PERFORMANCE MEASUREMENT AND MODELING OF AIR-SOURCE RESIDENTIAL HEAT PUMPS

Eric Dumont, PhD, Thermodynamics Division, Faculty of Engineering, Mons, Belgium
Marc Frère, Professor, Thermodynamics Division, Faculty of Engineering, Mons, Belgium

ABSTRACT

This paper presents the performance measurement and modeling of one-year monitoring of data of a static air-water residential heat pump used for space heating in a single-family house. Firstly, we present the method to obtain daily COP values from instantaneous measurements on the refrigerant and on the water. Secondly, annual energy consumption, seasonal COP and costs are presented for a one-year period. Third, measurements are analyzed in order to obtain a day-average thermodynamic cycle with constant characteristics parameters depending only of the heat pump specifications. Using this model, accurate values of daily heat demand and energy consumption can be obtained. This cycle is then used to define different models of decreasing complexity for the computation of the annual heat demand and energy consumption in function of evaporation and condensation temperature and running time of the heat pump. Analysis of these three last parameters over one year gives correlations between them and outdoor temperature. The results of the models with the correlations yields annual performance values (heat demand, energy consumption and seasonal COP) which differ less than 5 % from the measured values.

Key Words: ***heat pump, experimental measurements, seasonal COP, static air source, modeling.***

1 INTRODUCTION

Heat pumps are energy-efficient 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. From an energetic and environmental (CO_2 production) point of view, heat pump is the best choice. From an economical point of view, its interest depends on the COP values. These values are usually given by manufacturers for standard working conditions. The real values for a heat pump system (heat pump installed in a given house) for one heating season are therefore very different from the published ones. Moreover, the same machine installed in different houses will give different COP values, because of the interaction of the machine with house characterized by its own thermal features.

The scope of this project is the measurement and the modeling of the performance of heat pumps installed in single-family houses in Belgium. The project is devoted to a two-year monitoring of nine heat pumps used for space heating and two heat pumps used for sanitary water heating [Frère et al. 2004]. The measurements began in January 2001 and will be finished in December 2006. They are sponsored by Electrabel, the main electrical power producer and supplier in Belgium. To date, six surveys are finished, and this paper is devoted to the analysis and modeling of a static air-to-water heat pump.

2 PERFORMANCE MEASUREMENTS

2.1 Heat Pump and House Description

The heat pump investigated is a two compressors vapor compression static air-to-water heat pump and uses R407C as refrigerant. Evaporation takes place in a static evaporator (20 m^2 of finned tubes) and condensation in a plate heat exchanger. The evaporator is placed on the south outside wall of the house. The house has no electrical heaters. The heat pump mainly works with one compressor. For extreme weather conditions, when the outdoor temperature ($T_{OUTDOOR}$) is below -5°C, the second compressor switches on.

The source is the outdoor air, which is a “static” source and is known to vary from -5°C in winter to 20° in summer (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 floor has a high thermal inertia which allows storage of heat during the night.

2.2 Measurement Method

The most important value to be monitored is the COP, which is defined as the ratio of the heat flow released by the condenser (Φ_{COND}) and the electrical power used by the installation. For that purpose, the following measurements are performed (Fig. 1): evaporation and condensation pressures (P_2 and P_3), temperatures (T_2, T_3, T_6, T_9 and T_{10}), volumetric flows of refrigerant (q_{VREF}) and water (q_{VW}), and electrical power of the compressor (Po_{ELCOMP}) and of the auxiliary water pump ($Po_{EL\ AUX}$).

