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Performance Evaluation Of A Heat Pump Water Heater By Means Of Thermodynamic Simulation

Kevin Ruben Deutz[a,b], Axel D'Angelo[b], Odile Cauret[b], Romuald Rullière[a], Philippe Haberschill[a]

[a]*Université de Lyon, CNRS, INSA-Lyon, CETHIL, UMR5008, Villeurbanne*
[b]*EDF R&D, Energy in Buildings and Territories Department, Moret-Sur-Loing*

Abstract

Heat Pump Water Heaters are becoming more and more interesting technologies for efficient sanitary hot water production. The specificities of hot water production compared to the use of heat pumps for space heating are the relatively constant energy needs for different outdoor temperatures and the more rapid dynamics associated with water temperature elevation.
This study focuses on the performance evaluation of an R134a air to water heat pump water heater with a mantle heat exchanger which is the most common technology on the French market. A detailed Modelica based model is built and compared with experimental data in different operating conditions: draw-off, standby, heating and defrosting periods. It is shown that on a 5 day normative test sequence the Mean Absolute Percentile Error (MAPE) for modeled electric compressor power is of 7.4% and the maximum error in the tank temperature prediction is of 0.9K.
In a second part, a parametric study is made and the improvement potential of the system is expressed according to a set of design variables such as insulation level or condenser position. The model is then used to assess the heat loss locations on an annual basis. It is found that the system's thermal performance is mostly affected by three energy losses that in order of magnitude are the thermal losses from the tank (19.1%) , the compressor losses (7.9%) and the defrosting phases that are found to play a less significant importance (0.4%).
Annual simulations are finally made to assess new control strategies emerging on the HPWH market based on inverter compression such as fixed condenser power aiming to increase comfort by providing a fixed heat-up time, and smart control where the tank water temperature setpoint is controlled based on daily tapped energy records.

"Keywords: Heat Pump Water Heater; Simulation; Parametric study; Performances; Control; Design Optimisation"

Nomenclature

A	Surface	(m²)
U	Overall heat transfer coefficient	(W/ m²/°C)
T	Temperature	(K)
c_v, c_p	Specific heat capacity per unit volume or mass	(J/K/m^3) - (J/K/kg)
h	Specific enthalpy - or height	(J/kg/K) - (m)
$\dot{m}$	Mass flow rate	(kg/s)
Pr, Ra	Prandtl and Rayleigh dimensionless number	(–)
V	Volume	(m^3)
η	Efficiency	(–)
ρ	Density	(kg/m^3)
α	Thermal diffusivity	(m²/s)
p	Tank perimeter	(m)
Q	Heat	(J)
Subscripts		
HP	High Pressure side	
LP	Low Pressure side	
bl	Boundary layer	

1. Introduction

Heat Pump Water Heaters (HPWH) have been under investigation since the 1950s, [1]. However since the beginning of the 2000's and the emergence of progressively more stringent energy policies, HPWH have been gaining on market penetration. In the total energy demand of a building, Domestic Hot Water (DHW) has the particularity to be relatively climate independent, [2], and has a seemingly constant relative value of 10 % over the period of 1973 to 2012 of the final energy consumption of a typical French building, [3]. The legislative purpose to reduce the overall primary energy consumption implies that in the near future the relative share of the energy demand for hot water in the total energy demand of a building will increase.

Within this context of progressive reduction of building energy needs, HPWH performances are playing a more and more important role. In the literature, numerous studies can be found that analyse the energy performance of HPWH either by means of modeling, experimental investigation or field testing, [4], [5]. Most of the studies focused on particular aspects of the thermodynamic cycle configuration: condenser configuration, the use of refrigerant mixtures or double-stage compression with injection, [6], [7], [8]. Performances of a HPWH thermodynamic cycle can be expressed in different forms and on different timescales. In scientific literature, some authors use exergy defined methodologies to quantify the irreversibilities occurring in the components of the cycle and to perform cycle optimization, [5], [9]. Other authors, use pure energy based studies to perform techno-economic studies on system design or optimal control, [4]. On the other hand, in technical brochures, the performances are given in a certified way using a standardized procedure under fixed laboratory conditions.

To fully reflect the performances of the HPWH system according to the stochastic nature of user behaviour and the full range of climatic conditions, it is believed that a combination of the different identified metrics on different time horizons is needed (from a simple heat-up to an annual simulation). In the present article we use a model that was described at the Purdue conference, [10]. The following article relates first, to the improvements made to the initial model combined with an experimental comparison in different operating conditions. Then a parametric study is performed to assess the impact of design or operating parameters on the performance of the HPWH. At last the model is used in annual simulations to evaluate the prevailing thermal losses of the system and to assess new control strategies based on inverter compression that are now emerging on the HPWH market.

