Improved methodology for testing the part load performance of water-to-water heat pumps

Elena Fuentes[1], David Waddicor[1], Mohammed Omar Fannan[2], Jaume Salom[1]

[1]Catalonia Institute for Energy Research, Barcelona, 08930, Spain
[2] Schneider Electric Industries SAS, Grenoble, 38000, France

Abstract

With the continued expansion in the implementation of heat pump systems in buildings there is a strong need to better characterize their energy performance at different operating conditions. However, for the case of fixed-capacity water-to-water heat pumps, existing rating methods to determine systems performance at part load have not been assessed for their validity to represent efficiency at real operating conditions. An improved method is proposed in the present study for the characterization of the part load performance of fixed-capacity heat pumps based on reduced experimentation. The methodology is based on the performance of one experiment at stand-by and another at intermediate part load conditions, combined with the use of a partial load factor theoretical equation, in order to determine the degradation coefficients for start-up and stand-by efficiency losses. The method is compared with current methodologies in technical standards and is shown to improve the predictions of part load performance over previous techniques. A black-box empirical model for the heat pump under study is developed and validated with experimental data, demonstrating the potential of using computationally simple methods to model the transient behavior of fixed-capacity heat pump systems, including all energy parasitic losses.

1. Introduction

With the worldwide trend towards electrification, heat pumps have become a technology with increasing expansion for their application in building heating and cooling space conditioning. Although a significant number of studies have focused on studying the energy efficiency of air-to-water and water-to-water units, few works have focused on understanding the behavior of water-to-water systems at part load conditions [1]. The energy efficiency of fixed-capacity heat pumps is affected by part load operation, which leads to a loss of performance as a result of parasitic effects [1, 2]. Laboratory experiments on the performance of a ground source water-to-water heat pump by Man et al. [3] have revealed an important influence of the operating conditions and the intermittent regime operation of the system on the energy performance. Montagud et al. [4] developed a model of a water-to-water heat pump based on measured data from real equipment operated in an office building. Although the heat pump model developed by these authors provided an acceptable approximation of the system´s electrical energy consumption, the largest deviation in predicting the performance of the heat pump was due to the parasitic losses induced during its transient behavior. Research studies by Schibuola et al. [5], Bettanini et al. [6] and Riviere et al. [7] have shown that the degradation of the energy performance of heat pump units increases with decreasing load ratios, due to more frequent cycling and longer stand-by periods as the load is reduced. Further simulations by Madani et al. [8] and experiments by Waddicor et al. [9] have shown that the energy efficiency of water-to-water systems at part load is highly dependent on the specific control setting configuration.

In order to characterize and label the energy performance coefficient of heat pump systems at part load, several testing and calculation methods are proposed in technical standards (eg. EN14825, ARI 210/240 and UNI 10963) to calculate a characteristic annual seasonal performance factor for this equipment [10-12]. Although standard methods are generally accepted for application to water-to-water heat pump units, the validity of such methods has not been evaluated for all type of heat pumps and operating conditions. On the other hand, although parameterizations in standards can be applied to estimate overall coefficients of performance (COP) inclusive of energy parasitic losses, it is necessary to develop and test heat pump models that are able to predict the real-time transient behavior of heat pumps, including the parasitic phenomena leading to energy losses [13].

Within this context, in the present study diverse methods in technical standards to estimate the part load performance of water-to-water heat pumps are assessed and validated through a comparison with data from laboratory experiments. A new reduced experimentation method is proposed that allows determining the part load performance of fixed-capacity heat pumps with improved accuracy and a minimum number of experiments. Along with this methodology, a black-box heat pump model is developed and validated by comparing with transient real-time experimental data and hourly coefficient of performance values at different part load ratios.

a, b, c, d, e, f, g, i, j, k, l	empirical coefficients
$b_1 \ldots b_6$	empirical coefficients
COP	coefficient of performance
C_d	degradation coefficient (start-up loss)
C_c	degradation coefficient (stand-by loss)
P_{elec}	electrical power consumption (kW)
$P_{stand\text{-}by}$	electrical power consumption at stand-by (kW)
$P_{full\ load}$	electrical power consumption at full load conditions (kW)
PLF	partial load factor
PLR	partial load ratio
SCOPnet	annual coefficient of performance
$T_{cond.in}$	inlet temperature to condenser (°C)
$T_{cond.out}$	outlet temperature from condenser (°C)
$T_{evap.in}$	inlet temperature to evaporator (°C)
$T_{evap.out}$	outlet temperature from evaporator (°C)
Q_{cond}	heat rate at condenser (kW)
$Q_{cond,start\text{-}up}$	heat rate at condenser during start-up (kW)
Q_{evap}	heat rate at evaporator (kW)
Z	ratio between electrical power at cycling and steady-state conditions

heating capacity. The heat pump was tested in a laboratory that allowed operating the equipment in a hardware-in-the-loop configuration, with connection to a virtual storage tank and a simulated building heating load. The experimental set up consisted of two thermal test benches used to emulate the heat exchange with the ground and with a virtual storage tank connected to the heating load (Figure 1).