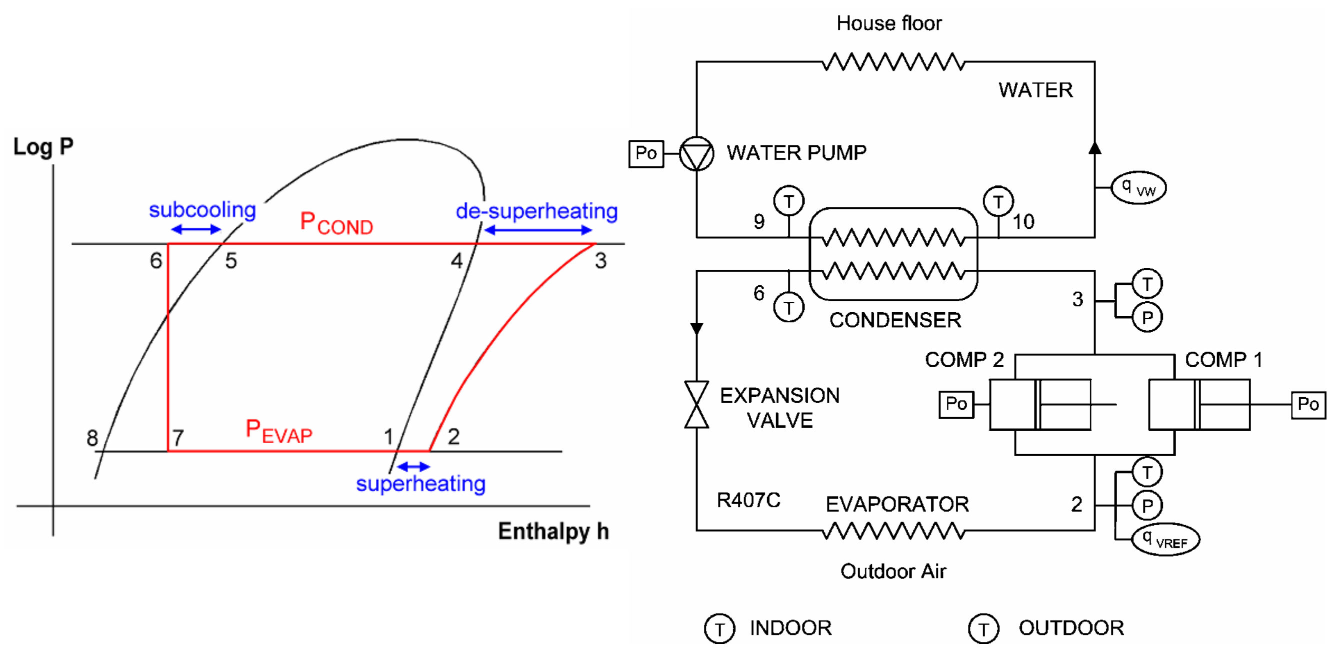

Fig. 1. Measurements performed on the heat pump.

Outdoor and indoor temperatures are also monitored ($T_{OUTDOOR}$ and T_{INDOOR}). All measurements are performed every second, then averaged over one minute and stored in a data logger for further analysis.

2.3 COP Calculation

The heat flow at the condenser (Φ_{COND}) is computed by balancing energy on the refrigerant side and on the water side, assuming steady-state conditions. The coefficients of performance are then calculated as:

$$\Phi_{COND} = q_{VREF}\ \rho(T_2, P_2)\ [h(T_3, P_3) - h(T_6, P_3)] = q_{VW}\ c_{PW}\ (T_{10} - T_9) \tag{1}$$

$$COP_{COMP} = \Phi_{COND} / Po_{ELCOMP} \tag{2}$$

$$COP_{HP} = \Phi_{COND} / (Po_{ELCOMP} + Po_{EL\ AUX}) \tag{3}$$

Instantaneous COP can be calculated considering the compressor consumption only (COP_{COMP}) or considering the compressor and auxiliary pump consumptions (COP_{HP}). Further calculations are performed to obtain the total heat supplied by the heat pump over one day (Q_{DAY}), the total electrical consumption of the heat pump (compressor and water pump) (E_{DAY}) and the average COP over the day (COP_{DAY}). The sum of all daily values for one heating season (September to May) yields annual heat (Q_{YEAR}), electrical consumption (E_{YEAR}) and seasonal COP (SCOP) values.

3 RESULTS

3.1 Daily Performance Results

COP values measured over a 14-month period are shown on Fig. 2. Each point is a daily COP value (COP_{DAY}). During this period, T_{EVAP} varies from -23 °C to -5 °C and T_{COND} from 35 °C to 44 °C.