2. Model of the HPWH cycle

2.1. System Description

During a market overview, it was observed that the French market offer is strongly dominated by external air HPWH consisting of a classic small capacity vapor compression cycle made up of an expansion valve, an evaporator, a compressor and a condenser connected to a water Thermal Storage Tank (TST) situated inside the living space. Typically three kinds of condenser layouts can be found for the connection between the heat pump unit and the TST which consist of mantle, immersed and external heat exchangers. Our investigation relates to the most prevailing system on the French market which is an external air source R134a HPWH with a mantle type heat exchanger (MHX) surrounding the tank wall (Figure 1).

The general specifications of the HPWH are given in Table 1. The original fixed speed rotary compressor was replaced with a variable speed one to perform experimental investigation regarding variable speed control.

Table 1. Heat Pump Water Heater Technical Specifications

Heat Pump Specifications	
Total power (W)	2800
Heat Pump Maximum electric power (W)	1000
Electric auxiliary heater power (W)	1800
Refrigerant type/mass (kg)	R134a / 1.1
Volume (l)	200

2.2. Modeling the Heat Pump

The heat pump is modeled dynamically using the TIL thermal component library developed under Modelica that comes with a modular architecture and a fluid thermal property calculation library, TILmedia, [11].

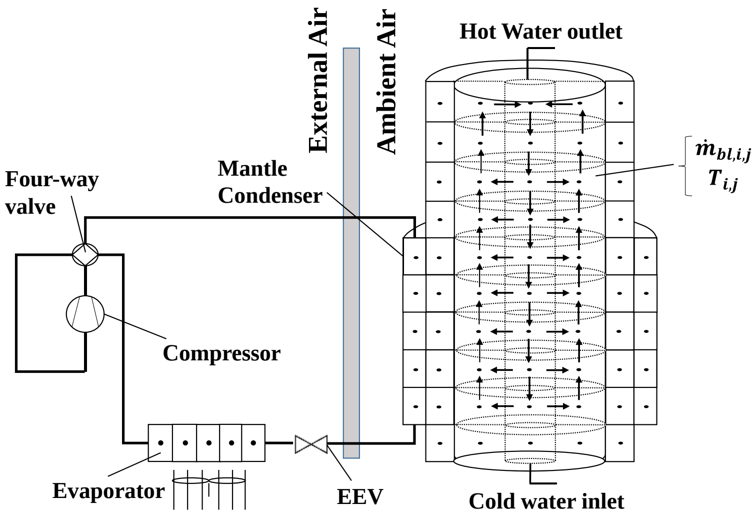

Figure 1. System configuration and zonal model illustration

The compressor model and the electronic expansion valve (EEV) model are developed using a black box methodology with manufacturer available data, see previous publication for more details, [10]. Oil is neglected and the system is assumed to operate stably in all conditions.
The defrosting method of the HPWH is done by a four way valve (FWV) that is modeled by means of two volumes representing the high and low pressure lines. To represent the energy transfer between the high pressure and low pressure line we perform an energy balance for each individual volume in the following form:

$$c_{v,HP} V_{valve,HP} \frac{dT_{valve,HP}}{dt} = \dot{m} \Delta h_{HP} + UA_{valve} (T_{valve,LP} - T_{valve,HP}) \tag{1}$$

In terms of HPWH control, defrosting is assumed to be done preventively once the external air temperature is below 5°C. Every hour and a half, the FWV reverses the flow for a 3 minute period. No detailed frost formation or removal model is applied for the evaporator and the preventive reversed defrost was assumed a 100% effective.

2.3. Modeling the Thermal Storage Tank (TST) combining a zonal and 1D approach

Due to natural convection a boundary layer upward flow is generated along the tank wall when heated up by the MHX (Figure 1). This was modeled by a zonal approach from the literature, [11] that was modified for greater flexibility, [10]. Zonal modeling uses a pattern based approach that allows to cancel out the momentum equation in the Navier-Stokes equations. In our case, we consider the boundary layer as an upward flow (Figure 1) that can be expressed by adopting the integral method proposed by Kenjo et al. (2007), [11]:

$$\dot{m}_{bl,i,j} = 0.098 \alpha \rho Pr^{\frac{1}{15}} \left[\frac{Ra_{i,j}}{1 + 0.494 Pr^{\frac{2}{3}}} \right]^{2/5} p \tag{2}$$

