Partialization losses of ON/OFF operation of water-to-water refrigeration/heat-pump units

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ABSTRACT

This paper presents the results of an experimental campaign for the characterization of the dynamic behavior of a water-to-water refrigeration/heat-pump unit under ON/OFF operation. The unit was previously tested at different water inlet temperatures under steady state conditions, and a very good agreement was found between the instantaneous dynamic performance of the heat pump and the corresponding quasi-steady state operation. In parallel, a series of tests were carried out to quantify the coefficient of performance (COP) degradation as a function of the load ratio, and a simple formula for the Part Load Factor is presented. Results lead to the conclusion that the only non-negligible factor in the COP degradation is the stand-by electrical consumption during the OFF period, especially at low load ratios. Finally, it is concluded that the minimization of the stand-by consumption is a key point for the future improvement of the seasonal performance of water-to-water systems.

Keywords: part load operation, heat pump performance, partialization losses

Nomenclature

COP	coefficient of performance [-]
C_C	degradation coefficient
c_p	specific heat at constant pressure [$J \cdot kg^{-1} \cdot K^{-1}$]
E	energy [J]
HPA	heat pump application [-]
$\dot{m}$	mass flow rate [$kg \cdot s^{-1}$]
$\dot{W}$	power consumption [W]

PLF part load factor [-]

$\dot{Q}$ heat pump capacity [W]

$\dot{q}$ building thermal demand [W]

T temperature [ºC]

t time [s]

V volume [m^3]

Subscripts

cond condensation

cycle heat pump working cycle

db deadband

el electrical

exp experimental measurement

HP heat pump

i inlet

inst instantaneous

o outlet

off compressor switched off

on compressor switched on

ref1 necessary to reach the operating pressures

ref2 necessary to reach the operating temperatures

sb stand-by losses

st-up start-up losses

Superscripts

cond condenser

ev evaporator

st steady state conditions

Greek symbols

α load ratio

Δ difference

ρ density

1. INTRODUCTION

The actual consumption of heating and air-conditioning (A/C) systems depends not only on the refrigeration/heat pump unit consumption and the consumption of the auxiliaries, such as circulation pumps and fans, but also on the thermal losses of the distribution system, which ultimately depend on the control/operation strategy. The optimum design of these systems, as well as the optimization of their operation, requires knowledge of the performance of the main components under different operating conditions. Among these, the refrigeration/heat pump unit is the most important and the most difficult to characterize under dynamic conditions.

The performance of a refrigeration/heat pump unit not only depends on the source temperatures and flow rates, but also on the regulation system. The real operation of a water-to-water ground source heat pump system is dynamic, and the control system typically consists of a thermostat so that the compressor switches off when the temperature of the return water coming from the building and entering the heat pump (HP) goes above the temperature setting T_{set}. The compressor switches on again when the return water temperature decreases below (T_{set} -ΔT_{db}), where ΔT_{db} is the thermostat deadband. In classical heating applications with fan-coil terminals, the temperature setting is typically 50ºC; however, nowadays the trend is to employ lower values of T_{set} in order to reduce the temperature lift of the heat pump and thus reach a higher COP. Additionally, nowadays the systems even tend to reduce the T_{set} value during mild days when the thermal load of the building is much lower. This control strategy is normally called temperature compensation. As a result, the actual operation of the heat pump (HP) consists of a series of successive ON/OFF periods which imply successive start-ups of the compressor with the consequent alternative migration of the refrigerant from the high pressure side to the low pressure side and vice versa, as well as alternate heating up and cooling down processes of all the fluids and metal masses, followed by periods at more or less steady conditions in which the compressor is either switched ON or OFF. The average performance of the HP under actual operating conditions is therefore different from the steady state performance, showing a certain degradation which depends on the characteristics of the HP and the operating conditions.

This paper presents the results of an experimental campaign for the characterization of the dynamic behavior of a water-to-water HP unit under on/off operation. The tests were carried out in the laboratory on a test rig especially developed to simulate the on/off operation of the unit at partial load operating conditions applied either at the evaporator side or the condenser side. The main purpose of the experimental campaign was to compare the performance of the unit under dynamic operation with the performance corresponding to steady state conditions and assess the partialization losses produced by the on/off operation.

Parken et al. (1977) previously defined and studied the efficiency degradation of air conditioners, and some years later they presented experimental values of the Part Load Factor (Parken et al., 1985). This subject attracted considerable interest around those years, and notable works include Didion and Kelly (1979), Goldschmidt et al. (1980), Janssen et al. (1992), and O'Neal and Katipamula (1993). Most of these works were dedicated to air-to-air systems but some also to air to water units, see for instance Tassou and Votsis (1991).

The degradation of the performance of refrigeration equipment at partial loads has again become an important topic in recent years mainly due to its implications for the seasonal performance evaluation of the complete building HVAC system but also because of the interest in comparing different system concepts, for instance, ground source vs. air source or on/off vs. inverter driven. A lot of work has been done on the characterization of the seasonal performance of air-conditioning and heating equipment, including the estimation of the performance at part load. In this regard should be cited the work of Henderson et al. (2000), Schibuola (2000), Anglesio et al. (2001), Bettanini el al. (2003), Riviere et al. (2005), and Cecchinato (2010). The PhD work of Riviere (2004) contains a very detailed analysis of the operation of different systems and of their part load performance and an evaluation of the corresponding seasonal performance.

As mentioned above, most of the cited studies were concentrated on air-to-air and water-to-air systems. The dynamic performance of water-to-water systems and their part load degradation have been studied very little. The main reason for this is that the partialization losses of water-to-water units are normally much lower than those of other systems. This paper presents for the first time experimental results of the dynamic performance of a water-to-water unit as well as the characterization of its Part Load Factor. The study has focused on the classic thermostat ON/OFF control applied at the return temperature. Some other control methods such as Floating Hysteresis or Degree Minute, as studied by Madani (2012), do not represent a difference in the dynamic behavior of the heat pump, so the conclusions of this paper are applicable to all ON/OFF control techniques.