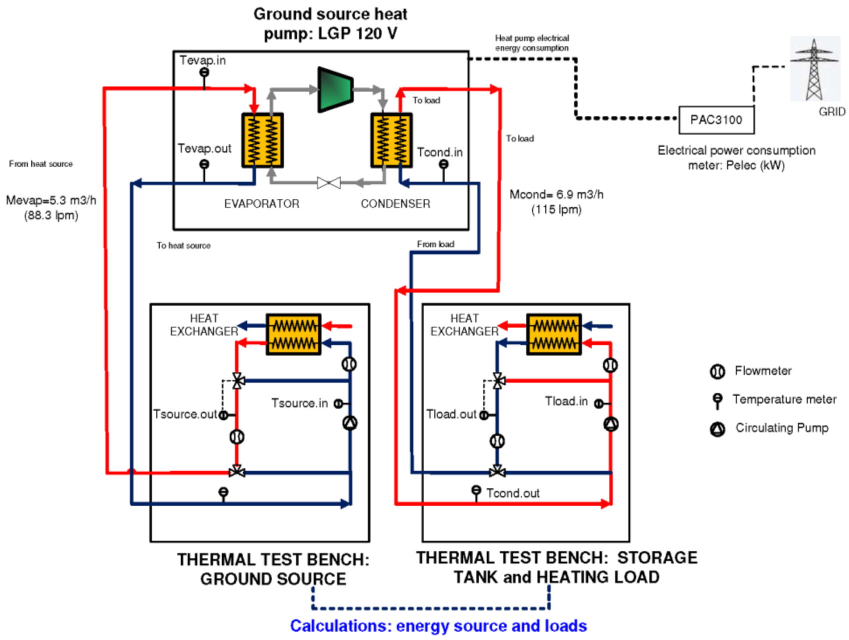

Figure 1: Experimental set-up

The thermal test benches are provided with flow and temperature sensors in a control system that allows emulating the thermal loads and the heat source. The test benches are connected to two external heating and cooling secondary circuits provided with a boiler and a chiller, that are used to control the inlet temperatures from the heat source and load, respectively. The evaporator inlet flow temperature was set to a constant value of 15 °C, while the condenser inlet water temperature was continuously adapted to emulate the return temperature from the simulated building load. Flow rates were controlled with a three-way electronic valve connected to an external by-pass and an induction flow meter. The water flow through the evaporator and condenser were set to constant values of 5.9 and 6.3 m^3/h, respectively. The heat pump was regulated with standard thermostat control on the condenser inlet flow temperature. The heat pump return temperature set point was set to 48 °C, with a dead band of 2 °C. The virtual tanks and loads were simulated using the TRNSYS software [14]. The tank model applied was the so-called type 534 from the TESS component libraries [15]. This is a detailed model for a stratified tank previously validated by Allard and Kummert [16]. Two port flows were defined for configuration of the tank model, one of them linked to a steady load that represents the simulated building heating load and a second flow connected to the condenser of the real heat pump. The number of nodes in the tank model was calculated using the approach of Kleinbach et al. [17], and the dimensions and insulation properties of the tank were obtained from manufacturer catalogue data. Different load levels between 8 and 40.5 kW were emulated for storage tank sizes of 50 l, 100 l, 300 l, 500 l and 1000 l. For each load conditions, a preconditioning period of 15 minutes was adopted, followed by a 60 minutes continuous measurement stage.

2.2. Results of energy performance experiments

The experimental results were analysed to obtain the coefficient of performance (COP) and partial load factor (PLF), which is the ratio between the COP at part load and the COP at full load conditions. To analyse the efficiency deterioration at part load, the PLF data is represented versus de partial load ratio (PLR). The effect of the inertia conditions on the efficiency degradation is shown in Figure 2 (left), where increasing degradation of the energy efficiency is seen as the partial load ratio decreases. The coefficient of performance (COP) is higher for increasing inertia due to lower degradation effects and because the modulating effect of inertia leads to a more moderate increase of the condenser inlet temperature during the time the compressor is operative. Calculations excluding the effect of the stand-by period did not affect the results of partial load factor substantially, thus indicating that the stand-by losses for the heat pump under study are not significant for PLR>0.2 (also shown in Figure 8 for the 1000 L case). This is because the stand-by power consumption is 15 W, which represents a 0.2% of the nominal electrical power consumption of the heat pump. However, only the losses related to the heat pump control operation were considered in this study. Additional losses related to the parasitic energy consumption by circulating pumps will increase the efficiency degradation. A close look at the transient test data during a cycle (Figure 3) shows an efficiency loss during the start-up period, which results from the heat release rate reaching its maximum value after 42 - 60 s from the start-up onset, while the electrical power consumption reaches its full value in less than 20 s. This behaviour was observed at all conditions of load and inertia levels, proving that start-up losses are of significance for this water-to-water heat pump.

