PERFORMANCE OF AN AIR-TO-WATER HEAT PUMP SYSTEM IN A LOW ENERGY RESIDENTIAL BUILDING: MONITORING RESULTS AND MODELING

C. Verhelst, H. Nolens, K. Schoovaerts, L. Helsen Professor, KULeuven, Mechanical Engineering Department, Leuven, Belgium

Abstract: A modulating air-to-water heat pump system, installed in a Belgian low-energy residential building, has been monitored from December 2006 to December 2007. This paper presents the system performance derived from the monitored data and the development of a heat pump model.
The heat pump is connected to a floor heating system and to a domestic hot water (DHW) tank. For the monitored heating seasons, with average outdoor temperatures of respectively 5.7°C, the average COP of the heat pump in building heating mode was 4.1 and in DHW production mode 2.4. The seasonal performance factor (SPF) of the whole system during the heating season (including building heating mode, DHW mode and transmission losses of the DHW tank) amounted to 3.3.
The building and heating system are modeled in TRNsys. A new heat pump model is developed to take the capacity modulation into account. A second order polynomial, with outdoor temperature, condenser outlet water temperature and instantaneous building heating demand as independent variables, is found to give the best fit.

Key Words: ***residential heat pump system, monitoring, modeling***

1 INTRODUCTION

In residential buildings, energy is used to improve thermal and visual comfort, to realize an acceptable indoor air quality, to produce domestic hot water and for electrical appliances. CO_2-emissions related to this energy use amount to almost 20% of the total Flemish CO_2-emissions. Heating is responsible for 50 to 75% of this amount (Flemish Authorities 2007). The correct use of efficient heating installations, such as heat pumps, can contribute to the reduction of primary energy consumption.
To achieve optimal energetic performance, the heat pump should be combined with a low-temperature heating system such as floor heating. The latter restricts the maximal heating power to 50 W/m², which means that heat pump systems perform best in low-energy dwellings. The lower the heating demand of the building, the lower the water supply temperature needed, the higher the coefficient of performance (COP). For residential buildings this decrease in building heating demand leads to a higher share of the DHW production in the total energy demand. Current heat pump technologies are capable of producing hot water at a temperature of 50°C to 55°C. The higher condenser outlet temperature is, however, realized with a lower COP.
The aim of this study is to assess the SPF of a heat pump system in real working conditions. A modulating air-to-water heat pump system for both building heating and DHW production, installed in an occupied single-family house, has been monitored during one heating season. In this paper the performance of the system is analyzed and the related CO_2-emission is compared to that of a condensing boiler.

The studied heat pump has a frequency inverter controlled compressor for capacity modulation. This modulation technique positively affects the performance and the service life of the heat pump since it reduces on-off cycling of the heat pump and the related transient losses, improving the part load characteristics. The second part of the paper focuses on the modeling of the heating system with special focus on the inverter-controlled modulating heat pump.

2 SYSTEM DESCRIPTION

2.1 Building

The heat pump system studied is installed in a low energy single-family house in Belgium with 4 inhabitants. The building has a total volume of 942 m³ and a total floor surface of 602 m². The average UA-value amounts to 0.35 W/m²K and the air-tightness of the building, represented by the n_{50}-value, is 2.5 h^{-1}. There is a CO_2-controlled ventilation system without heat recuperation.

2.2 Heat pump system

The heat pump system is represented in figure 1. The air-to-water heat pump has a variable-speed swing compressor. The heat pump is connected to the floor heating system and – in parallel – to a DHW tank. To satisfy peak heating demand there is an electrical backup heater. Furthermore, a booster heater is installed in the DHW tank to enable DHW heating at temperatures above 50°C. The system catalogue data are given in Table 1.