2. EXPERIMENTAL TEST RIG

The experimental test rig, shown in Figure 1, has been specifically developed for this study. It mainly consists of two hydraulic groups, simulating the internal and external loops of the system (building and ground). In each of these loops it is possible to impose a given water flow rate and a given thermal load or temperature. The tanks in the loops (100 liters each) provide the characteristic thermal inertia of this kind of system and have been sized to provide a reasonable minimum ON time for an adequate HP compressor oil return (around two minutes).

As it can be observed in Figure 1, there are two secondary brazed plate heat exchangers (element 1), which exchange heat with the water network of the lab (hot or cold water). The regulation of the flow entering the secondary heat exchangers is performed by a three-way motorized valve (element f) and an external by-pass. For a given water temperature and a given flow rate on exiting the heat pump condenser (element B), the controller of the three-way valve will modify the flow rate entering the secondary heat exchanger in order to keep a desired temperature difference ΔT across the heating loop (condenser side). Regarding the other loop, the controller of the three-way valve will modify the flow rate in order to keep a constant temperature at the inlet of the heat pump evaporator (element C). This regulation is done by means of a proportional integral derivative (PID) controller.

Inside each of the closed hydraulic circuits, the flow is measured by a coriolis flow meter and it can be regulated by varying the position of the valves located at the by-pass circuits (element 3).

The management and control of the heat pump are done by means of its own thermostat, and its operation will depend on the temperature setting, T_{set} and the deadband ΔT_{db}.

The installation is equipped with a complete monitoring and data acquisition system where the main parameters of the performance of the unit at the refrigerant and water circuits are measured and collected. Heat transfer at both evaporator and condenser are measured with very accurate Coriolis mass flow meters and inlet and outlet temperatures with RTD Pt100 , especially calibrated at the temperature range of the measurements. Compressor consumption is measured with an accurate 0.2% Wattmeter. Table 1 shows the required information about accuracy of the employed instrumentation. The uncertainty obtained in the calculation of the COP and the heating capacity of the HP results in ±0.1 and ±2.4% respectively.

As the intention of the experimental setup is to simulate the conditions of a ground source water-to-water heat pump providing hot water to a building, the hydraulic loop at the evaporator side (element C) will be adjusted to simulate the ground loop so that both the flow rate and the inlet water temperature are kept constant at conditions similar to the ones existing in this kind of system during the heating season, e.g., flow rate leading to an approximately 5K temperature drop across the evaporator and, for instance, 18ºC at the inlet of the evaporator (see more details in Montagud et al. (2011)). The other loop simulates the water distribution loop to the building, thus the water flow rate is kept constant at a value which provides an approximately 5K temperature increase across the condenser at design conditions and a temperature of, for instance, 45ºC at the inlet of the condenser (element B). During the tests, the flow rates are both kept constant at design values in the closed hydraulic circuits and the water inlet temperature at the evaporator is kept constant at the specified value (T_i) by means of the control of the external loop. In order to simulate the thermal load of the building, a specified ΔT is imposed by the control system of the internal loop between the inlet and outlet of the secondary heat exchanger. So, for higher building thermal loads, a higher ΔT will be considered and vice versa. As the heating capacity of the heat pump is higher than the imposed thermal load, the return water temperature increases during an ON period until the heat pump thermostat switches off the compressor when the set temperature is reached T_{set}. The flow rates are kept constant as well as the thermal load, so that during the following OFF period, the return water temperature decreases until the thermostat detects that the temperature has dropped beyond the selected deadband ΔT_{db} and switches ON the compressor again. With the described test rig and control system, the unit can be tested in the lab under conditions very similar to those encountered in a real building with the advantage of being able to periodically repeat the working cycles and therefore to accurately characterize the heat pump performance under dynamic conditions.

The refrigeration unit employed in this study is a water-to-water reversible refrigeration/heat pump unit of around 25 kW working with R410A. Table 2 describes the characteristics of the main components of the unit.

3. RESULTS

Figure 2 shows a sample of results for three different building loads at 18ºC at the evaporator inlet, a return temperature setting of 40ºC and a deadband of approximately 5K. The indicated load ratios are nominal. The results in fact correspond to 16%, 52% and 72% load, respectively. As explained in section 2, the thermal load of the building is regulated by means of a three-way valve that regulates the entering flow rate at the secondary heat exchangers (element 1 in Figure 1), so it is very difficult to reach such precision in the regulation of the valve that exactly fixes a 20%, 50% and 80% load ratio.

Figure 2 also includes, at the bottom of each graph, the compressor consumption in order to illustrate the ON/OFF cycles. The registered water temperature evolution is very similar to the observed at actual conditions in the building (Corberan et al., 2011a; Riviere, 2004), with the advantage that, in the lab, the working conditions can be maintained without any variation until the cycles are repeated more or less exactly, and the performance during the ON/OFF periods can be measured with a high accuracy.

Results show that the control system of the external loop (ground side) is able to keep the inlet water temperature to the evaporator almost constant during the tests and equal to the desired value (18ºC in the example). On the other hand, it can be observed that the inlet and outlet water temperatures at the evaporator and condenser follow almost parallel lines, which indicates that the heating and cooling capacities in the refrigerant circuit are almost constant during the test period. Figure 3 shows a zoom of Figure 2 for a 50% load.

Figure 3 shows that the heating up and cooling down periods appear almost immediately at the water side once the compressor is ON, which indicates that the start-up period is very short (see ‘a’ in the graph). In this regard, it should be pointed out that the sampling interval of the measurements is 10 seconds. As it can be observed, there is a delay between the start-up of the compressor and the beginning of the inlet water temperature heating up period (see ‘b’ in the graph). The same delay is observed between the compressor switch off and the beginning of the cooling down period of the inlet water temperature. This delay depends on the water thermal inertia of the system and the stratification in the buffer tank, which is characteristic of this type of installation.