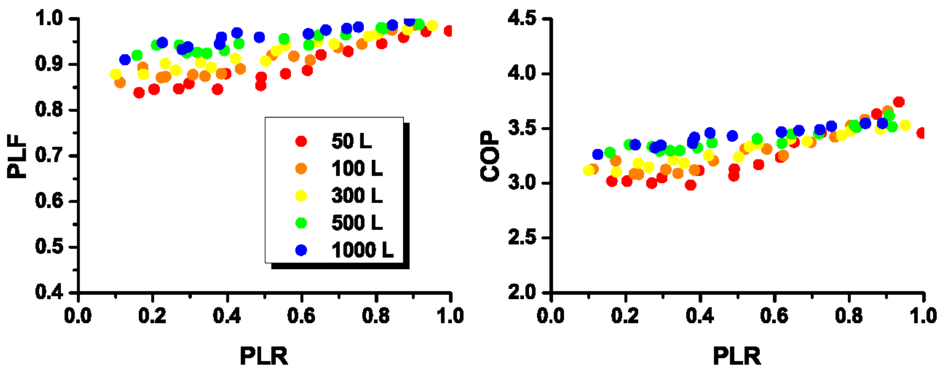

Figure 2: partial load factor (left) and coefficient of performance (right) as a function of partial load ratio for different inertia conditions.

3. Efficiency analysis and comparison of part load performance estimation methods

3.1. Comparison of part load performance estimations by technical standards

The experimental data obtained was compared with partial load factor predictions from parameterizations in the European EN14825, American ARI 210/240, and Italian UNI 10963 technical standards [10-12]. These standards are based on the estimation of the partial load factor by means of correlations that depend on degradation coefficients (C_d: start-up degradation coefficient; C_c: stand-by degradation coefficient). The results from the 50 l storage tank experiments in this study were used for comparing the different parameterizations from standards, as shown in Figure 4. The experimental data follows a linear trend with a loss in the energy efficiency as the partial load ratio decreases, similarly to the profile predicted by the ARI standard equation, with a C_d coefficient equal to 0.22:

$$PLF = 1 - C_d(1 - PLR) \quad (1)$$

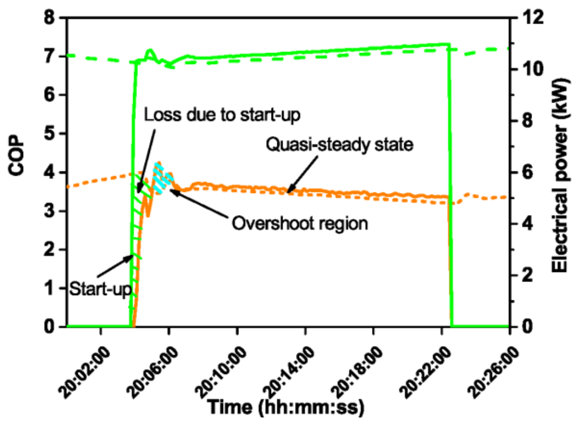

Figure 3: heat pump start-up behaviour for 1000 L storage tank experiment. Orange: COP, Green: electrical power, dashed line: prediction from steady state performance.

The method proposed in the Italian standard UNI 10963 [6,12] is based on the so-called Z parameter, which is the ratio between the electrical energy consumption at cycling and the electrical consumption at steady state conditions. The equation for calculating PLF in this standard is given the following expression:

$$PLF = \frac{PLR}{Z} = \frac{PLR}{a\ PLR + b} \quad (2)$$

Using the UNI 10963 approach, it results in a significant overestimation of the partial load factor with respect to the experimental data, as shown in Figure 4. The relationship between Z and partial load ratio was fitted to a polynomial equation with a correlation factor of R^2>0.99 for all the experiments performed. This equation was used to extrapolate the experimental data to partial load ratios below 0.2 and it has also been represented in Figure 4 with a blue line, in order to obtain a complete picture of the dependence of PLF on PLR.