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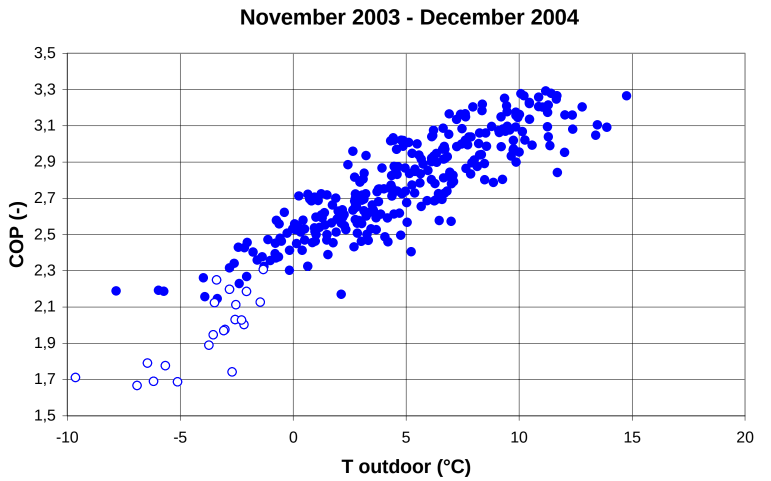

Fig. 2. Daily COP values (●: one compressor; ○: two compressors).

3.2 Annual Performance Results

Results for one year are presented in Table 1. 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: 0.16 Eur/kWh (peak) and 0.08 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 (0.043 Eur/kWh for fuel oil, 0.035 Eur/kWh for natural gas).

Table 1. Measured annual performance.

Period	**Nov 2003 - Oct 2004**
$\mathbf{E_{YEAR}}$ (kWh)	6453
$\mathbf{Q_{YEAR}}$ (kWh)	16737
SCOP (-)	2.59
Off-peak perc. (%)	78
Cost HP (Eur)	630
Cost Gas (Eur)	651
Cost Fuel oil (Eur)	801
Cost Elec (Eur)	1634

4 MODELING RESULTS

4.1 Cycle Modeling

For a given heat pump system, the thermodynamic cycle is well-defined (Fig. 1). Its characteristics may be determined from the measurements:

$$P_{EVAP} = P_2 \tag{4}$$

$$P_{COND} = P_3 \tag{5}$$

$$\Delta T_{SH} = T_2 - T_{DEW}(P_2) \tag{6}$$

$$\Delta T_{SC} = T_{BUBBLE}(P_3) - T_6 \tag{7}$$

$$\eta_{ISOS} = [h_{3ISOS}(s_2, P_3) - h_2(T_2, P_2)]/[h_3(T_3, P_3) - h_2(T_2, P_2)] \tag{8}$$

$$\eta_{EL} = Po_{MEC\ COMP} / Po_{EL\ COMP} = [q_{VREF}\ \rho_2(T_2, P_2)\ [h_3(T_3, P_3) - h_2(T_2, P_2)]]/Po_{EL\ COMP} \tag{9}$$

Knowing these parameters (P_{EVAP}, P_{COND}, ΔT_{SH}, ΔT_{SC}, η_{ISOS}, η_{EL}), it is possible to calculate the cycle of the heat pump:

$$h_2 = h(T_{DEW}(P_{EVAP}) + \Delta T_{SH}, P_{EVAP}) \tag{10}$$

$$s_2 = s(T_{DEW}(P_{EVAP}) + \Delta T_{SH}, P_{EVAP}) \tag{11}$$

$$\rho_2 = \rho(T_{DEW}(P_{EVAP}) + \Delta T_{SH}, P_{EVAP}) \tag{12}$$

$$h_{3ISOS} = h(s_2, P_{COND}) \tag{13}$$

$$h_3 = h_2 + (h_{3ISOS} - h_2)/\eta_{ISOS} \tag{14}$$

$$h_6 = h(T_{BUBBLE}(P_{COND}) - \Delta T_{SC}, P_{COND}) \tag{15}$$

4.2 Daily Performance Modeling

Using the previous approach, Q_{DAY} and E_{DAY} may be computed. They depend on four sets of parameters:

- cycle parameters (ΔT_{SH}, ΔT_{SC}, η_{ISOS}, η_{EL}) (set1)
- source-related parameters (P_{EVAP}, P_{COND}) (set 2)
- compressor characteristics (q_{VREF}) (set 3)
- running time of the heat pump over one day (τ_{DAY}) (set 4)

Sets 1 to 3 are averaged to define a "day-average" cycle and "day-average" parameters. Q_{DAY} and E_{DAY} are then calculated by:

$$Q_{DAY} = \tau_{DAY}\, q_{VREF}\, \rho_2\, (h_6 - h_3) \tag{16}$$

$$E_{DAY} = \tau_{DAY}\, [q_{VREF}\, \rho_2\, (h_3 - h_2)]/\eta_{EL} \tag{17}$$

The day-average cycle replaces all the running cycles of the heat pump. Its duration is the sum of all cycle durations over one day. Q_{DAY} and E_{DAY} values computed with "day-average" parameters and measured τ_{DAY} values (equations 16 and 17) are compared with measured ones (figures 3 and 4).

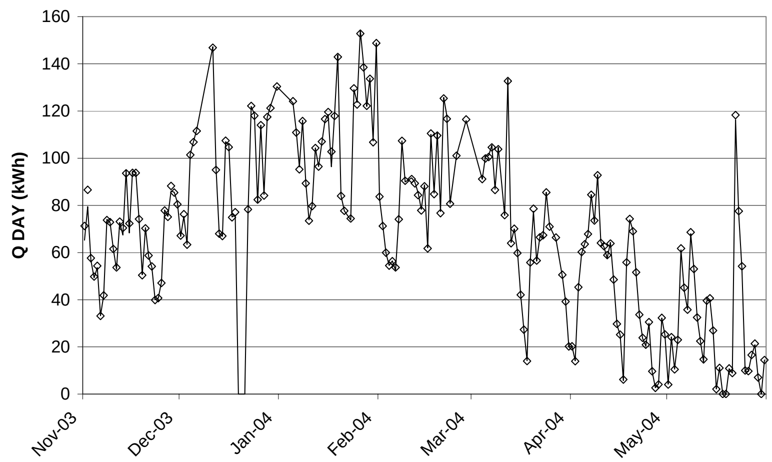

Fig. 3 Q_{DAY}: measurements (◊) and computed values with day-average cycle (-)

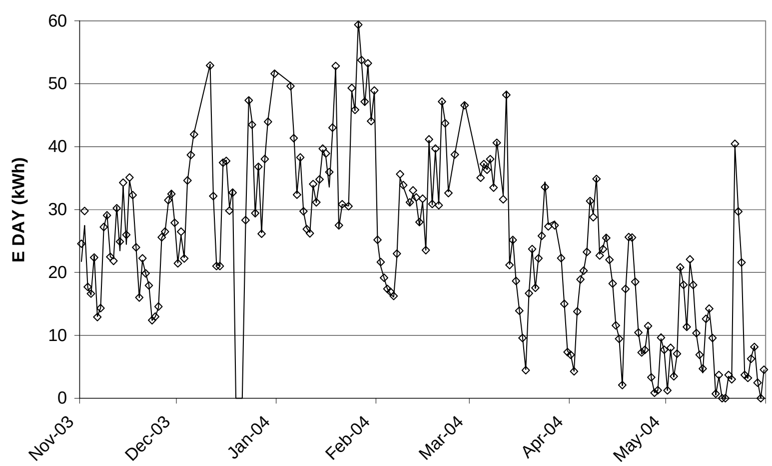

Fig. 4. E_{DAY}: measurements (◊) and computed values with day-average cycle (-).

The average error on Q_{DAY} is 1.1% and on E_{DAY} IT is 1.6%. This method is then reliable in order to obtain day-average quantities.

4.3 Annual Performance Modeling

Cycle parameters (set 1) change slowly with time and are quite constant during a whole year. At the opposite, P_{EVAP} and P_{COND} (set 2) may vary in a wide range because they depend on heat transfer mechanisms and are related to outdoor and floor temperatures.