Figure 4 shows an example of the recording of the high and low pressure refrigerant sides (in the case of 80% load), showing the synchronisation between the water temperature and the pressure readings at both the condenser and the evaporator. As it can be observed, there is an initial high pressure peak, which quite exactly follows the inlet water temperature evolution. At the low pressure side, there is quite a fast pressure drop at the evaporator, which is not exactly followed by the evaporator water outlet temperature. However, this transition is quite fast and the evaporation temperature rapidly becomes stable and follows the inlet water temperature variation. This pressures trend is associated to the refrigerant going towards saturation conditions corresponding to the surrounding water temperature thanks to very fast mass transfer phenomema. Figures 2 and 3 have been presented in such a way that it is possible to visually distinguish the relationship between the ON (t_{on}) and OFF (t_{off}) periods and the thermal load. Assuming a constant building thermal load $\dot{q}$ and a constant HP heating capacity $\dot{Q}$, both represented in Figure 1, the ON and OFF periods and the total cycle time can be calculated from the following expressions:

$$t_{off} = \frac{\rho V c_p \Delta T_{db}}{\alpha \dot{Q}} \tag{1}$$

$$t_{on} = \frac{\rho V c_p \Delta T_{db}}{\dot{Q}(1-\alpha)} \tag{2}$$

$$t_{off} + t_{on} = \frac{\rho V c_p \Delta T_{db}}{\dot{Q}} \left(\frac{1}{\alpha} + \frac{1}{1-\alpha} \right) \tag{3}$$

Where $\alpha = \dot{q}/\dot{Q}$ corresponds to the definition of the load ratio; $\rho V c_p$ represents the total water thermal inertia of the distribution loop; and ΔT_{db} is the thermostat deadband. The ON (t_{on}) and OFF (t_{off}) periods can therefore be represented vs. the load ratio α. Figure 5 shows the results of expressions (1) to (3) as a function of α. The volume of the inertia tank of the system has been selected so as to provide a minimum t_{on} value of 2 minutes in order to ensure a good oil return (Corberan, 2012).

As Figure 5 illustrates, in the case of constant building thermal load and HP capacity, the OFF period decreases monotonically with the load ratio, the ON period increases monotonically with the load ratio, and the cycle period shows a parabolic behavior with a minimum 50% load ratio. It should be pointed out that the characteristic ON and OFF periods present a hyperbolic behavior for load ratios higher than 75% and lower than 25%, respectively. Results presented in Table 3 show that the experimentally determined characteristic periods agree reasonably well with the theoretical ones presented in Figure 5, with a maximum deviation of around 12% for the zones where the behavior of the theoretical expressions is hyperbolic.

4. PERFORMANCE CHARACTERIZATION AT STEADY CONDITIONS

As the above figures show, when the HP is working (ON period) the inlet water temperature at the evaporator is almost constant, while the inlet water temperature at the condenser follows the evolution of the average water temperature of the water inside the hydraulic distribution loop, decreasing initially until the heating produced when the compressor switches on starts to elevate the return water temperature (this short period is the delay period mentioned above). After the delay period, the return water temperature increases monotonically until the thermostat switches off the compressor. The variation of the return temperature, which is the water temperature at the inlet of the condenser, is narrow and is approximately bounded by the return temperature setting and the thermostat deadband, just shifted by approximately 1K, which occurs in the delay period.

Therefore, in order to compare the performance of the HP during the actual operation with the performance corresponding to steady state conditions, it is necessary to characterize the steady performance of the HP under different water inlet temperatures at both the condenser and the evaporator. Figure 6 shows the experimentally determined heating capacity and the heat pump COP working at a fixed water inlet temperature of 18ºC in the evaporator and at different water inlet temperatures in the condenser in the range of those registered during the dynamic tests.

As it can be observed in Figure 6, the heating capacity of the HP is around 27 kW and the COP is around 4.2 under the described conditions. At constant water temperature conditions at the evaporator, an increase of the water inlet temperature at the condenser leads to an increase in the pressure ratio and consequently an increase of the compressor consumption and a reduction in the heating capacity. The variation of both performance parameters is almost linear. The decrease of the COP could be non-linear, but in the case of the present work, as the tested variation of the temperature is small, the variation of the COP turns out to be almost linear. The corresponding linear fitting lines are shown in the graphs and the expressions are shown on the right hand side at the top of each graph.

5. DYNAMIC PERFORMANCE

One of the objectives of the present study is to compare the actual dynamic performance of the HP with the performance assuming that the HP is working under steady state conditions at the instantaneous values of the inlet water temperatures in the evaporator and the condenser. This would imply the assumption that the inertia of the HP is very small compared to the other thermal inertias existing in the system, namely the water loop thermal inertia. This assumption is typically known as quasi-steadiness.

During this study, the registered performance of the system under dynamic conditions was compared to the instantaneous quasi-steady performance, which was estimated by means of the developed linear fits, and the coincidence was always very high. Expression (4) presents the equations used to calculate the instantaneous COP of the heat pump (dynamic performance) and the steady COP of the heat pump working at the same instantaneous values of inlet water temperatures and flow rates at the condenser and the evaporator.

$$COP_{inst} = \frac{\dot{m}_w c_p \left(T_o^{cond} - T_i^{cond}\right)}{\dot{W}_{HP}^{inst}} \;\; vs \;\; COP^{st} = \frac{\dot{Q}_{cond}^{st}\left(\dot{m}_w^{ev}, T_i^{ev}, \dot{m}_w^{cond}, T_i^{cond}\right)}{\dot{W}_{HP}^{st}\left(\dot{m}_w^{ev}, T_i^{ev}, \dot{m}_w^{cond}, T_i^{cond}\right)} \qquad (4)$$

As an example, Figure 7 shows a comparison of COPs for the three different conditions already presented in Figure 2.

As can be easily distinguished, the results are remarkably similar, supporting the conclusion that the quasi-steady assumption is accurate enough and indicating that the dynamic behavior can be analyzed as a continuous series of steady operation along the variation of the inlet water temperatures. Additionally, these results show that no special losses affect the HP performance due to the dynamic working conditions, or at least that they fall within the uncertainty of the measurements.

6. PERFORMANCE DETERIORATION AT PARTIAL LOADS

Another important objective of the present study was to experimentally assess the possible losses arising from the continuous ON/OFF intervals of the HP under actual operating conditions and also the weight of the parasitic losses during the OFF period on the average performance of the HP in comparison to steady operation of the HP.

The electrical consumption during one cycle ($t_{cycle} = t_{on} + t_{off}$) can be decomposed into the consumption during the ON period (t_{on}) and the consumption during the OFF period (t_{off}):

$$\int_0^{t_{cycle}} \dot{W}_{el}\; dt = \int_0^{t_{on}} \dot{W}_{el}\; dt + \int_0^{t_{off}} \dot{W}_{sb}\; dt \qquad (5)$$

The electrical power consumption during the OFF period is constant and corresponds to the standby losses ($\dot{W}_{sb}$), mainly caused in this kind of HP by the consumption of the electronics, activated solenoid valves and the crankcase heater.