The partial load factor correction given by the standard EN14825 [10] is defined as:

$$PLF = \frac{PLR}{C_c PLR + (1 - C_c)} \tag{3}$$

, with C_c given by:

$$C_c = 1 - \frac{P_{stand-by}}{P_{full\,load}} \tag{4}$$

where $P_{stand\text{-}by}$ is the electrical power consumption at stand-by conditions and $P_{full\ load}$ is the electrical energy consumed when the heat pump operates at full load. The values of partial load factor obtained with equation 3 with C_c=0.9 and C_c=0.998, deviate from the experimental results and the line that was fitted to experiments to cover the whole PLR range. In particular, a significant overestimation of performance is obtained when using the default coefficient value for C_c. Figure 4 indicates that the current technical standards are not able to fully represent the relationship of the partial load factor and load ratio for the heat pump under study, since these standards assume the existence of stand-by or start-up parasitic effects in isolation, while in this study the simultaneous occurrence of both effects has been observed.

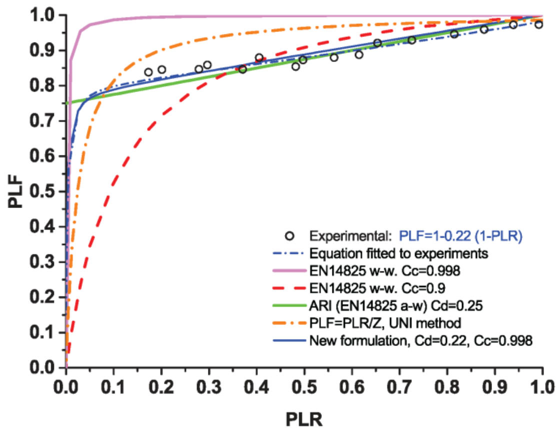

Figure 4: Comparison of experimental data and predictions of partial load factor (PLF) from parameterizations in technical standards. Case of 50 L storage inertia conditions.

A recent parameterization derived from the definition of the partial load factor for water-to-water heat pumps has been shown to better predict the partial load factor that existing correlations [18]. This parameterization, which accounts for both stand-by and start-up losses is given as:

$$PLF = \frac{1}{1 + \frac{C_d(1-PLR)}{1-C_d(1-PLR)} + (1-C_c)\frac{1-PLR}{PLR}} \tag{5}$$

The above equation reduces to expression (3) for negligible start-up efficiency losses (C_d=0), while it reduces to formula (1) for negligible stand-by losses (C_c=1). The results from using this alternative parameterization are represented in Figure 4 with coefficient values of C_d=0.22 and C_c=0.998 as determined experimentally from the 50 l storage tank data experiment. This correlation (new formulation in legend of Figure 4) is shown to closely reproduce the partial load factor expected for the heat pump under study, when compared with the line fitted to experiments that covers the whole partial load ratio range. This latter correlation will be adopted as part of the new method proposed in the present study to determine the part load performance of heat pumps, as described in next section.

3.2. Novel experimental reduced method to estimate the part load performance

Generally, current methods in technical standards are based on measurements of performance at full load with further correction to account for part load degradation effects, using the corresponding partial load factor correlation. The partial load factor equations depend on the degradation coefficients that are to be determined empirically or through the adoption of default values. However, as shown in the previous section, using coefficient default values or an inadequate correlation might lead to substantial error in the estimation of the efficiency degradation. On the other hand, determining the performance of heat pump units through experimentation by testing their behavior at different part load ratios to obtain the relationship between the partial load factor and the load ratio is time and effort consuming.

With the aim of optimising experimental methods for partial load testing of fixed-capacity units, the UNI standard correlation proposes a reduced experimentation methodology. This method is based on conducting a single experiment at part load to obtain the linear relationship between the Z parameter and PLR and then applying equation (2) to obtain the coefficients a and b that allow predicting the partial load factor. Although the method has been shown to work for a number of air-to-air and air-to-water systems, it has not been proved whether it is valid for all equipment and operating conditions [6].

A novel improved method is proposed in the present study which is based on the combination of reduced experimentation and the expression in equation (5) to obtain the specific values of the degradation coefficients. The method requires one measurement at stand-by conditions to determine the C_c factor as defined in the standard EN14825 and given by equation (4) and a single measurement at intermediate partial load ratio ($0.4<PLR<0.6$). Once C_c is known through the use of equation (4), the single partial load test data is applied to estimate the C_d coefficient value by means of equation (5). Once the two degradation coefficients C_c and C_d are determined, equation (5) can readily be used to estimate the performance of the equipment at part load.