Day-average cycle parameters (set 1) for each day of a whole year can be computed by three methods:

- use measured day-average cycle parameters (as used above in § 4.2) (model 1);
- correlate measured day-average cycle parameters with P_{EVAP} over one year and use the correlation (model 2). It is not useful to correlate these parameters with P_{COND} because it has few variations in comparison with P_{EVAP};
- average measured cycle parameters over one year and use them as constant day-average cycle parameters (model 3).

The three methods, used with measured day-average P_{EVAP}, P_{COND} and q_{VREF} and measured τ_{DAY} were used to compute Q_{YEAR}, E_{YEAR} and SCOP for the period November 2003 to May 2004. They were then compared with measured values (Table 2).

Table 2. Annual performance – measurements and models.

	Q_{YEAR} (kWh)	Error Q_{YEAR} (%)	E_{YEAR} (kWh)	Error E_{YEAR} (%)	SCOP (-)	Error SCOP (%)
Meas.	13092	/	4683	/	2.80	/
Model 1	13061	-0.2	4671	-0.3	2.80	0.0
Model 2	12968	-1.0	4713	+0.6	2.75	-1.6
Model 3	12910	-1.4	4588	-2.0	2.81	+0.7

4.4 Modeling of P_{EVAP}, P_{COND}, q_{VREF} and τ_{DAY}

Table 2 shows that using constant day-average cycle parameters over one year (model 3) for computation of annual performance gives accurate predicted results for Q_{YEAR}, E_{YEAR} and SCOP. Unfortunately, a serious drawback is that this computation needs measured daily values for P_{EVAP}, P_{COND}, q_{VREF} and τ_{DAY}.

The refrigerant volumetric flow q_{VREF} can be obtained from compressor characteristics. It depends on the compression ratio:

$$q_{VREF} = q_{VREF}\,(P_{COND}/P_{EVAP}) \tag{18}$$

The annual amount of heat Q_{YEAR}, and therefore τ_{DAY} depend on the thermal losses of the house and can be evaluated efficiently with thermal building software like TRNSYS [Sautier et al. 2005]. As a consequence, τ_{DAY} should be correlated to $T_{OUTDOOR}$.

P_{EVAP} (related to T_{EVAP}) and P_{COND} (related to T_{COND}) are correlated to the heat transfer in the evaporator and in the house floor respectively. T_{EVAP} and T_{COND} should be available from $T_{OUTDOOR}$ and T_{FLOOR}.

We then tried a simple model, which correlates P_{EVAP} and τ_{DAY} with independent, easy-to-find parameters, namely $T_{OUTDOOR}$ (Figures 5 and 6). It appears that P_{COND} was well-correlated with P_{EVAP} because the temperatures of the source ($T_{OUTDOOR}$) and of the sink (T_{FLOOR}) are related via the running time of the heat pump: when $T_{OUTDOOR}$ is low, the demand for heat demand is high and the floor becomes warmer.

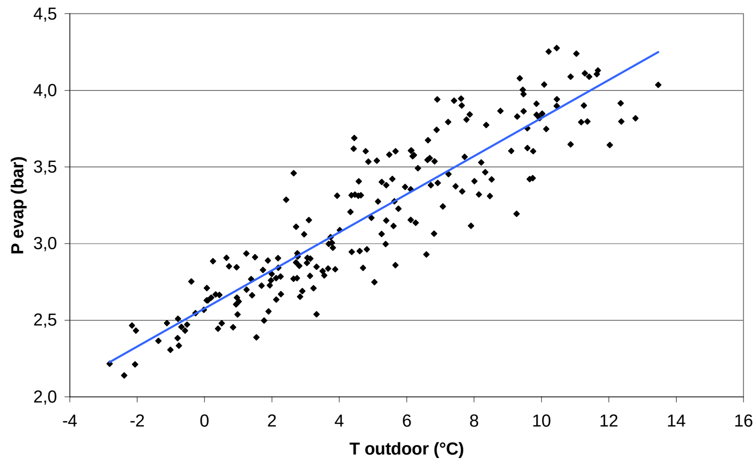

Fig. 5. Correlation between P_{EVAP} and $T_{OUTDOOR}$ for one year measurements.