The electrical consumption during the ON period for this kind of HP mainly consists of:

1. The electrical energy consumption during the start-up transient ($E_{st\text{-}up}$), which is composed of:
    - The electrical energy losses inherent in the electrical start-up process ($E_{el\ st\text{-}up}$)
    - The energy necessary to compress the refrigerant from the low pressure side to the high pressure side and reach adequate refrigerant pressures at both the condenser and the evaporator ($E_{ref1\ st\text{-}up}$)
    - The energy required to reach the operation temperatures: heating up of the compressor and the high pressure side of the HP and cooling down of the low pressure side: $E_{ref2\ st\text{-}up}$
2. The electrical consumption corresponding to the interval after the start-up period, which can be assumed under quasi-steady state conditions:

$$\int_{after\ st-up}^{t_{on}} \dot{W}_{after\ st-up}\ dt = \int_0^{t_{on}} \dot{W}_{HP}^{st}\left(\dot{m}_w^{ev}, T_i^{ev}, \dot{m}_w^{cond}, T_i^{cond}\right)\ dt \tag{6}$$

Therefore, the electrical power consumption during a complete cycle is mainly composed of the following terms:

$$\int_0^{t_{cycle}} \dot{W}_{el}\ dt = E_{el\ st-up} + E_{ref1\ st-up} + E_{ref2\ st-up} + \int_0^{t_{on}} \dot{W}_{HP}^{st}\left(\dot{m}_w^{ev}, T_i^{ev}, \dot{m}_w^{cond}, T_i^{cond}\right)\ dt + \int_0^{t_{off}} \dot{W}_{sb}\ dt \tag{7}$$

All three start-up energy losses, ($E_{el\ st-up} + E_{ref1\ st-up} + E_{ref2\ st-up}$) will mainly depend on the operating conditions and can be associated with the start-up period of the unit. These start-up losses will occur once during every ON period, so that for cyclic operating conditions they remain constant independently of the cycle time, i.e., of the load ratio.

From the experimental results (see Figure 3) it can be easily concluded that the start-up period is very short for this kind of HP, as the temperatures evolve very quickly to the corresponding steady state conditions. In this regard, it is important to notice that the energy employed to compress the refrigerant and reach the steady state temperatures in water-to-water units is very small, as the water loops keep the refrigerant temperatures and pressures very close to the steady-state values during the OFF-period, as can be easily distinguished in Figure 4. It should be pointed out that the tested unit has a solenoid valve upstream of the expansion valve which closes when the compressor switches off, thereby avoiding the liquid refrigerant migration into the low pressure side of the unit.

On the other hand, the heat produced at the condenser and transferred to the water loop during one ON/OF cycle can be estimated by:

$$\int_{0}^{t_{cycle}} \dot{Q}_{heating}\ dt = \int_{0}^{t_{cycle}} \left(\dot{Q}_{cond} - \dot{Q}_{losses}\right) dt \tag{8}$$

The total heat transferred to the water will be the total heat generated by the condensation of the refrigerant minus the heat transfer losses to the surroundings from the outer surface of the condenser. Anyhow, the condenser of the unit is well insulated so that the losses are negligible in regard to the heat transferred. It should be noted that there is a diphase between the heat generated by the condensation of the refrigerant and the heat transferred to the water which is due to the thermal inertia of the metal mass of the condenser; however, this diphase does not imply any energy losses, since the heat which is first applied to heating up the walls above the water temperature at the start of the ON period, is recovered afterwards along the beginning of the OFF period when the water inlet temperature starts to drop and the walls are still hotter.

Therefore, assuming negligible losses and that the heat generated at the condenser at any instant is equivalent to the one generated under steady state conditions for the same instantaneous values of inlet water temperatures and flow rates, the total heat transferred to the water can be estimated by Expression (9):

$$\int_{0}^{t_{cycle}} \dot{Q}_{heating}\ dt = \int_{0}^{t_{on}} \dot{Q}_{cond}^{st}\left(\dot{m}_{w}^{ev}, T_{i}^{ev}, \dot{m}_{w}^{cond}, T_{i}^{cond}\right) dt \tag{9}$$

Consequently, the heating COP of the HP along one single ON/OFF cycle can be estimated as indicated in Expression (10):

$$COP_{heating}=\frac{\int_0^{t_{cycle}}\dot{Q}_{heating}\ dt}{\int_0^{t_{cycle}}\dot{W}_{el}\ dt}=\frac{\int_0^{t_{on}}\dot{Q}^{st}_{cond}\left(\dot{m}^{ev}_{w},T^{ev}_{i},\dot{m}^{cond}_{w},T^{cond}_{i}\right)dt}{E_{st-up}+\int_0^{t_{on}}\dot{W}^{st}_{HP}\left(\dot{m}^{ev}_{w},T^{ev}_{i},\dot{m}^{cond}_{w},T^{cond}_{i}\right)dt+\dot{W}_{sb}\cdot t_{off}} \tag{10}$$

Considering now that the steady performance is almost linear for small variations of the operating conditions, as shown in Figure 6, the heating COP can be determined by Expression (11):

$$COP_{heating}=\frac{\overline{\dot{Q}^{st}_{cond}}\Big/\overline{\dot{W}^{st}_{HP}}}{1+\frac{E_{st-up}}{\dot{W}^{st}_{HP}t_{on}}+\frac{\dot{W}_{sb}}{\dot{W}^{st}_{HP}}\left(t_{off}\Big/t_{on}\right)}=\frac{COP^{st}}{1+\frac{E_{st-up}}{\dot{W}^{st}_{HP}t_{on}}+\frac{\dot{W}_{sb}}{\dot{W}^{st}_{HP}}\left(t_{off}\Big/t_{on}\right)} \tag{11}$$