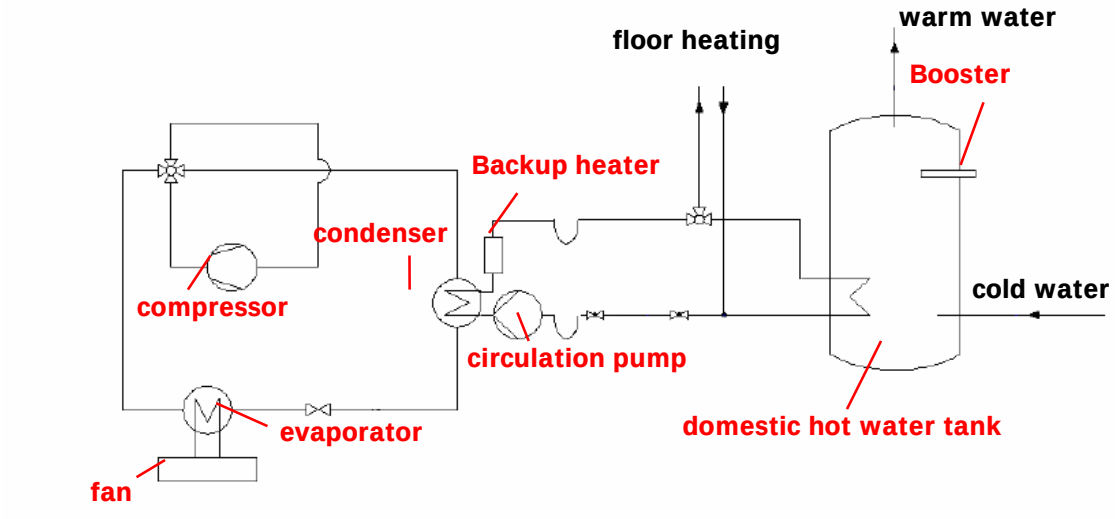


Figure 1: Scheme of heat pump system

Table 1: Heat pump system characteristics

Component	Parameter		Units	Value
Heat pump	Heating capacity	Minimum	kW	4.36
		Nominal(*)	kW	8.43
		Maximum	kW	9.58
	Outdoor unit power	Nominal(*)	kW	2.08
	COP	Nominal(*)	-	4.05
Backup heater	Power		kW	2 x 4.5
Booster heater	Power		kW	3
DHW tank	Volume		liter	300
Circulation pump	Power		W	130
	Flow rate		liter/min	16.5

(*)T_{out} dry bulb/ wet bulb 7°C/ 6°C – T_c 35°C (ΔT = 5°C)

2.3 System control

The scheme of the system control is presented in Figure 2. The floor heating system is controlled by a central room thermostat. The set point for the water supply temperature is set by a weather compensated supply temperature control. The set point supply temperature is 35°C for outdoor temperatures below -8°C and linearly decreases to 25°C for outdoor temperatures of 12°C and higher. The outdoor temperature T_{out}, the water supply temperature T_s and the compressor frequency f determine the COP of the heat pump (Shao et al 2004). The controller determines the compressor's power P by modulating the compressor's frequency as a function of the difference between actual water supply temperature T_s and set point T_s*.

DHW demand has priority to heating. The DHW tank is heated during night. After a given time interval (which can be changed from 30 up to 90 minutes), the heat pump switches back to building heating mode to prevent the house from cooling down too much. Then the booster heater takes over the DHW production. Defrost is realized by reversing the heat pump from heating to cooling mode. Only when the outdoor temperature is below 0°C, the backup heater is used.

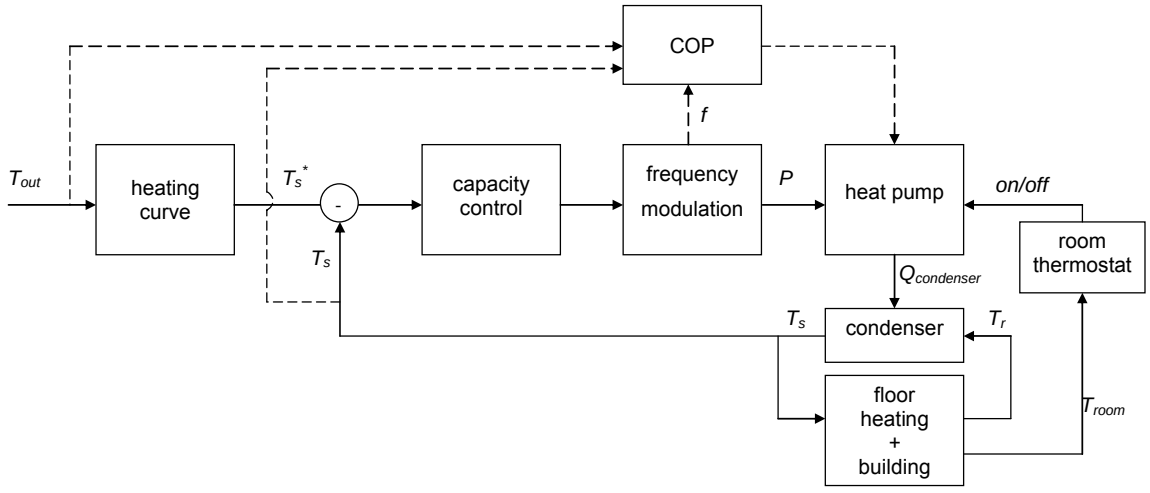


Figure 2: Control scheme for heat pump in residential heating mode (solid lines) and physical dependency (dotted lines)

3 MONITORING PROGRAM

3.1 Measured data

The aim of the monitoring program is to assess the global system performance in an occupied low energy dwelling.
Figure 3 and Table 2 show the measured variables of the heat pump system.