The results obtained from using the UNI standard method described above and the methodology proposed in the present study are compared in Figure 5 for the cases with storage tanks of 50 L, 100 L and 1000 L, in comparison with the line fitted to the experimental data. The single experimental point at partial load chosen for applying the reduced method is shown in the figures as a single dot. The comparison between the results from applying the UNI method and the experimental line shows a close match for 1000 L storage; however, deviation becomes more important as the inertia decreases. This deviation is due to the fact that the correlation in the UNI standard is based on the assumption that Z follows a linear relationship with PLR; however, when this hypothesis is tested with the experimental data from this study it is found that the relationship between Z and PLR deviates from a linear curve as the inertia is reduced. This is seen as a drop of the R^2 linear correlation coefficient from a value of 0.999 for 1000 L to 0.986 for 50 L storage tank volume. Unlike the UNI method, the methodology proposed in this study gives a good match to the experimental data at any inertia conditions. This is because the proposed method accounts for the effect of inertia on both start-up losses and stand-by losses through the two degradation coefficients. The results also indicate that the simple methodology presented here allows determining the C_d coefficient from just a single part load experiment with enough accuracy to appropriately predict the start-up efficiency losses at any load conditions.

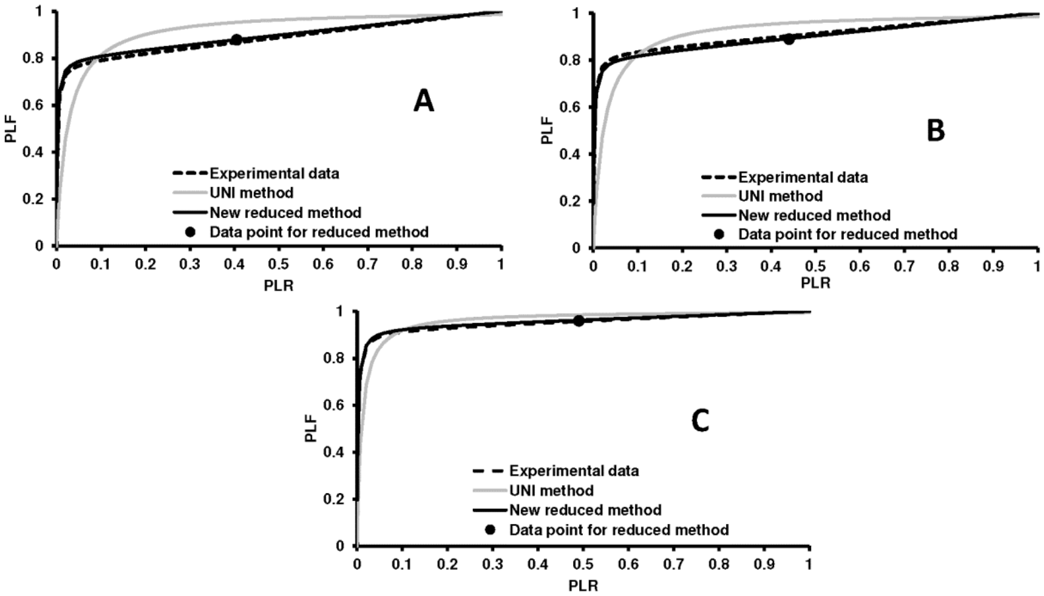

Figure 5: Predictions of partial load factor using the reduced method in UNI standard and the proposed new method for inertia conditions of (A) 50 L, (B) 100 L and (C) 1000 L. Equation fitted to exp. represents the experimental data in the whole load ratio range.

3.3. Improvements in the estimations of annual coefficient of performance

The calculation of the annual COP in the EN14825 standard is based on the the so-called bin method. This method is based on the integration of the energy produced and consumed during a year at three different climate design temperatures (-22°C (colder), -10°C (average) and 2°C (warmer)) considering a variable load profile throughout the year. In this section the bin calculation method is applied to evaluate the annual energy performance obtained when applying different methodologies in comparison with performance results determined from experiments. The annual COP (defined as SCOPnet in EN14825) is defined as the coefficient of performance resulting from dividing the annual thermal energy produced by the heat pump by its total yearly electrical energy consumption. Calculations were done for the three standard climatic conditions (colder, warmer and average) with application of 1) the UNI standard reduced experimentation methodology 2) the EN14825 standard partial load factor estimation method for C_c=0.9 and C_c=0.998 and 3) the methodology proposed in the present study as described in the previous section. The results from using these approaches are compared with the values obtained using experimental data on part load performance from this study.

Results in Figure 6 show that the estimation of the annual COP is sensitive to the inertia conditions. This implies that the energy performance is dependent not only on the equipment itself but also on the particular arrangement of heating equipment in a building. Hence, similar inertia conditions are required for comparison purposes between studies and standards. In contrast to this finding, the standard EN14825 does not make any considerations regarding the influence of the inertia conditions on the part load performance determination. Figure 6 shows that predictions using the EN14825 methods deviate from experiments in as much as 12%, depending on the inertia conditions. At 1000 L storage conditions the EN14825 standard prediction with C_c=0.998 leads to a small deviation of 3.5%, confirming that this method is reliable for inertia conditions high enough to yield start-up effects negligible. Similarly, the prediction from the UNI standard deviates from real performance as the inertia is reduced, with acceptable predictions only for the 1000 L storage tank case. On the other hand, the close match between the estimations from using the new method proposed in the present study and the experimental data shows a significant improvement in the estimation of the annual COP at any inertia conditions with respect to other methodologies evaluated here. Therefore, the reduced experimental method proposed in this study is recommended to better characterise the partial load behaviour of water-to-water heat pumps for comparison between equipment and for performance rating purposes.