Where $\overline{\dot{Q}^{st}_{cond}}$ and $\overline{\dot{W}^{st}_{HP}}$ are the mean values during the ON period or, taking into account their linear behavior, the steady performance values corresponding to the mean value of the condenser inlet water temperature, which can be estimated approximately by: $T_{set}-\frac{\Delta T_{db}}{2}$. In order to simplify the nomenclature, here and after, $\dot{W}^{st}_{HP}$ and COP^{st} will be the performance parameters corresponding to the heat pump working at steady state conditions with a condenser inlet temperature of $T_{set}-\frac{\Delta T_{db}}{2}$. Dividing by $\dot{W}^{st}_{HP}$ and employing expressions (1) to (3), the relationship between the dynamic coefficient of performance in heating mode, $COP_{heating}$, and the steady one, COP^{st}, as a function of the load ratio is obtained in expression (12):

$$PLF=\frac{COP_{heating}}{COP^{st}}=\frac{1}{1+\frac{E_{st-up}}{\dot{W}^{st}_{HP}t_{on}}+\frac{\dot{W}_{sb}}{\dot{W}^{st}_{HP}}\left(t_{off}\Big/t_{on}\right)}$$

$$PLF=\frac{COP_{heating}}{COP^{st}}=\frac{1}{1+\frac{E_{st-up}COP^{st}\left(1-\alpha\right)}{\rho Vc_{p}\Delta T_{db}}+\frac{\dot{W}_{sb}}{\dot{W}^{st}_{HP}}\frac{\left(1-\alpha\right)}{\alpha}} \tag{12}$$

This relationship between the dynamic $COP_{heating}$ and the steady one represents the efficiency degradation and is frequently known as the Part Load Factor (PLF). As can be observed, it is mainly a function of the load ratio, $\alpha = \dot{q}/\dot{Q}$. The first term in the denominator represents the influence of the start-up losses, while the second one represents the influence of the stand-by consumption during the OFF period. As can be seen, the influence of the start-up losses depends on the hydraulic loop energy storage ($\rho V c_p \Delta T_{db}$). This is simply because the higher the thermal mass of water and the deadband, the longer the minimum ON period and, therefore, the lesser the influence of the start-up losses.

Looking at expression (12), it should be pointed out that, for systems with a very high water volume in the distribution loop, the second term in the denominator can be negligible; that is to say, the start-up losses would have a negligible influence and expression (12) would result in expression (13).

$$PLF = \frac{COP_{heating}}{COP^{st}} = \frac{1}{1 + \frac{\dot{W}_{sb}}{\dot{W}_{HP}^{st}} \frac{(1-\alpha)}{\alpha}} \tag{13}$$

In order to be able to estimate the start-up losses of the studied heat pump, a dedicated series of tests was performed at different load ratios for two different deadband values: 3K and 5K. Separately, the stand-by losses were measured and turned out to be 60 W, approximately. Figure 8 shows the measured results of the COP degradation with the corresponding error bars and the theoretical results (represented by a continuous line) obtained by considering in expression (12) a value of 0 kJ for the start-up energy losses: E_{st-up}. As it can be observed, the agreement of results with expression (13) is good. So, just considering the measured stand-by losses is enough to depict well the measured performance. Therefore, it can be concluded that start-up losses in this kind of HP and system are negligible; therefore the stand-by losses of the HP are the only losses to be taken into account for the estimation of the degradation of performance at partial load operation.

As shown in Figure 8, the losses during the OFF period are the main reason for the COP degradation at low load ratios; therefore, it is of great importance to keep those losses as low as possible in order to reach high seasonal performance factors. The 60 W stand-by losses approximately represent 0.9% of the average compressor consumption, $\left(\dot{W}_{sb}^{st}/\dot{W}_{HP}^{st}=0.009\right)$. It should be pointed out that the fact that the start-up losses E_{st-up} have a negligible influence on the dynamic COP is twofold: on the one side, they are small in magnitude in this kind of HP, and on the other side, as expression (12) indicates, their effect is enhanced by the reduction of the time ON; however, the minimum value of the time ON is limited by the thermal inertia of the water loop and is typically around 2–3 minutes. In contrast, as shown in expression (12), the influence of the stand-by losses depends on the ratio between the OFF period and the ON period and this ratio increases asymptotically at low thermal loads. That is the reason for the observed fast degradation at low load ratios.

The stand-by losses are mainly caused by the consumption of the electronics, the consumption of the valves (kept energized during the OFF period), and the consumption of the crankcase heater. These last two consumptions should be critically reviewed and solutions with null or low energy consumption should be adopted. The great importance of the consumption of the water circulation pumps on the seasonal performance of the complete system should also be emphasized, as it is also necessary to minimize this consumption. The optimization of the design and operation of all these components becomes a key point for the future improvement of water-to-water systems (see more details in Corberan et al. (2011b)).

In regard to other previous studies on partialization losses, Anglesio et al. (2001) were the first to analyze the effect of partialization losses on the degradation of the Part Load Factor. After carrying out an experimental study where the total power consumption of the heat pump (stand-by losses and the heat pump consumption) were measured at different load ratios, it was observed that there was a linear relationship between the heat pump consumption for a given load ratio and the heat pump consumption at 100% load $(\alpha=1)$, as indicated by expression (14):

$$\frac{\dot{W}_{HP}(\alpha)}{\dot{W}_{HP}(\alpha=1)}=C_C\cdot\alpha+b \qquad (14)$$

The PLF defined by Anglesio et al. resulted in expression (15):

$$PLF = \frac{\alpha}{\left(1 - \frac{\dot{W}_{sb}}{\dot{W}_{HP}^{st}}\right) \cdot \alpha + \frac{\dot{W}_{sb}}{\dot{W}_{HP}^{st}}} \quad (15)$$

which can also be written as:

$$PLF = \frac{1}{1 + \frac{\dot{W}_{sb}}{\dot{W}_{HP}^{st}} \cdot \frac{(1-\alpha)}{\alpha}} \quad (16)$$

These results are considered in the present draft of EN 14825:2010, concerning the rating at part load conditions of refrigeration and heat pump equipment, which proposes in section 6.4.2.1 the following expression (17) for the evaluation of the PLF:

$$PLF = \frac{\alpha}{C_C \alpha + (1 - C_C)} \quad (17)$$

where α is the load ratio and C_C is the degradation coefficient.