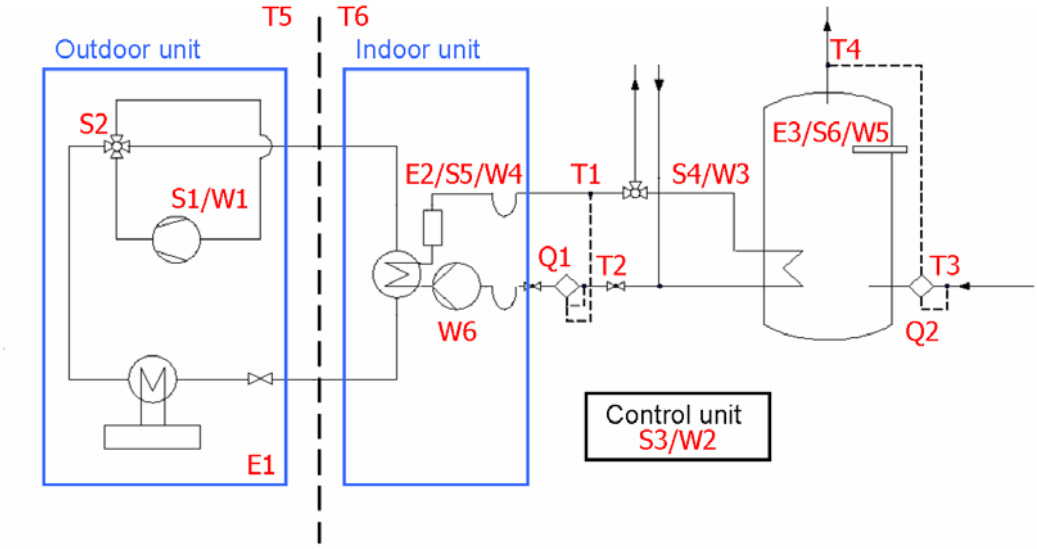


Figure 3: Scheme of the measured variables

Table 2: Meaning of the measured variables

Measured quantity	Symbol	Description
Energy [kWh]	Q1	Condenser energy
	E1	Electricity consumption outdoor unit
	Q2	Total energy domestic hot water production
	E2	Electricity consumption backup heater
	E3	Electricity consumption booster heater
Temperature [°C]	T1	Supply water temperature
	T2	Retour water temperature
	T3	Domestic hot water tank inlet temperature
	T4	Domestic hot water tank outlet temperature
	T5	Outdoor temperature
	T6	Indoor temperature
Status [# switches/day]	S1	Compressor
	S3	Defrosting
	S4	Domestic hot water
	S5	Backup heater
	S6	Booster heater
Working time [%/interval]	W1	Compressor
	W2	Defrosting
	W3	Domestic hot water
	W4	Backup heater
	W5	Booster heater
	W6	Circulation pump

Water temperatures and energy quantities are monitored with a multical energy meter, with a standard sampling time of 15 minutes. The measurement accuracy is respectively ± 0.5 °C and ± 0.5 kWh. Air temperatures are monitored with a HOBO H8 temperature logger with an accuracy of ± 0.7°C. Electrical power consumption is measured by a Kamstrup 382 meter and is read out with a resolution of 1kWh. For the development of the heat pump model, the sampling frequency was increased to 1 per minute from 19 to 29 March. This was needed in order to increase the accuracy of the thermal and electrical power, derived from the measured cumulated energy values.

3.3 Monitoring results

3.3.1 Energy flows

Figure 4 presents the Sankey energy diagram which visualizes the energy flows in the system. The values give the accumulated energy during the entire monitoring period (4 December 2006 – 30 December 2007).

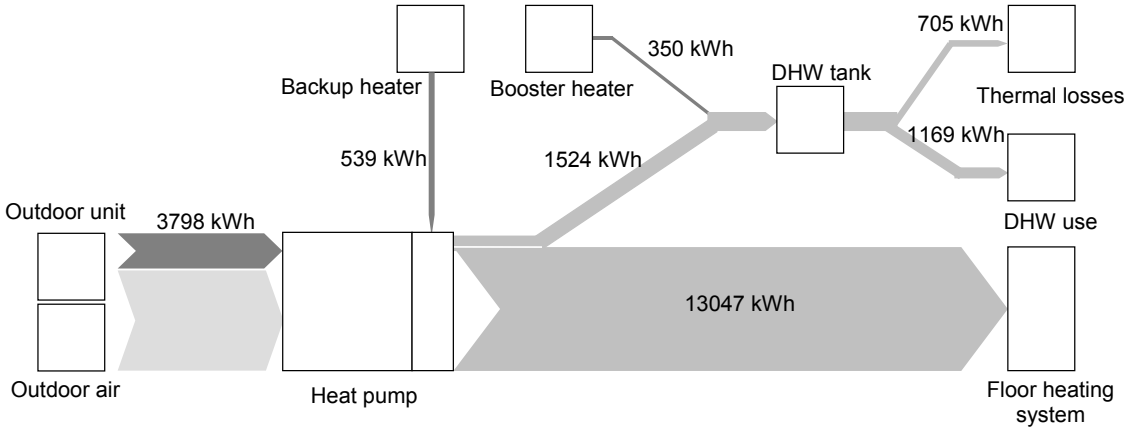


Figure 4: Sankey energy diagram of heat pump system (electrical energy in dark grey, thermal energy in light grey)

The measurements show that on yearly basis 87% of the total heat produced is used for building heating and 13% for the production of DHW. The backup heater and the booster heater provide respectively 4% and 2% of the total heat demand. Their share in the total electricity consumption, however, is 18%. The booster heater has a share of 19% in the production of DHW. Transmission losses of the DHW tank amount to 38%.