For draw-off periods at a given flow-rate, it was observed experimentally that two types of flow region exist in the tank. In the bottom region, intense mixing was observed close to the inlet up to a given transition point. From there up to the upper part of the tank a perfect 1D piston flow seemed to take place. Hence, the tank model was divided in two areas based on a mixing height, see previous publication, [10]. Experimentally, we observed that the mixing height reaches a minimum constant value for low flow rates and a maximum value for higher flow rates and decreases linearly in between. To estimate the mixing height's linear relation with draw-off flow we performed draw-offs from an initial hot tank until the water in the tank reaches an asymptotic minimum temperature, defined as the mixed temperature, T_{mix}. For the period, the energy balance can be expressed as:

$$Q_{drawoff}(t = t_{end}) = Q_{stored}(t = 0) - Q_{stored}(t = t_{end}) - \int_{t=0}^{t=t_{end}} \dot{Q}_{loss}(t) dt \tag{3}$$

Neglecting the thermal losses from the tank during the draw-off period and considering a fixed mixing height:

$$Q_{stored}(t = t_{end}) = (1 - \eta_{drawoff}) Q_{stored}(t = 0) \tag{4}$$

By correlating the tapping efficiency $\eta_{drawoff} = \frac{Q_{drawoff}(t=t_{end})}{Q_{stored}(t=0)}$ linearly and empirically to the tapping flow rate and by assuming constant initial temperature in the tank, we can express the equivalent mixing height as:

$$h_{mix} = (1 - \eta_{drawoff}) h_{tank} \tag{5}$$

The hypothesis expressed are particularly true for large draw-offs such as a showers. For intermediary conditions such as low flow rates in stratified tanks or during tank heat ups, we assume no mixing in the tank.

3. Experimental Comparison

The experimental comparison is done on two different timescales, first we confront the model to a normative draw-off scenario defined by the French national testing standards. In a second part, we perform different heat-up cycles from an initial cold water tank at 10°C to 55°C while varying compressor speed.

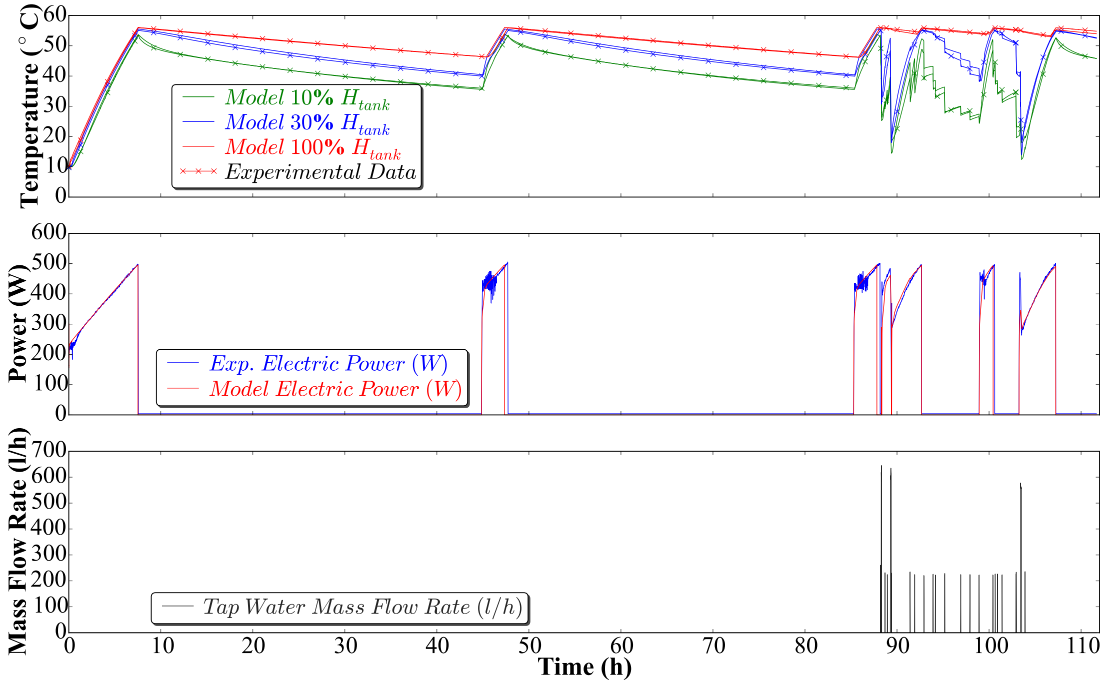

Figure 2. Model to experimental data comparison on a normative test sequence at Tair=7°C

3.1. Normative Test sequence

The normative test sequence consists of a first water heat up period from 10 to 55°C followed up by a 48hour stand-by period. Thereafter a tapping sequence is initiated involving two morning and one night shower at flow rates of 600 l/h separated by smaller draw-offs of 240l/h during the day. Such a testing procedure allows to compare the model with experimental data in various operating conditions: uniform initial temperature heating, stratified heating, standby periods and draw-offs. From figure 2, a good agreement is found between the model and the experimental data both in the temperature profile measured in the tank and the compressor electric shaft power, see table 2 for a quantitative error analysis. Deviations can be found particularly in the phases combining draw-offs and heating and during compressor start-up where experimentally we experienced instabilities mainly due to EEV hunting (repeated opening and closing of the valve).