As it is stated in section 8.5.2.1 of the norm, for air-to-water and water-to-water units, C_C due to the pressure equalization effect when the unit starts up can be considered negligible and therefore the only effect impacting EER/COP at part load operation is the effect of the stand-by losses when the compressor is switched off, and these losses must be measured when the compressor is off for at least 10 minutes. Thus, by applying expression (14) for a load ratio of 0%, ‘b’ will correspond to the stand-by losses of the heat pump, $\dot{W}_{sb}$, divided by the heat pump power consumption at 100% load, which is coincident with the consumption of the heat pump at steady state conditions, $\dot{W}_{HP}^{st}$. Therefore, C_C can be calculated for a load ratio of 100%, as indicated in expression (18):

$$C_C = 1 - b = 1 - \frac{\dot{W}_{sb}}{\dot{W}_{HP}^{st}} \quad (18)$$

Taking into consideration the definition of C_C , the Part Load Factor can be written as expression (19):

$$PLF = \frac{1}{1 + \frac{\dot{W}_{sb}}{\dot{W}_{HP}^{st}} \frac{(1-\alpha)}{\alpha}} \tag{19}$$

which is coincident with expression (13) analytically obtained in the present work where the start-up energy losses can be considered negligible.

It should be pointed out that the formula developed by Anglesio et al. and the one considered in the draft of EN 14825:2010 were empirically obtained, whereas the proposed formula of the present work was analytically developed taking into consideration the thermo-physical phenomena that take place during the ON/OFF operation of a water-to-water heat pump unit . Furthermore, it can be concluded from this section that the expression (12) developed in the present work is more general than the one proposed by Anglesio et al. or the one employed in the draft of EN 14825:2010, as it also takes into account the effect of the start-up losses on the degradation of the PLF. Even if these losses are negligible for water-to water heat pumps, they can be considerable in other applications such as fridges or other systems working with capillary tubes where the start-up losses (pressure equalization effect) and their effect should also be accounted for.

7. CONCLUSIONS

The main conclusions that can be drawn from the performed study on the dynamic behavior of a water-to-water heat pump unit –are the following:

- The dynamic performance of the HP is almost quasi-steady, i.e., the HP can be considered as working at each instant under steady conditions at the instantaneous values of the inlet water temperatures to the evaporator and condenser.
- The instantaneous quasi-steady COP agrees very well with the instantaneous dynamic COP.
- The start-up periods are very short and negligible compared to typical ON periods of 2–3 minutes.

- A new formula for the evaluation of the Part Load Factor including the start-up losses has been proposed. However, the experimental results prove that these losses are very small on water-to-water systems, so their effect on the formula can be neglected. In that situation, the developed formula becomes exactly the same as the one proposed by Anglesio et al. and the one employed in En14825:2010 for air-to-water and water-to-water units. Nevertheless, the proposed formula is more general as it is also valid for those applications such as fridges or other systems working with capillary tubes where the start-up losses are not negligible.
- The 60 W measured stand-by losses approximately represent 0.9% of the average consumption of the compressor, so they are negligible when the compressor is switched on. However, the stand-by losses produce a high deterioration of the Part Load Factor at low loads (below 20%).
- The stand-by losses are mainly caused by the consumption of the electronics, the consumption of the valves (energized during the OFF period), and the consumption of the crankcase heater. These last two consumptions should be critically reviewed and solutions with null or low energy consumption adopted.