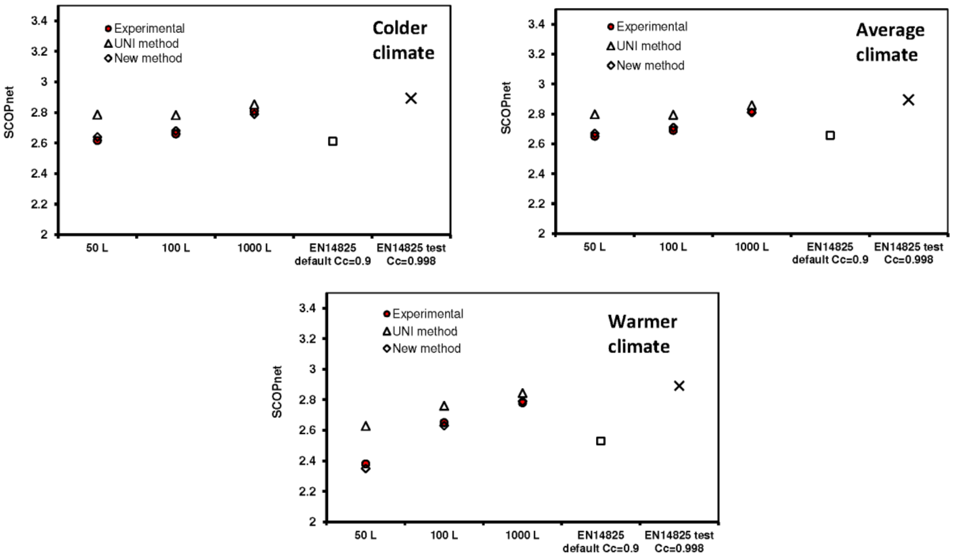

Figure 6: Calculations of heat pump annual COP (SCOPnet) derived with the bin integration method for colder, average and warmer climate, using different methods.

4. Water-to-water heat pump transient behavior model

Although the use of correlations is highly convenient when characterising and labeling the overall performance of heat pump systems, more sophisticated modeling approaches are necessary when the focus is modelling the transient real time performance behavior of heat pumps and the influence of their operation on the comfort of building users. The use of simulation models rather than parametrizations provides more in-detailed information on the real-time performance of heat pumps; however, these models should be able to account for the influence of parasitic behavior on equipment performance. The water-to-water heat pump tested in this study was modeled using a black-box approach based on the performance map obtained from laboratory experiments. In order to account for transient effects, the model was built by reproducing the start-up periods using a polynomial equation, while the quasi-steady state periods were modeled by means of correlations that allowed determining the thermal and electrical power as a function of the inlet temperatures to the condenser and the evaporator. The equations used for modeling the steady state behavior are the following:

$$\dot{Q}_{cond} = c + d\,T_{evap.in} + e\,T_{cond.in} \qquad (6)$$
$$\dot{Q}_{evap} = f + g\,T_{evap.in} + i\,T_{cond.in} \qquad (7)$$
$$\dot{P}_{elec} = j + k\,T_{cond.in} + l\,T_{cond.in}{}^{2} \qquad (8)$$

Where $\dot{Q}_{cond}$, $\dot{Q}_{evap}$ and $\dot{P}_{elec}$ are the heat rate at condenser and evaporator and electrical power consumption in kW, respectively, $T_{cond.in}$ and $T_{evap.in}$ are the inlet temperatures to the condenser and the evaporator and c, d, e, f, g, i, j, k and l are empirical coefficients obtained from fitting the equations to the experimental data. On the other hand, the start-up process is modeled using a sixth order polynomial equation as a function of time, which is given by:

$$\dot{Q}_{cond,start-up} = \dot{Q}_{cond}(b_1 t + b_2 t^2 + b_3 t^3 + b_4 t^4 + b_5 t^5 + b_6 t^6) \qquad (9)$$

Where $\dot{Q}_{cond,start-up}$ is the heat rate at condenser during start-up, $\dot{Q}_{cond}$ is the heat rate at steady-state at equivalent operating conditions, given by equation (6) and t is the time from start-up.