Table 2. Error analysis of the HPWH model versus experimental data

Error Analysis	Temp. (K) 100% Htank	Temp. (K) 30% Htank	Temp. (K) 30% Htank	Power
Mean Absolute Error (MAE)	0.31	0.89	0.90	25.45 W
Mean Absolute Percentile Error (MAPE)	-	-	-	7.43%

3.2. Comparison at different compressor speeds

It can be observed in figure 3, that the heating time reduces with higher compressor speeds as the average heating condenser power increases. The electric consumption increases along the heating period for all frequencies as the condensing pressure increases with the water temperature rise. The heating condenser power decreases along with the compressor efficiency that decreases with the increase in compression ratio. The effect is more pronounced for higher speeds as the compressor efficiency also decreases with the compressor speed. In a general manner, it can be stated that the model complies well with the experimental data. However at 120 Hz there seems to be a bigger deviation. This can be related to the fact that the compressor efficiency maps were established for speeds ranging from 30 to 90Hz due to a lack in available data from the compressor manufacturer.

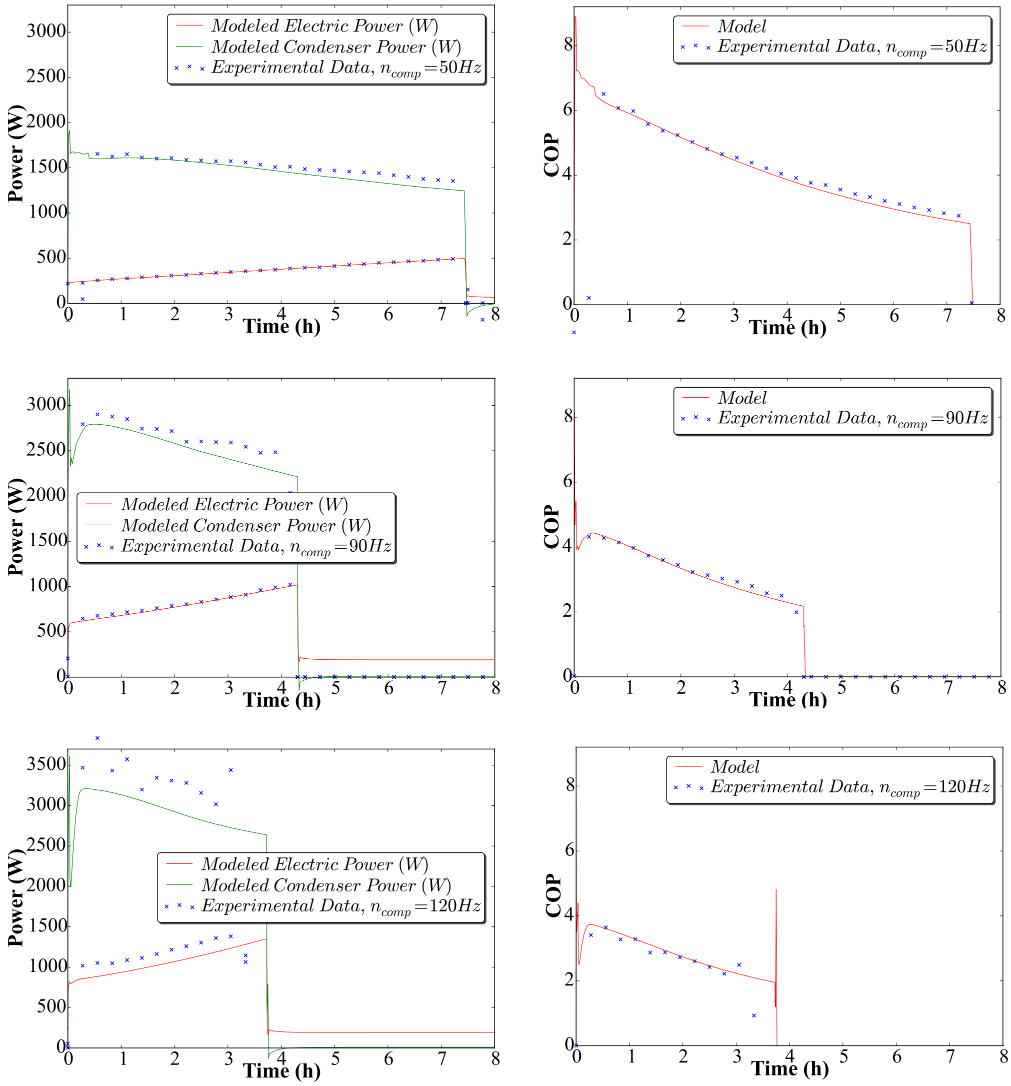

Figure 3. Model to experimental data comparison for heat-ups from 10 to 55°C at various compressor frequencies and Tair=7°C