8. ACKNOWLEDGMENTS

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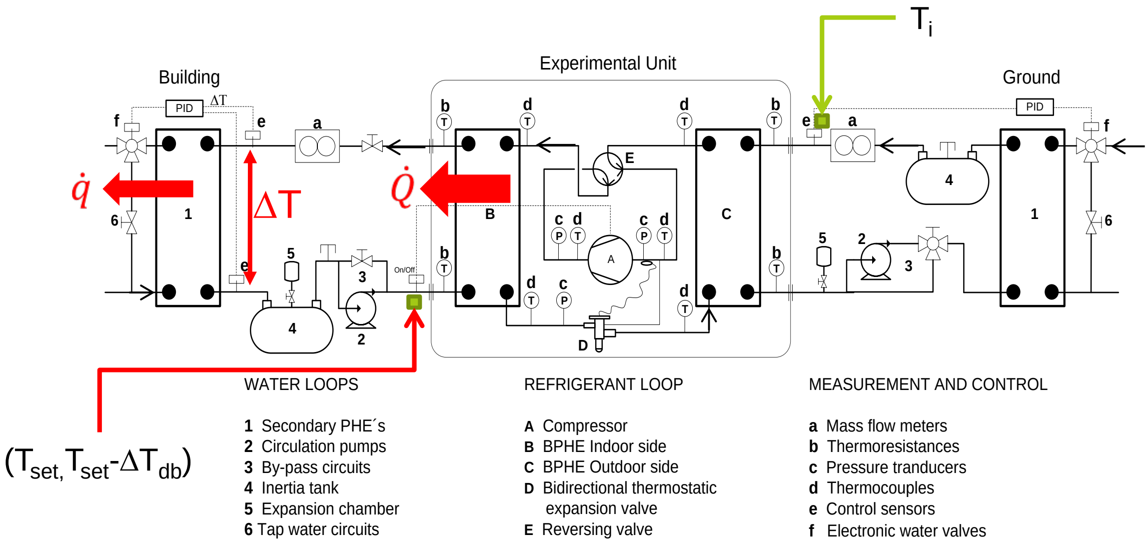
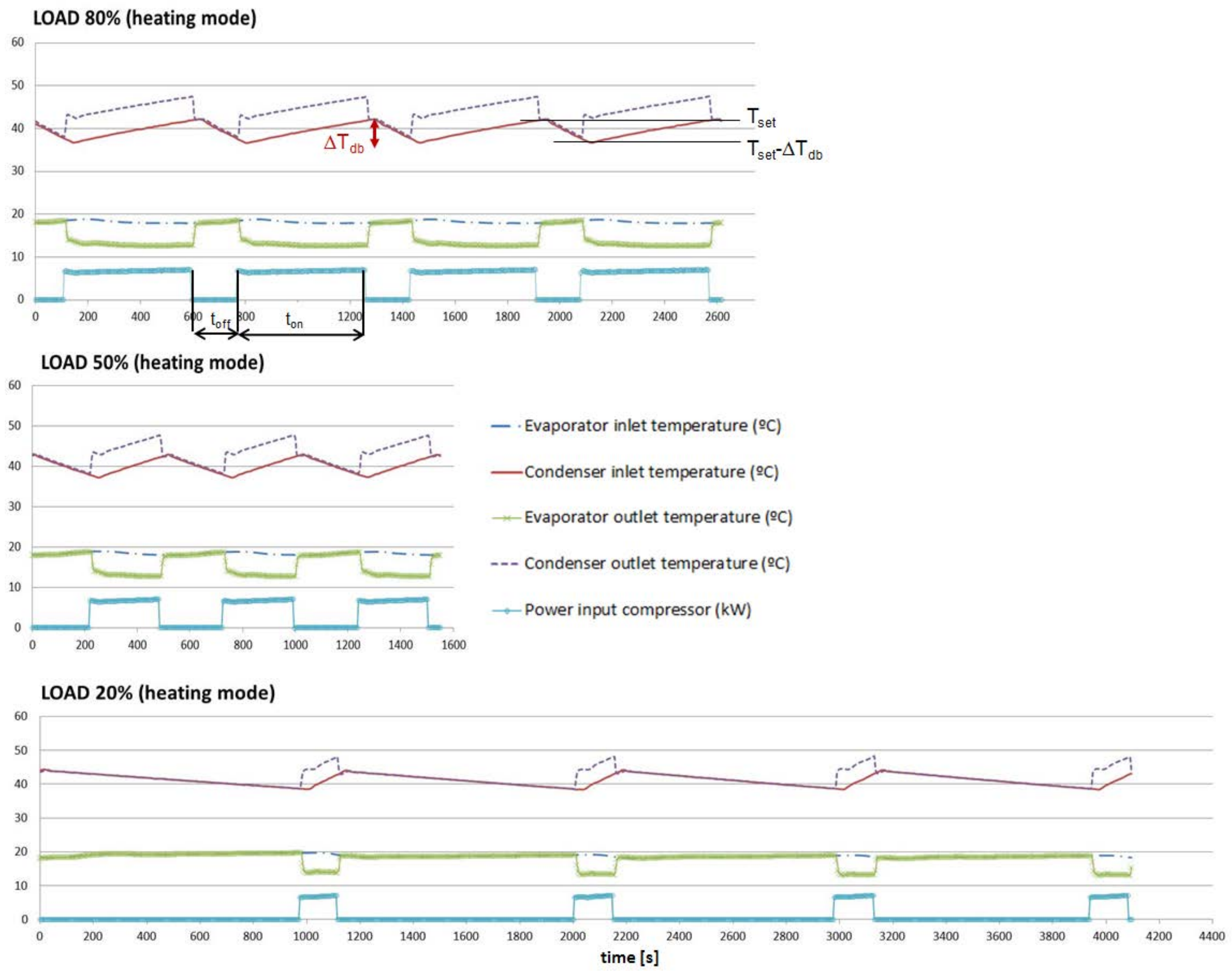
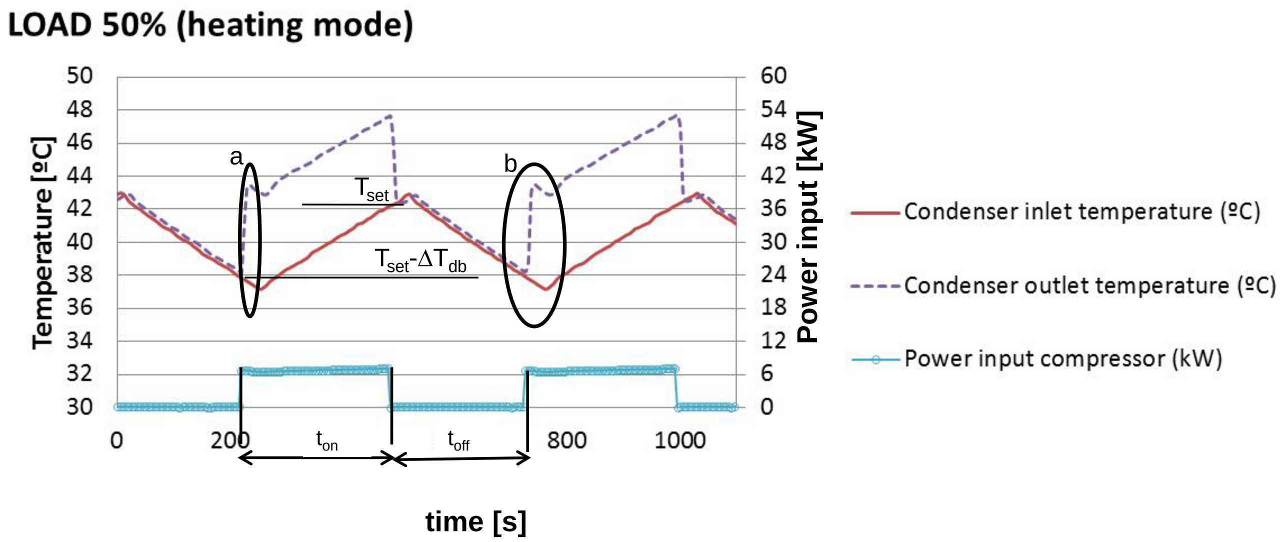


T_i
Building
Experimental Unit
Ground
PID
ΔT
ΔT
(T_set, T_set-ΔT_db)
On/Off
WATER LOOPS
1 Secondary PHE´s
2 Circulation pumps
3 By-pass circuits
4 Inertia tank
5 Expansion chamber
6 Tap water circuits
REFRIGERANT LOOP
A Compressor
B BPHE Indoor side
C BPHE Outdoor side
D Bidirectional thermostatic expansion valve
E Reversing valve
MEASUREMENT AND CONTROL
a Mass flow meters
b Thermoresistances
c Pressure tranducers
d Thermocouples
e Control sensors
f Electronic water valves