3.3.2 System performance

In order to assess the system performance, the COP and the SPF of the system are calculated based on the monitored data. The data corresponding to the heating season are analyzed on a weekly basis in order to observe the effect of the outdoor temperature. The data related to supply temperatures above 40°C are allocated to the production of DHW. The PF, which gives the average performance on a weekly basis, is calculated for building heating mode (PF_{BH}) and DHW mode (PF_{DHW}) separately and for the combination of both (PF_{sys}). $PF_{sys,G}$ stands for the heat generation efficiency and $PF_{sys,U}$ stands for the heat utilization efficiency. They differ in the way DHW tank thermal losses are handled. Table 3 shows the results for the heating season and the summer. Following equations are used (notations from Table 2):

$$COP_{BH} = \frac{Q_{c,BH}}{E_{1,BH}} = \frac{Q_{1,BH} - E_{2,BH}}{E_{1,\ BH}} \qquad COP_{DHW} = \frac{Q_{c,DHW}}{E_{1,DHW}} = \frac{Q_{1,DHW} - E_{2,DHW}}{E_{1,DHW}}$$

$$PF_{BH} = \frac{Q_{1,BH}}{E_{1,BH} + E_{2,BH}} \qquad PF_{DHW} = \frac{Q_{1,DHW} + E_3}{E_{1,DHW} + E_{2,DHW} + E_3}$$

$$PF_{sys,G} = \frac{Q_1 + E_3}{E_1 + E_2 + E_3} \qquad PF_{sys,U} = \frac{Q_{1,BH} + Q_2}{E_1 + E_2 + E_3}$$

Table 3: Heat pump system performance (4 December 2006 - 30 December 2007)

Period	$T_{out,average}$	COP_{BH}	COP_{DHW}	PF_{BH}	PF_{DHW}	$PF_{sys,G}$	PF_{sysU}
Heating season	**5.7**	**4.1**	**2.4**	**3.6**	**1.6**	**3.3**	**3.25**
04.12 - 10.12	9.1	5.1	2.4	5.1	1.4	4.4	4.3
11.12 - 17.12	10.1	4.9	2.5	4.8	1.5	4.4	4.3
18.12 - 24.12	3.4	4.3	2.1	3.6	1.3	3.4	3.3
24.12 - 30.12	3.2	4.0	2.4	3.2	1.4	3.0	2.9
31.12 – 06.01	7.8	5.0	3.1	5.0	1.5	4.3	4.2
07.01 – 13.01	9.5	4.9	2.1	4.9	1.3	4.1	3.9
14.01 – 20.01	8.4	4.8	2.1	4.8	1.4	4.1	4.0
21.01 – 27.01	0.7	3.4	2.3	2.8	1.4	2.7	2.6
28.01 – 03.02	7.2	4.8	2.9	4.8	1.5	4.2	4.1
04.02 – 10.02	2.5	3.9	2.1	3.3	1.4	3.1	3.0
11.02 – 17.02	6.8	4.7	1.6	4.7	1.2	4.1	3.9
18.02 – 24.02	7.6	4.8	2.4	4.8	1.6	4.2	4.1
25.02 – 03.03	7.3	5.1	2.5	5.1	1.9	4.4	4.3
04.03 – 10.03	7.5	5.0	2.5	4.8	1.7	4.1	4.0
11.03 – 17.03	7.7	4.8	2.2	4.8	1.9	3.6	3.5
18.03 – 24.03	3.7	4.5	2.3	4.2	1.4	3.8	3.7
25.03 – 31.03	9.6	3.0	2.5	3.0	2.3	2.6	2.5
22.10 - 28.10	10.5	4.1	1.9	3.8	1.5	3.3	3.2
29.10 - 04.11	7.8	4.5	2.8	4.4	1.9	3.6	3.3
05.11 - 11.11	10.2	4.4	2.4	4.2	1.7	3.7	3.6
12.11 - 18.11	9.0	3.3	2.1	2.8	1.6	2.7	2.6
19.11 - 25.11	3.3	3.8	3.2	3.5	1.9	3.3	3.3
26.11 - 02.12	7.0	3.9	2.4	3.5	1.7	3.3	3.2
03.12 - 09.12	7.2	4.3	2.7	4.2	2.0	3.8	3.6
10.12 - 16.12	8.8	3.4	2.0	2.9	1.6	2.8	2.8
17.12 - 23.12	2.9	2.6	2.4	2.1	1.7	2.0	2.0
24.12 - 30.12	-2.0	2.9	2.5	2.3	1.8	2.3	2.2
Summer season 01.04 – 21.10	**15.7**	**0.3**	**2.4**	**0.3**	**2.4**	**2.0**	**1.2**