We define the Coefficient of Performance (COP) as the ratio of the heating useful condenser power to the electric compressor power. Given the previous analysis, with decreasing condenser power and increasing electric power, the COP decrease in figure 3 can be explained. The transient start-up phases from an ambient temperature equilibrium seem to be well represented as we see the inertia of the system affecting the COP from a low start (when storing non useful energy in the system's components) to a maximum quasi-steady state where the tank water is still cold enough to permit a low compression ratio for full refrigerant condensation.

4. Parametric Study

In this section we test a set of design and operating conditions in a parametric study and analyze how they affect the system's COP. We believe that any enhancements will have different impacts on the system and might affect the performances either during heat-up, draw-off or standby periods. Hence two different COP values are used:

- The heating COP from an initial cold tank at 10°C to 55°C
- The normative COP performed by imposing the French testing protocol described in section 3.1.

4.1. Heating COP according to Boundary Conditions

Here we vary the compressor speed to assess the impact of inverter compression at different air temperatures for a constant fan speed. From figure 4, it can be seen that the maximum COP varies both with temperature and compressor speed. In general, the optimum is situated around the expected value of 40 to 50 Hz where the best compromise is achieved between compressor thermal losses and friction losses. However, for higher outdoor temperatures this maximum tends to shift towards the 30 Hz limit, in line with the reduction of compressor pressure ratio and the reduction of compressor thermal losses. When the pressure ratio increases with the decrease of outdoor temperatures, the shell temperature increases and the thermal losses per revolution become more important hence the maximum COP moves to the higher value limit of 60Hz.

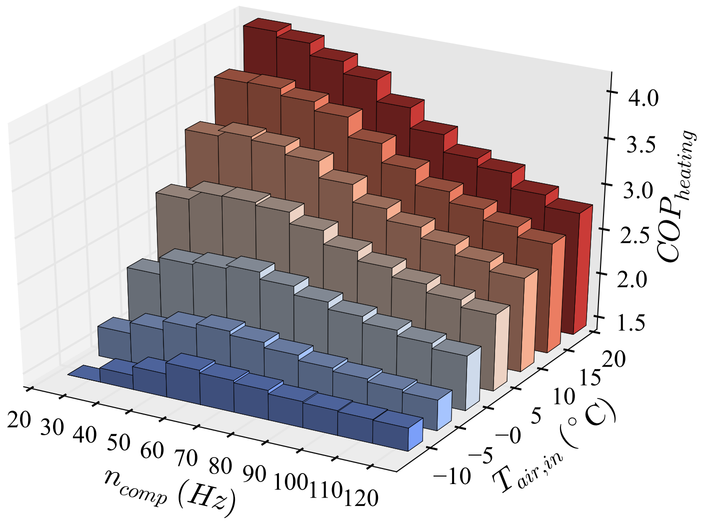

Figure 4. HPWH heating COP from 10°C to 55°C according to compressor speed and air temperatures at fixed ventilation speed

It can also be seen, that the decrease in COP is more pronounced towards the high compressor frequency limit of 120Hz than towards the low frequency limit of 30Hz, except when the temperature is between -10°C to 5°C. For those cases, a 30Hz frequency combined with high water temperatures and low external temperatures results in low condenser thermal capacity. This causes longer heating times and increased tank thermal losses.

4.2. Heating COP as a function of mantle condenser area and contact heat transfer coefficient

The heating COP can be affect by a multitude of parameters that are mostly located on the heat pump side. Looking at the MHX, it can be seen in figure 5 that the COP increases towards an asymptote when increasing the thermal conductance from the condenser wall to the tank wall. This can be explained by the fact that at a given conductance, the contact heat resistance plays a less significant role in the overall heat transfer coefficient from the condenser to the water. It was found by authors, [12], that micro-channels on the condenser side don't have a significant impact on the heating COP neither and that the limiting thermal resistance was on the water side.

