WALL}	wall surface area	(m^2)
$A_{WINDOWS}$	windows surface area	(m^2)
C_{FLOOR}	floor thermal capacity	(J/K^{-1})
C_{INDOOR}	indoor house thermal capacity	(J/K^{-1})
c_{Pi}	specific heat of refrigerant at point i	($J/kg^{-1}K^{-1}$)
c_{PSEVAP}	specific heat of water-glycol blend in the evaporator	($J/kg^{-1}K^{-1}$)
c_{PSCOND}	specific heat of water in the condenser	($J/kg^{-1}K^{-1}$)
c_{Vi}	constant volume specific heat of refrigerant at point i	($J/kg^{-1}K^{-1}$)
C_{Wi}	wall thermal capacity at node i	(J/K^{-1})
COP	coefficient of performance of the heat pump	(-)
E	total energy consumption over a period of time	(kWh)
F_{CO}	condenser surface area fraction devoted to condensation	(-)
F_{DSH}	condenser surface area fraction devoted to desuperheating	(-)
F_{EV}	evaporator surface area fraction devoted to evaporation	(-)
h_i	specific enthalpy at point i	(J/kg)
$LMTD_{CO}$	log-mean temperature difference for condensation process	(°C)
$LMTD_{DSH}$	log-mean temperature difference for desuperheating process	(°C)
$LMTD_{EV}$	log-mean temperature difference for evaporation process	(°C)
$LMTD_{SC}$	log-mean temperature difference for subcooling process	(°C)
$LMTD_{SH}$	log-mean temperature difference for superheating process	(°C)
P_{COND}	condensation pressure	(MPa)
P_{ELEC}	electrical power of the compressor	(W)
P_{EVAP}	evaporation pressure	(MPa)
Q	total heat delivered over a period of time	(kWh)
q_{REF}	refrigerant mass flow rate	(kg/s)
q_{SCOND}	water mass flow rate in the condenser	(kg/s)
q_{SEVAP}	glycol-water blend mass flow rate in the evaporator	(kg/s)
q_V	swept volumetric flow rate of the compressor	(m^3/s)
SPF	seasonal performance factor of the heat pump	(-)
T_{COND}	refrigerant condensation temperature	(°C)
T_{EVAP}	refrigerant evaporation temperature	(°C)
T_{FLOOR}	floor temperature	(°C)
T_{GROUND}	ground temperature	(°C)
T_i	refrigerant temperature at point i	(°C)
T_{INDOOR}	indoor temperature	(°C)
$T_{OUTDOOR}$	outdoor temperature	(°C)
$T_{SCONDIN}$	water inlet temperature in the condenser	(°C)
$T_{SCONDOUT}$	water outlet temperature in the condenser	(°C)
$T_{SEVAPIN}$	glycol-water blend inlet temperature in the evaporator	(°C)

$T_{SEVAPOUT}$	glycol-water blend outlet temperature in the evaporator	(°C)
T_{WALLi}	wall temperature at node I	(°C)
U_{COND}	condenser heat transfer coefficient	($W/m^{-2}K^{-1}$)
U_{EVAP}	evaporator heat transfer coefficient	($W/m^{-2}K^{-1}$)
U_{FI}	floor-indoor thermal transmittance	($W/m^{-2}K^{-1}$)
U_{Fi}	floor thermal transmittance at node i	($W/m^{-2}K^{-1}$)
U_{HF}	tube-heating floor thermal transmittance	($W/m^{-2}K^{-1}$)
U_{I}	insulation layer thermal transmittance	($W/m^{-2}K^{-1}$)
U_{ROOF}	roof thermal transmittance	($W/m^{-2}K^{-1}$)
U_{WI}	wall-indoor thermal transmittance	($W/m^{-2}K^{-1}$)
U_{Wi}	wall thermal transmittance at node i	($W/m^{-2}K^{-1}$)
$U_{WINDOWS}$	windows thermal transmittance	($W/m^{-2}K^{-1}$)
U_{WO}	wall-outdoor thermal transmittance	($W/m^{-2}K^{-1}$)
ΔT_{SC}	subcooling at the condenser	(°C)
ΔT_{SH}	superheating at the evaporator	(°C)
ε	dead volume ratio of the compressor	(-)
η_{ELEC}	compressor pseudo-electrical efficiency	(-)
η_{ISOS}	compressor isentropic efficiency	(-)
η_{V}	compressor volumetric efficiency	(-)
ρ_{i}	refrigerant density at point i	(kg/m^{-3})
Φ_{COND}	heat flow rate at the condenser	(W)
Φ_{INT}	internal gains	(W)
Φ_{SUN}	solar gains	(W)
Φ_{VENT}	ventilation losses	(W)

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