This heat pump model was implemented in the simulation software TRNSYS that was previously used for the laboratory experimentation, where the real heat pump was replaced by the model described above. In order to consider the inertial effects from the laboratory distribution system, two control volumes of 50 L each (total inertia of distribution system of 100 L) were placed at the outgoing and return lines of the condenser side of the heat pump in the simulation model. Results in Figure 7 illustrate the comparison of the transient behavior of the heat pump as obtained from simulations and laboratory experiments for a 1000 L hot storage tank, when considering and neglecting the inertial effects of the laboratory distribution system. Although the magnitude of the heat rate and temperatures is consistent for all the cases studied, the simulations and experimental data match rather closely when the distribution system inertial effects are accounted for, while a significant de-synchronization is obtained when inertial effects are neglected.

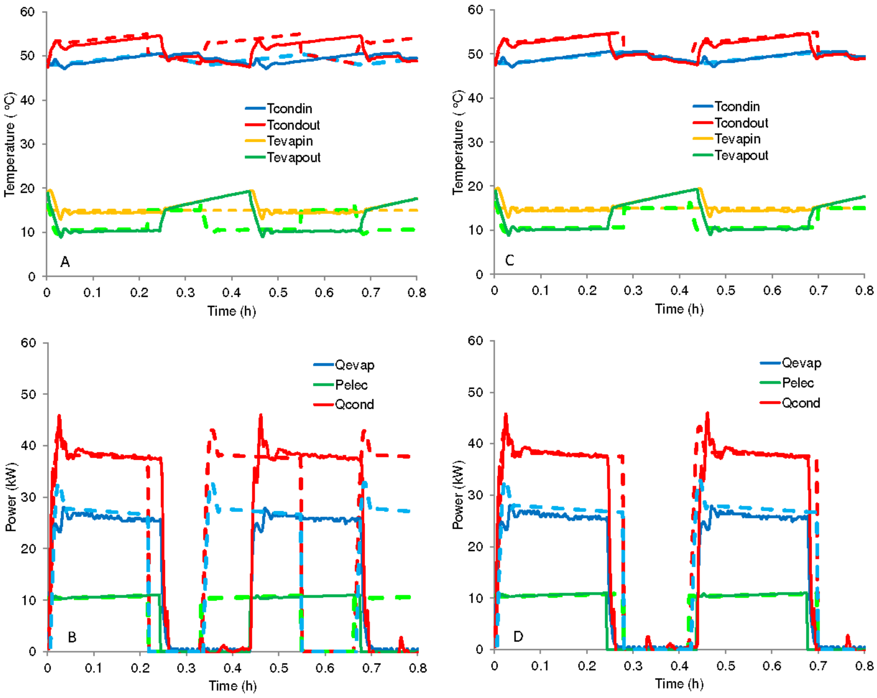

Figure 7: Comparison of simulated and experimental transient performance of a water-to-water heat pump operated at PLR=0.55 with a storage tank of 1000 L. Continuous line: experimental; Dashed line: model. Tcond.in=condenser inlet temperature;Tcond.out=condenser outlet temperature; Tevap.in=evaporator inlet temperature;Tevap.out=evaporator outlet temperature. A,B: without consideration of inertia from the laboratory distribution system, C,D: with consideration of inertial effects from the laboratory distribution system.

Further simulations were performed at different load conditions in order to obtain data of hourly COP versus partial load ratio, using the simulation model that accounted for all the inertial effects. Comparison of experimental and simulated COP values is presented in Figure 8. Results show that the experimental COP values are lower when the parasitic losses are accounted for, and that the simulation model is able to reproduce the transient performance of the heat pump, including all the parasitic losses. This demonstrates that a properly calibrated black-box model such as the one developed and tested here is accurate enough to simulate the behavior of a heat pump under transient conditions, including parasitic effects, without the need of using sophisticated heat pump models including detailed simulation of system components. The simple approach presented here is potentially useful to be applied in simulation studies aiming at optimizing the integration of heat pumps in energy systems in buildings, and in large scale simulation studies with multiple heat pumps integrated in district networks.

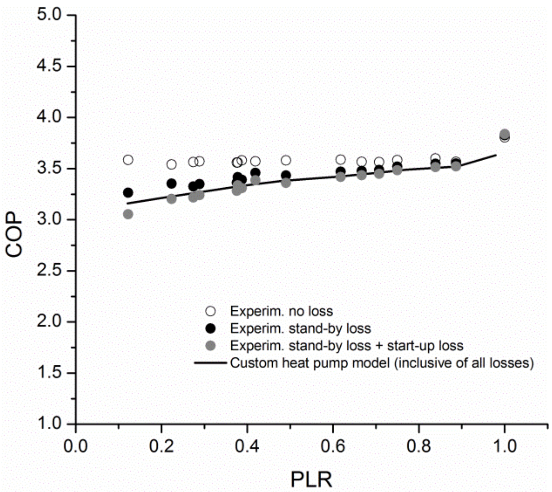

Figure 8: Experimental values of hourly COP for no parasitic losses, stand-by losses and stand-by + start-up losses in comparison with predictions from the black-box heat pump model. PLR=0.55.