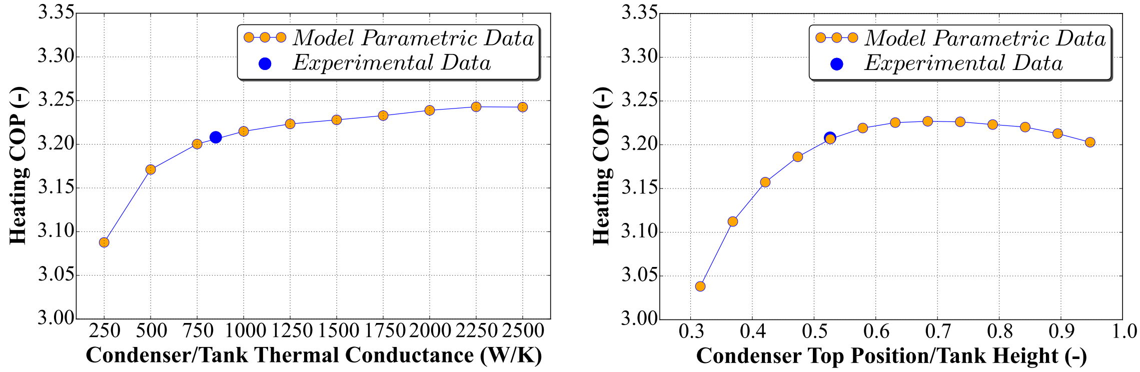

Figure 5. Heating COP according to thermal conductance between condenser and tank wall and according to condenser position

To increase the heat transfer surface between the condenser and the tank wall, it is possible to increase the amount of coils surrounding the tank. From figure 5, it can be seen that by doing so, with the given charge of 1.1kg a maximum value seems to be reached around 70% of the tank height. When the condenser area is too small the COP is low because to fully condense, the refrigerant pressure is increased. When the condenser area becomes large enough, the heat pump can operate at its minimum operating pressure related to the minimum pinch point between the water and the refrigerant temperature. In the actual state, the studied HPWH is already close to the maximum value, hence the improvement potential is here quite low.

4.3. Normative COP according to design parameters

Two parameters that can affect the normative heating COP without significantly affecting the heating COP are the mixing effects induced by incoming cold water and the insulation of the tank that affect the storing efficiency. From figure 6, it can be seen that there is still room for improvement both on the inlet water diffuser and on the insulation level of the tank. Simply by doubling the insulation level, the normative COP of the actual system can be improved by 0.16. The room for improvement for the inlet diffuser is however more restrained as we can see the difference between the actual and maximum COP is of 0.048. Perfect stratification during draw-off periods is something difficult to attain and due to the fluid nature of water, some mixing will ineluctably take place.

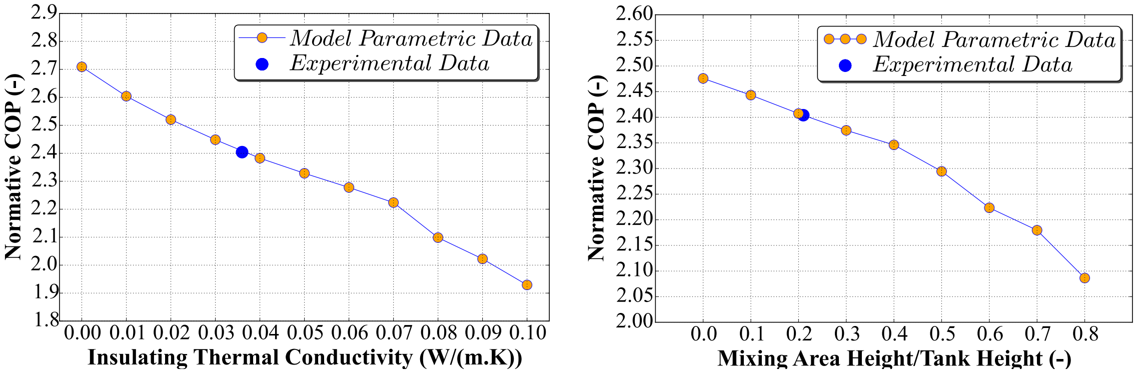

Figure 6. Normative COP according to tank design variables

5. Annual simulations

5.1. Thermal and Electrical energy breakdown by use

In this section we assess the HPWH system on an annual level. For that we use a set of given real weather conditions at a time step of an hour issued using the software Meteonorm for 5 different climates: Paris, Marseille, Strasbourg, Karlsruhe and Birmingham. As for the tapping scenario, we use a real tapping profile recorded on site corresponding to a family of four persons tapping 157l/day of hot water at 40°C.

The control strategy for the heat pump is the most forthcoming one proposed on the market that uses a backup auxiliary heater and the reverse defrosting technique discussed in section 2.2. The backup electric heater is activated once the air temperature gets below 2°C and takes over the heat pump for water temperatures between 50°C and 55°C. From figure 7, the global energy balance on the HPWH as a whole {Heat Pump + Tank} on yearly simulations is decomposed in the thermal energies produced by the HPWH and the electric energy used by the compressor and the fan for heating and defrosting and the auxiliary heater consumption.