Figure 5 shows the performance factors in function of the outdoor air temperature. The PF of building heating is strongly related to the outdoor temperature. Below 6°C the PF significantly drops because of the need for defrosting. Assuming a linear relationship between PF_{BH} and T_{out} (and thus neglecting the discontinuity caused by defrosting below 6°C), the PF for the monitored outdoor temperature interval (-2°C – 10°C) can be approximated by $PF_{BH} = 0.2\ T_{out} + 2.6$ (with T_{out} in °C). For the specified outdoor temperature range this linear regression has a R^2 of 0.5, which indicates this relationship is only a rough approximation. The PF_{DHW} shows only a small dependency on the outdoor temperature.

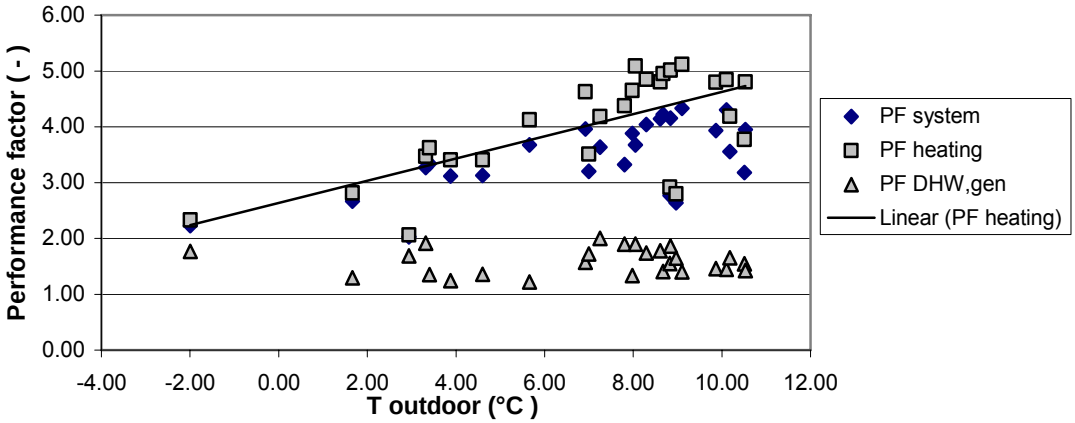


Figure 5: Calculated performance data in function of outdoor temperature (dots) and linear regression for the temperature dependency of PF_{BH} (solid line)

3.4 Performance analysis

The average COP_{BH} and SPF_{BH} of the analyzed heating season, respectively 4.1 and 3.6, are very satisfactory for an air-to-water heat pump. The difference between both is mainly due to defrosting. The COP_{DHW} is 2.4, which is lower than COP_{BH} because of the requested condenser water outlet temperature of 50°C. The difference between COP_{DHW} and PF_{DHW} is due to the use of the booster heater. The difference between $PF_{sys,G}$ and $PF_{sys,U}$, respectively 3.3 and 3.25, is caused by the thermal losses of the DHW tank. The difference between PF_{BH} and PF_{sys} shows the impact of the DHW production performance. The measurements reveal that the booster heater is not used during summer. The DHW is then produced by the heat pump only.

In order to improve the system performance in the studied case, better insulation of the DHW tank is recommended, as well as a reduction of the share of the booster heater during the heating season. The latter can be achieved by modifying some control settings for the DHW production. The current settings are such that the heat pump can work maximally 30 minutes in DHW mode. Figure 6 shows that this restriction causes the booster heater to take over (see periodical peak in $T_{c,out}$ at 3h). Increasing the time allowed in DHW mode would decrease the share of the booster heater and so improve the system performance. The effect of an increased time interval on the indoor air temperature is studied in section 5.1.

Figure 6 also shows the operation of the heat pump in defrost mode. Defrosting is realized by reversing the heat pump operation mode during 3 minutes. Heat is extracted from the water supply side and given off at the outdoor unit side. This explains the temperature drop of the supply water, observed in the night and the afternoon of 20 March and in the afternoon of 23 March. When T_{out} is below 0°C, the backup heater is used to prevent supply water temperature from cooling down too much. This explains the difference between the defrosting on 20 March at night (T_{out} < 0°C) and in the afternoon (T_{out} > 0°C). In the latter case the backup heater is not used and the heat pump cycles more often from heating to defrosting mode than in the former case. To decrease the amount of cycling, control settings should be adjusted to admit the use of the backup heater also in case of defrosting with outdoor temperatures above 0°C.