5. Conclusions

Laboratory experiments with a water-to-water heat pump have been conducted in order to characterize the equipment behavior at part load conditions. Experiments were used to assess the validity of correlations in technical standards for predicting the part load performance of water-to-water heat pumps at different load ratios. A new method is proposed that is based on reduced experimentation and the use of a novel parameterization that accounts for both stand-by and start-up losses. Calculations of the annual coefficient of performance using different methods showed that the existing correlations and methodologies in the European EN14825 and Italian UNI 10963 standards are acceptable for predicting performance only when using empirical values for part load degradation coefficients for high inertia conditions. On the other hand, the new proposed method to characterise the part load performance of heat pumps presented in this study is proved to provide an improvement in the estimation of the annual coefficient of performance with high accuracy with respect to existing methods, including inertia effects and with a minimum number of experiments. In the study it is also proved that a simple empirical black-box model is valid for predicting the transient behavior of a fixed-capacity heat pump at part load and that this model is able to reproduce the parasitic losses induced from cycling operation.

Acknowledgments

Part of this work has been developed under the project TRIBUTE, co-founded by the EU-FP7 program, Grant Agreement 608790.

References

[1] Corberán, J. M. , Donadello, D., Martínez-Galván I., Montagud C. 2013 "Partialization losses of ON/OFF operation of water-to-water refrigeration/heat-pump units". International Journal of Refrigeration, 36 (8) 2251–2261.

[2] Safa AA, Fung AS, Kumar R. "Comparative thermal performances of a ground source heat pump and a variable capacity air source heat pump systems for sustainable houses." Appl Therm Eng 2015;81: 279–87.

[3] Man Y, Yang H, Wang J, Fang Z. In situ operation performance test of ground coupled heat pump system for cooling and heating provision in temperate zone. Applied Energy 2012;97:913–20.

[4] Montagud C, Corberán JM, Ruiz-Calvo F. Experimental and modeling analysis of a ground source heat pump system. Appl Energy 2013;109:328–36.

[5] Schibuola L, Massimiliano S, Chiara T. Modelling of HVAC system components for building dynamic simulation. In: 13th conference of international building performance simulation association, ChambTry, France; August 26–28, 2013.

[6] Bettanini E, Gastaldello A, Schibuola L. "Simplified models to simulate part load performances of air conditioning equipments." In: 8th international IBPSA conference, Eindhoven, Netherland; 2003.

[7] Riviere P, Malaspina NF, Lebreton JM. A new installation for part load testing of air to water single stage chillers and heat pumps. In: International refrigeration and air conditioning conference at Purdue; July 12–15, 2004.

[8] Madani, H. Claesson, J., Lundqvist P., 2013 "A descriptive and comparative analysis of three common control techniques for an on/off controlled Ground Source HeatPump (GSHP) system", Energy Build.,6,1-9.

[9] Waddicor, D.A, Fuentes, E., Azar, M., Salom J. , Partial load efficiency degradation of a water-to-water heat pump under fixed set-point control. Applied Thermal Engineering, 106, 275–285, 2016

[10] EN 14825:2013, "Air conditioners, liquid chilling packages and heat pumps, with electrically driven compressors, for space heating and cooling. Testing and rating at part load conditions and calculation of seasonal performance."

[11] ARI standard 210/240, 1989. "Unitary air-conditioning and air-source heat pump equipment", Air Conditioning & Refrigeration Institute, Arlington, Virginia.

[12] UNI standard 10963 "Air conditioners, chillers and heat pumps. Determination of the part load performances"

[13] Ruschenburg J., Tomislav, C., Herkel, S., 2014. "Validation of a black-box heat pump simulation model by means of field test results from five installations", Energy and Buildings, 84, 506–515.

[14] Klein, TRNSYS 17, 2010. "A Transient System Simulation Program", Solar Energy Laboratory, University of Wisconsin, Madison, USA.

[15] TESS library 17 04, 2010. LLC 22 North Carroll Street Suite 370 Madison, WI 53703 U.S.A, 2010.
[16] Allard, B. M., Kummert M, 2011. “Intermodel comparison and experimental validation of electrical water heaters in TRNSYS”. 12th Conference of International Building Performance Simulation Association.
[17] Kleinbach K. S., Beckman E.M., 1993. “Performance study of one-dimensional model for stratified thermal storage tanks”, Solar Energy 50 (2) 155–166.
[18] E. Fuentes, D.A. Waddicor, J. Salom, “Improvements in the characterization of the efficiency degradation of water-to-water heat pumps under cyclic conditions.” Applied Energy, 179, 778-789, 2016