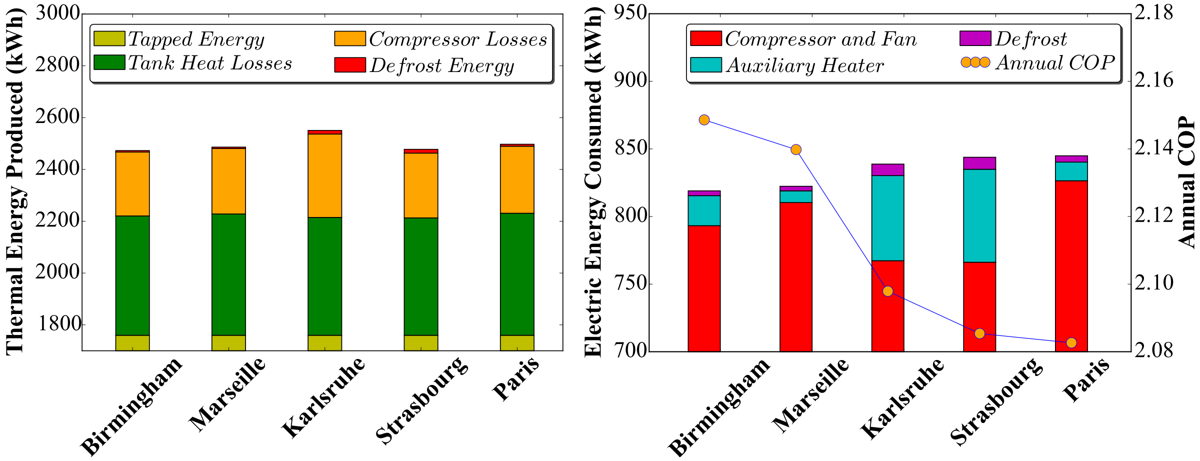

Figure 7. Annual energy consumption and production decomposition and COP with varying weather profile

It can be point it out that in terms of energy losses of the system, thermal losses from the tank play the most significant share with 19.1% of the average total thermal energy produced, followed by the compressor losses (corresponding to the compressor thermal losses and inertia) with 7.9%, and finally followed by the defrost losses that only represent 0.4%. On the electric energy side, the dominant share is the compressor with the external fan unit, followed by the auxiliary heater (that is started when the temperatures are low) and the defrost cycle electric energy (consisting of the direct electricity share needed to defrost and the indirect share needed to reheat the tank after the reversed cycle defrost operation is terminated).

Surprisingly, in figure 7, it can be noticed that even though the climate is colder between Paris and Strasbourg, the annual COP of the HPWH seems to be equivalent. One of the reasons is that although the heating COP is globally higher for a heat up from 10°C to 55°C, the instantaneous COP in between 50°C and 55°C could be in the region of one at very low temperatures. Those could be the particular conditions when a simple stage heat pump could effectively be replaced with a double stage heat pump providing better efficiency at higher compression ratio [8]. Further investigation will be made to confirm this hypothesis in later studies.

5.2. Annual simulations using different control strategy and tapping profiles

For now the standard control uses a backup electric heater to supplement the heat pump when the heat pump thermal capacity is insufficient, typically for low external temperatures. On the HPWH market, some new HPWH systems come with a fixed condenser power with inverter compressors using the compressor speed as an actuating variable. In such a manner, the HPWH can provide a fixed heating time in any external air temperature.

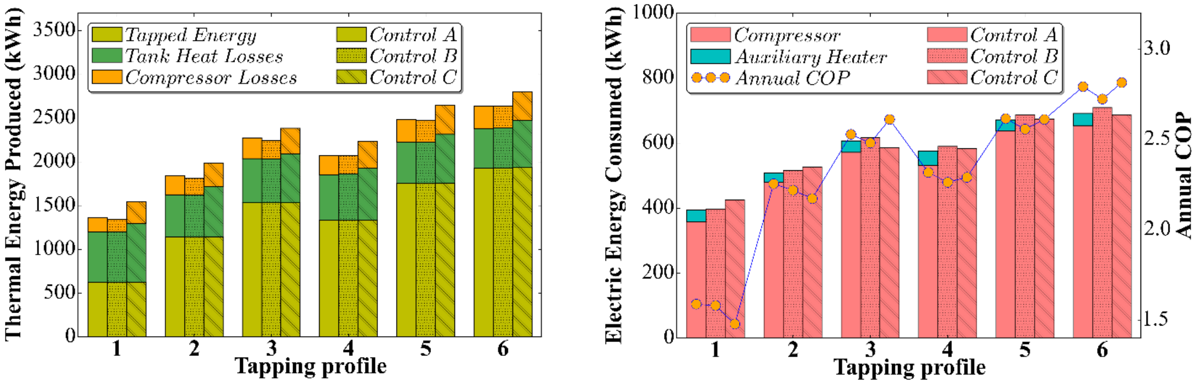

Figure 8. Annual energy consumption and production decomposition and COP varying tapping profile and control strategy

Other constructors provide a “smart control” algorithm for the HPWH that records the tapped energy by the household on a daily bases and adapts the thermal energy supplied by the HPWH by varying the water setpoint temperature. The drawbacks of this method is that an external air HPWH is very climate sensitive and the heating time required to produce the given recorded energy in a day varies according to the external temperature.