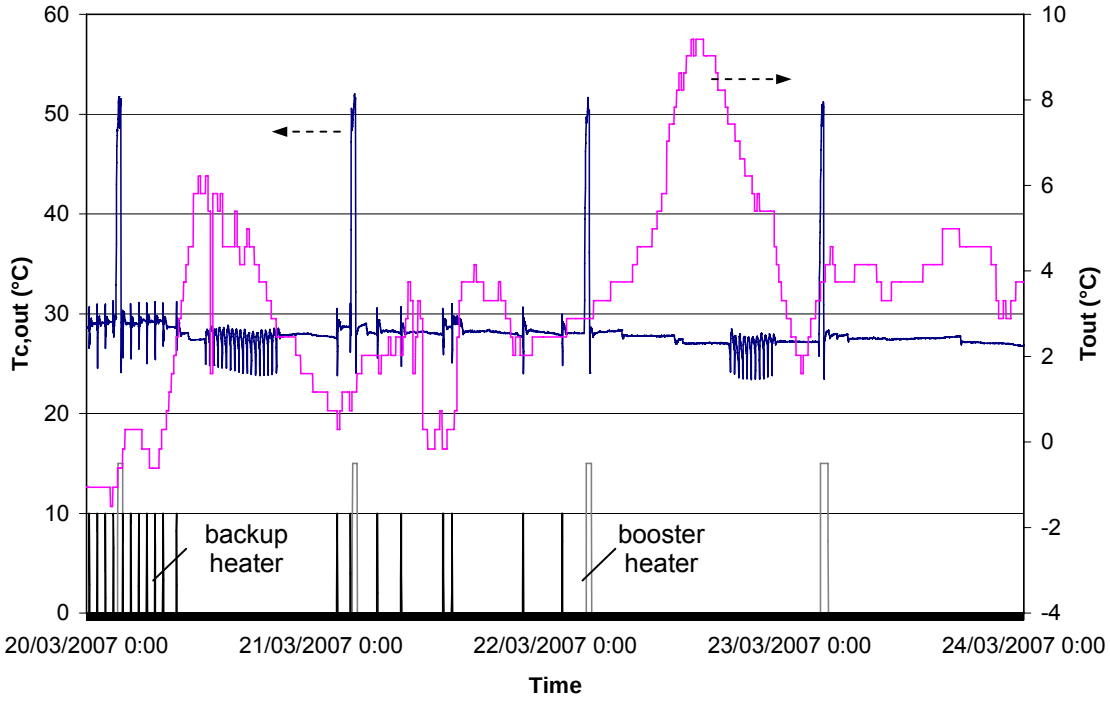


Figure 6: Evolution of the outdoor temperature (right axis), the condenser outlet water temperature (left axis), the on/off operation of the booster heater (gray) and backup heater (black) for a 4 days period.

4 MODELING

This section presents the development of a model of the building and the heating system. The aim of the model is to perform simulations for other weather conditions and to analyze the effect of some control settings on the system performance.

4.1 Building model

The building is modeled in TRNsys using building type56b. Local weather data are available from a local weather station (KULeuven 2007). Internal heat gains and ventilation losses are not included.

Table 4: Comparison of measured and simulated indoor air temperatures

	$T_{living\ room}$ (°C)	σ	$T_{bedroom}$ (°C)	σ
Measurement	20.5	2	19.6	1.5
Model	19.5	0.6	20	0.2

The simulated indoor air temperatures are compared to the measured values. The results are summarized in Table 4. The smaller temperature fluctuations observed in the simulation are probably due to the neglect of heat gains and ventilation losses. Despite this approximation, however, the average relative error on the weekly heat demand is only about ± 4%.

4.2 Heat pump model

A new heat pump model, which includes the effect of capacity modulation, is developed. The model is restricted to heating mode. DHW production, defrost and cooling mode are thus not included in the model. Both physical and equation fit models have been considered. Physical models provide greater flexibility, as the model can be extrapolated to working conditions other than the measured ones. They also provide more physical insight, as the influence of model parameters can be analyzed. The drawback is that physical knowledge of the process and a good estimation of the physical parameters are required. Equation fit models, on the contrary, are easy to set up but their validity is restricted to the data range. In this study, there is a lack of information concerning the part load characteristics of the heat pump and of the capacity control to develop a physical model. Therefore, an equation-fit type model is developed. As it concerns a modulating heat pump, the standard ARI540 (ARI 2007) representation of a heat pump, i.e. a cubic polynomial in T_c and T_e, can not be applied. The model should include the effect of capacity modulation. Therefore, two new approaches are worked out and evaluated.