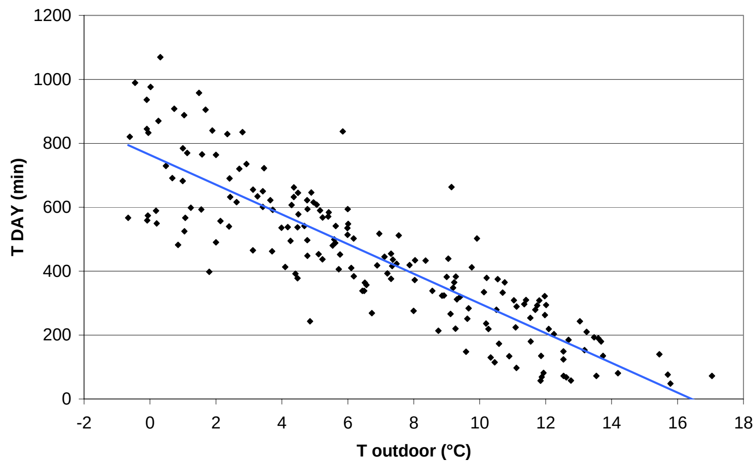

Fig. 6. Correlation between τ_{DAY} and $T_{OUTDOOR}$ for one year measurements.

With these correlations and equation 18, we were able to compute all day-average values using constant day-average cycle parameters defined in model 3 (§ 4.3). The results of this computation (model 4) are given in Figures 7 and 8 for daily performance and in Table 3 for annual performance.

Table 3. Annual performance – measurements and model 4.

	Q_{YEAR} (kWh)	Error Q_{YEAR} (%)	E_{YEAR} (kWh)	Error E_{YEAR} (%)	SCOP (-)	Error SCOP (%)
Meas.	13092	/	4683	/	2.80	/
Model 4	13741	+5.0	4887	+4.3	2.81	+0.6

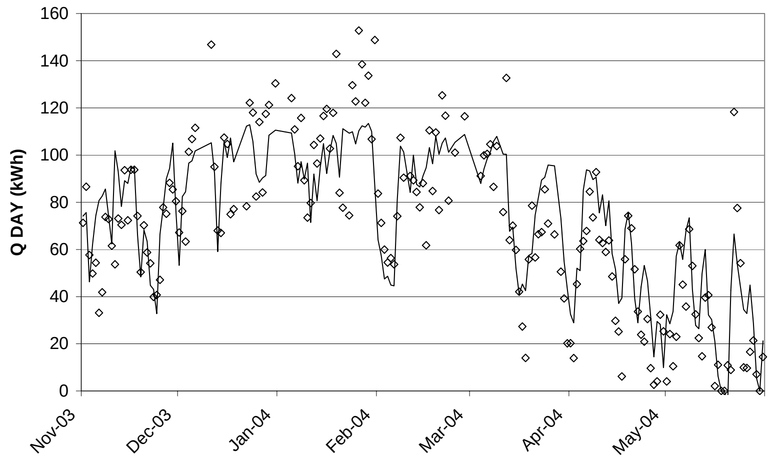

Figure 7: Q_{DAY}: measurements (◊) and computed values with model 4 (-).

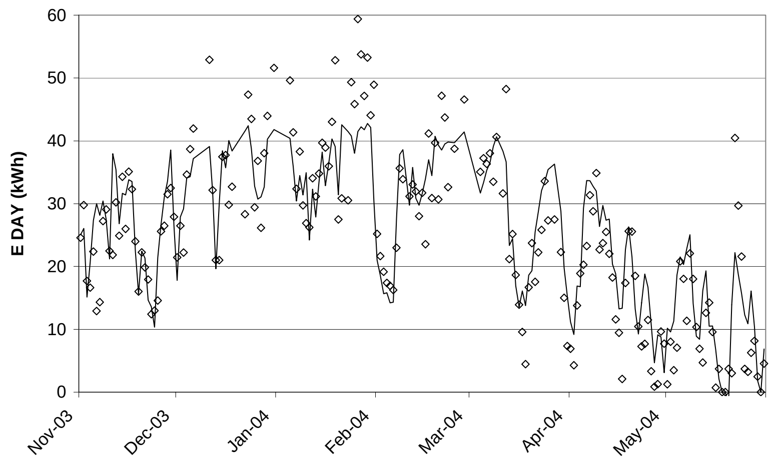

Fig. 8. E_{DAY}: measurements (◊) and computed values with model 4 (-).