We developed a combination of the two previous control strategies aiming at producing the recorded thermal load in a fixed heating time in valley tariffs while avoiding auxiliary heating usage. We assume that the valley tariff is of 11.5c€/kWh from 10 p.m. to 6 a.m. and that the base tariff is of 16.3c€/kWh. The original control scenario will be referred to as scenario A, the fixed condenser power as scenario B, and the combination of the fixed power and smart control as scenario C. For simplicity, we assume that the defrosting cycles can be neglected as they have shown negligible impact for scenario A in various climates (Figure 7).

To confront the control strategies with different tapping profiles, we run annual simulations on six different realistic tapping profiles going from a low water usage of 50l/day of 40°C water for scenario 1 up to 157l/day for scenario 6. From figure 8, it is perceptible that the COP generally tends to be higher when the water tapping volume is larger. This tendency can be explained by the fact that the absolute share of tank thermal losses becomes less important relatively to the tapped energy.

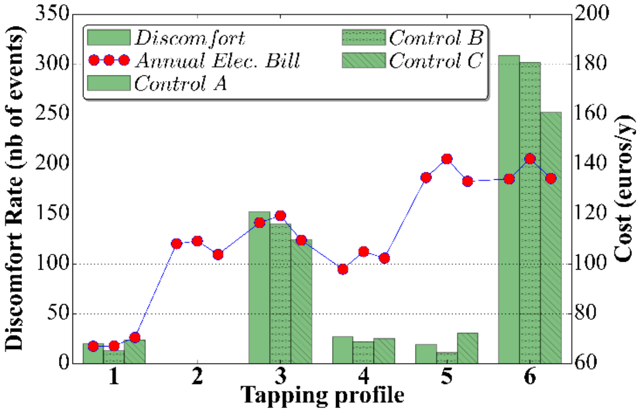

Figure 9. Annual electricity bill and discomfort rate varying weather profile and control strategy

In terms of control strategy, scenario A and B have approximately the same repartition in terms of produced energy. However, in scenario B the compressor generally runs at higher speeds to perform the fixed heating time causing a degradation in the annual COP. As what concerns control C, an increase in thermal losses can be noticed from figure 8. This can be related to the fact that the HPWH operates during the valley periods at night shifted from the actual usage which leads to longer standby periods and increased electricity consumption.

In terms of cost and comfort, from figure 9, it can be noticed that scenario B leads to an increase in annual bill against a slight reduction of the discomfort rate for all the tapping scenarios tested. Scenario C on the other hand, by optimizing low fare valley price operation generally leads to a reduction in cost while significantly reducing the discomfort rate in scenario 3 and 6. This is because this scenario guarantees a full tank reheat during the night while being hysteresis backup controlled during the day. Also those scenarios were dominated by regular important morning draw offs anticipated by the smart control algorithm producing hot water during the night.

6. Conclusion

A general thermodynamic model of a HPWH was developed using a semi-empirical approach for the heat pump and a zonal hybrid 1D model for the TST. This model was validated on a normative test sequence and in heat up phases at various compressor frequencies. The model was later used to perform a parametric study on both the two validated performance variables according to a set of design parameters and operating parameters. Finally annual simulations were performed to decompose the energy usage of the system according to different climatic conditions and different control strategies were tested varying the user draw-off profiles. It was found that:

- The optimal compressor speed to maximize performances varies according to the external air temperatures between 30 and 60 Hz for external air temperatures ranging between -10°C and 20°C.
- The actual insulation level of the tank could be improved. Doubling the insulation thickness could lead to a 0.16 normative COP increase compared to the actual system design.

- Annual COP increases with the total annual tapped energy. By assuming that defrosting is optimal, preventive reversed defrosting doesn't affect the performances of the machine much. Static thermal losses from the tank and cycle performance are found to offer the best improvement potential.
- The fixed condenser power control strategy has not been found to provide interesting annual cost reductions or significant increased thermal comfort. Combining this method with a smart control approach leads to slightly better results in some of the tested profiles but further research is still needed to improve the control strategy.

The perspectives of the study are to improve the prediction methods, to perform multi-criteria design based optimization, and to develop and test advanced cycle configurations such as double stage compression.

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