4.2.1 Equation fit based on the supply water temperature error

The first approach is to consider compressor power and heating power as the product of the nominal power and a correction factor, which is a fourth order polynomial in function of the instantaneous difference ΔT between T_s and T_s*. This relationship is derived assuming a quadratic relationship between this temperature difference and compressor frequency on the one hand and a quadratic relationship between compressor frequency and power on the other hand (Shao et al. 2004). The 'temperature error based' equation fit has the following form:

$$\dot{P} = \dot{P}^*(a_1\Delta T^4 + b_1\Delta T^3 + c_1\Delta T^2 + d_1\Delta T + e_1)$$

$$\dot{Q} = \dot{Q}^*(a_2\Delta T^4 + b_2\Delta T^3 + c_2\Delta T^2 + d_2\Delta T + e_2)$$

$\dot{P}^*$ and $\dot{Q}^*$ are the nominal compressor and heating power, derived from catalogue data.

4.2.2 Equation fit based on building heating demand

The second approach is to consider the instantaneous heating demand of the building as a third independent variable, besides T_c and T_e. The idea is that the control action of the modulating heat pump is such that the instantaneous heating demand is always met. This means that the condenser outlet temperature equals the set point temperature given by the heating curve.

$$\dot{Q} = \dot{Q}_{demand} = \dot{m}_{water}(T_s^* - T_s)$$

As the heating demand is determined by the building, this equation fit can only be used when coupled to a representative building model. The 'heating demand based' equation fit has the following form:

$$\dot{P} = a + b\dot{Q} + cT_e + dT_{c,i} + e\dot{Q}^2 + fT_e^2 + gT_{c,i}^2 + h\dot{Q}T_{c,i} + i\dot{Q}T_{c,i} + jT_eT_{c,i}$$

In this polynomial T_c and T_e are normalized (Afjei and Wetter 2002):

$$T_{normalized} = \frac{T}{273{,}15} + 1$$

4.2.3 Data processing

Before deriving the coefficients of the equation fit models, the measured data are processed. Data containing information on defrosting and DHW production mode are filtered out. Electrical

power of the outdoor unit (compressor and fan) and condenser heating power are derived from the respective cumulative energy data. Only the measurements with a sampling time interval of 1 minute (as used in the period of 19 to 29 March) are used for identification of the coefficients as a lower sample frequency (e.g. 1 per 15 minutes) leads to unacceptable large averaging errors.

4.2.4 Results

The equation fit coefficients are determined using Gauss-Newton least sum of squared errors method (Afjei and Wetter 2002). The fitness of both approaches, evaluated for the given filtered data set, is given in Table 5.

Table 5: Evaluation of equation fits for representing heat pump operation in heating mode

	Temperature error based	Heat demand based
R^2	0.21	0.90
σ	14 %	7%
95% interval	± 28%	± 14%

As can be derived from Table 5, the heat demand based equation fit clearly performs better. This heat pump model, for which the coefficients are tabulated in Table 6, is now validated by using the data of the entire monitoring period, i.e. including the measurements with lower sampling frequency. In order to avoid the conversion from energy to power, which would introduce truncation and averaging errors, the simulation results and the measurements are compared in terms of energy. Figure 7 represents the evolution of cumulative energy in time for both simulation and measurements. The average relative error on the weekly electricity consumption is ± 18%.

Table 6: Coefficients of heat demand based equation fit for heat pump model

a	-13.10	f	-10.23
b	-1.622	g	-43.77
c	-1.155	h	-3.726
d	43.57	i	5.204
e	-0.011	j	26.48

4.2.5 Applicability of heat pump model

The equation fit derived is specific to the system studied since the model not only incorporates the heat pump characteristics, but also the heating curve and the building dynamics. This dependency is caused by the fact that capacity modulation control and heat pump are necessarily considered as one black box. This dependency is introduced in the model by the variable Q.
The model has been set up for an outdoor temperature range of 0°C to 10°C. As extrapolation may result in large errors, the flexibility of the heat pump model is limited to the temperature range observed.

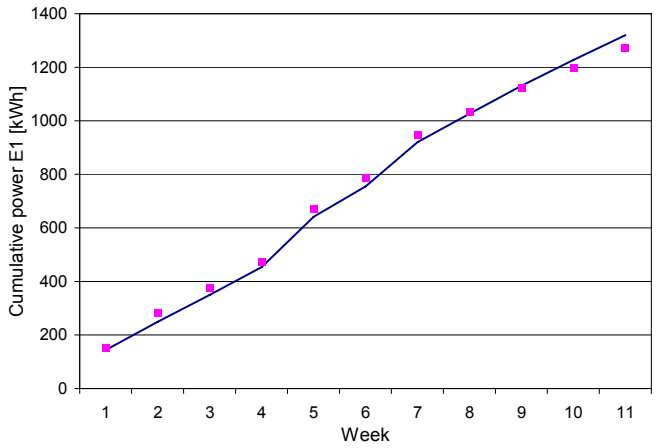


Figure 7: Cumulative power consumption outdoor unit: simulation (line) versus measurement (dots)

Nevertheless, the form of the equation fit (a second order polynomial with T_e, T_c and $Q_{building}$ as independent variables) is applicable to represent the building heating mode of other modulating heat pumps of this type, requiring only the coefficients to be redefined.