Model 4 gives poor results for E_{DAY} and Q_{DAY} but gives quite accurate results for a whole year (Table 3). The accuracy can be explained by the effect of averaging many daily values with different conditions happening within one year. The computed values of Q_{YEAR} and E_{YEAR} are overestimated by 5 %. This is due to overestimation of q_{VREF} by equation 18. This model can then be useful for quick evaluation of SCOP and energy consumption over one year.

In order to compute annual performance (Q_{YEAR}, E_{YEAR} and SCOP), we need the four sets of parameters defined above. The first and third sets can be obtained by a few measurements on the heat pump itself or by processing manufacturer data. Moreover, the first set of parameters can be assumed constant without loss of accuracy on the results (model 3 § 4.3). The second and fourth sets can be obtained with two methods. The first one assumes all thermal characteristics of the heat exchangers and of the building known and models all thermal behaviours. This is not easy because of the complex modelling and of the usually partial information about the systems. The second method uses correlations (like those determined above) but needs extensive measurements to obtain these correlations.

5 CONCLUSIONS

The results of the monitoring of the heat pump gave a SCOP value of 2.59. The analysis of these results showed that we can define a “day-average” cycle characterized by “day-average” parameters constant over a whole year. In order to be able to compute annual performance values, the running time of the heat pump for each day as well as evaporation and condensation temperatures must be known. These values can be quickly computed by correlating them with the average outdoor temperature. Computed values for Q_{YEAR}, E_{YEAR} and SCOP gave results overestimated by 5% with measured values.

6 NOMENCLATURE

COP	instantaneous coefficient of performance (-)
COP_{COMP}	instantaneous coefficient of performance considering compressor consumption only (-)
COP_{HP}	instantaneous coefficient of performance considering compressor and water pump consumptions (-)
COP_{DAY}	daily coefficient of performance (-)
c_{PW}	heat capacity of water (J kg^{-1} K^{-1})
E_{DAY}	energy used by the heat pump over one day (kWh)
E_{YEAR}	energy used by the heat pump over one year (kWh)
h	specific enthalpy of the refrigerant (J kg^{-1})
P	pressure (bar)
P_{COND}	condensation pressure (bar)
P_{EVAP}	evaporation pressure (bar)
$Po_{EL\ AUX}$	electrical power of the auxiliary water pump (W)
$Po_{EL\ COMP}$	electrical power of the compressor (W)
$Po_{MEC\ COMP}$	mechanical power received by the refrigerant in the compressor (W)
Q_{DAY}	heat supplied by the heat pump over one day (kWh)
Q_{YEAR}	heat supplied by the heat pump over one year (kWh)
q_{VREF}	volumetric flow of refrigerant in the heat pump (m^3 s^{-1})
q_{VW}	volumetric flow of water in the heat pump (m^3 s^{-1})
s	specific entropy of the refrigerant (J kg^{-1} K^{-1})
SCOP	seasonal coefficient of performance (-)
T	temperature (°C)
T_{BUBBLE}	bubble temperature (°C)
T_{DEW}	dew temperature (°C)
T_{FLOOR}	temperature of the heating floor in the house (°C)
T_{INDOOR}	indoor temperature of the house (°C)
$T_{OUTDOOR}$	outdoor temperature (°C)
ΔT_{SC}	condenser subcooling (°C)
ΔT_{SH}	evaporator superheating (°C)
η_{ISOS}	isentropic efficiency of the compressor (-)
η_{EL}	electric efficiency of the compressor (-)
ρ	density of the refrigerant in the heat pump (kg m^{-3})
τ_{DAY}	running time of the heat pump over one day (min)
Φ_{COND}	heat flow released at the condenser (W)

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