5 DISCUSSION

5.1 Effect of increased heat pump operation time for DHW production

During the night the DHW tank is fully heated to 55°C. The first 30 minutes the heat is delivered by the heat pump. Then, current control settings force the heat pump to switch back to floor heating mode in order to prevent the house from cooling down too much. This negatively affects the total system performance, as the electrical booster heater takes over while the heat pump could still provide a part of the heating load. Therefore, the effect of increasing the time period in DHW mode on the indoor temperature has been studied. The simulation results for a cold winter night show that after one hour the house only cools down by less than 1°C. In this case, increasing the period in DHW mode could thus improve the system performance without having a negative impact on thermal comfort.

5.2 Estimation of CO_2-emissions and comparison with condensing boiler

The CO_2-emissions related to building heating and DHW production during the monitored period (4687 kWh_e) are estimated at 1.15 ton, using the value of the Belgian electricity park, i.e. a specific CO_2-emission of 246 g/kWh_e (Commission Energy 2030, 2007). This is only a rough estimation as time dependency of the specific emission is neglected (e.g. other, less environmentally friendly power plants during on-peak then during off-peak period).
The calculated CO_2-emisssion is compared to that of a system with a condensing boiler, characterized by an efficiency of 102% (based on the lower heating value) for building heating mode (exit flue gas temperature 40°C) and an efficiency of 99% for DHW mode (exit flue gas temperature 60°C) (Che et al, 2004). Contrary to the heat pump system, no DHW tank is needed, which avoids the related thermal losses. For a building heating demand of 13 047 kWh and a DHW heating demand of 1169 kWh, the total primary energy consumption then equals 14.0 MWh_{prim}. Given the specific CO_2-emission for natural gas of 184 g/kWh_{prim}, the total CO_2-emission would amount to 2.57 ton.

The results indicate that the heat pump system studied realizes a CO_2-reduction of more than 50%, compared to a condensing boiler. Mind that this figure is obtained for the Belgian electricity park, which has a 55% share of nuclear power (Commission Energy 2030, 2007). For the given case, there is no CO_2-reduction if the specific CO_2-emission of electricity production exceeds 550 g/kWh$_e$.

6 CONCLUSIONS AND RECOMMENDATIONS

The experimental results show very good COP values for both floor heating and DHW mode, i.e. 4.1 and 2.4. The SPF for the system amounts to 3.5 for the heating season. Including the DHW production in summer, the annual SPF is 3.0. Analyzing the measurement data, two opportunities for improvement could be identified. First, control settings should be adjusted such that the share of the booster heater is reduced. Second, a higher insulation rate of the buffer tank is recommended.
A new equation fit model has been developed which takes the capacity modulation into account. This is realized by introducing a third variable, the instantaneous building heating demand, into the classical quadratic polynomial with variables T_c and T_e. This model, which needs to be connected to a building model, can describe the heat pump operation in heating mode with an accuracy of 18% for the given outdoor temperature range. In order to develop a physical model of the heat pump, information on the part load characteristics and frequency modulation control is required.

ACKNOWLEDGEMENTS

The monitoring work presented in this paper is realized in cooperation with Daikin Europe, Belgium. The authors also want acknowledge the owners of the studied building, Leen Peeters and Diederik Verkest, for their commitment to the project.

7 REFERENCES

Afjei T., M.Wetter, 2002. “Berechnung von polynomkoeffizienten” (biquadr-polynom-1.0-a.xls)
ARI, 2008. *Air Conditioning and Refrigeration Institute*. www.ari.org
Che, D., Y. Liu, C.Gao, 2004. "Evaluation of retrofitting a conventional natural gas fired boiler into a condensing boiler," *Energy Conversion and Management*, Vol.45, Part 20, pp. 3251-3266.
Commission Energy 2030. 2007. *Belgium’s energy challenges towards 2030*, Commissioned by Minister Marc Verwilghen, Brussel.
Flemish Authorities, department Environment. 2007. http://lucht.milieuinfo.be, Belgium
KMI 2008. *Koninklijk Meteorologisch Instituut*. www.kmi.be
KULeuven, 2007. Monitoring of Vliet building, Leuven.
Shao S., S. Wenxing, L. Xianting, and C. Huajun. Jr.2004. "Performance representation of variable-speed compressor for inverter air conditioners based on experimental data," *International Journal of Refrigeration,* Vol.27, Part 8, pp. 805-815.