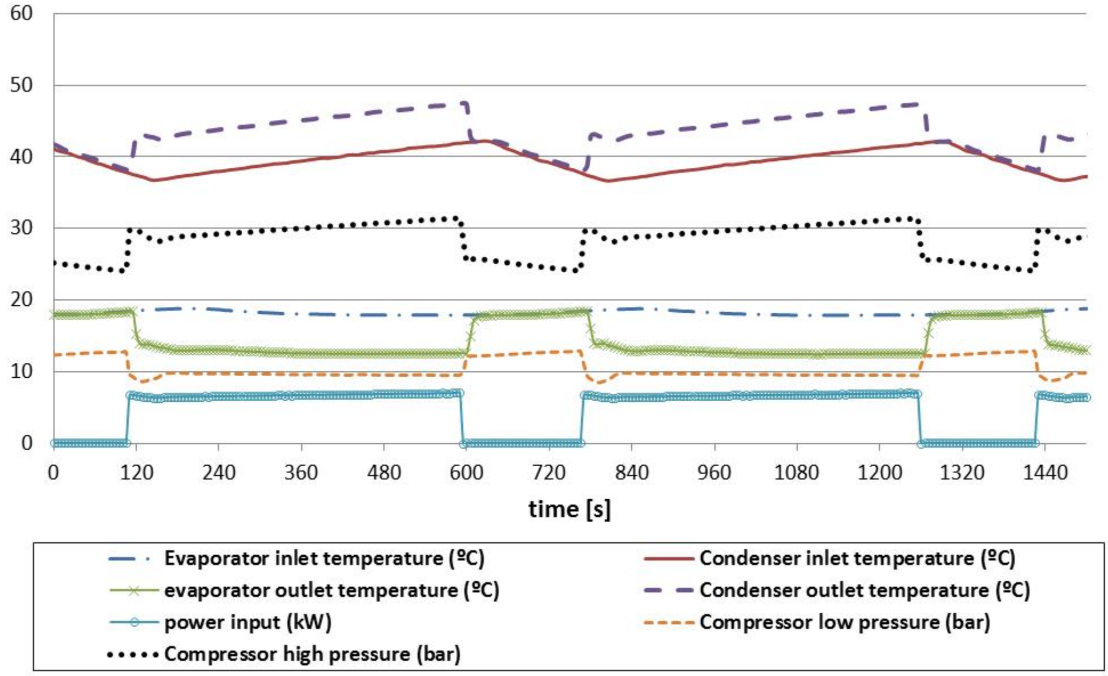


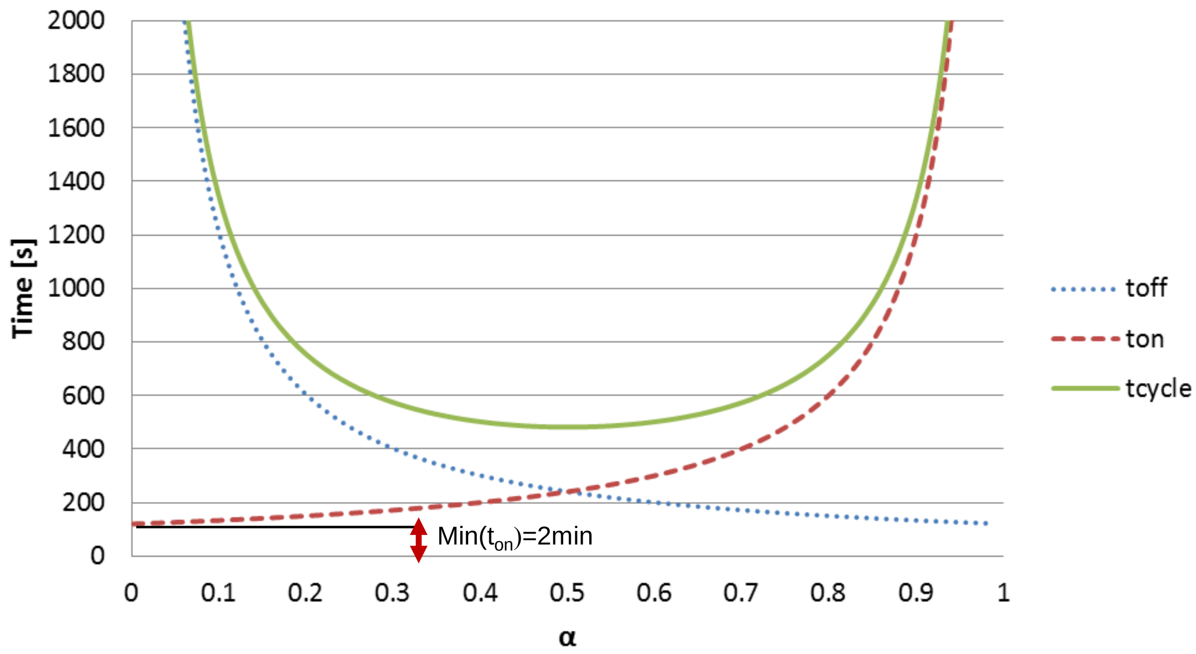
LOAD 80% (heating mode)
T_set
T_set-ΔT_db
ΔT_db
t_off
t_on
LOAD 50% (heating mode)
Evaporator inlet temperature (ºC)
Condenser inlet temperature (ºC)
Evaporator outlet temperature (ºC)
Condenser outlet temperature (ºC)
Power input compressor (kW)
LOAD 20% (heating mode)
time [s]



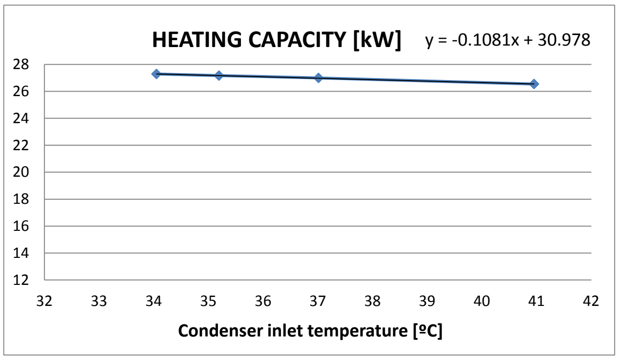
LOAD 50% (heating mode)
Temperature [ºC]
Power input [kW]
time [s]
a
b
Tset
Tset-ΔTdb
ton
toff
50
48
46
44
42
40
38
36
34
32
30
60
54
48
42
36
30
24
18
12
6
0
0
200
800
1000
Condenser inlet temperature (ºC)
Condenser outlet temperature (ºC)
Power input compressor (kW)

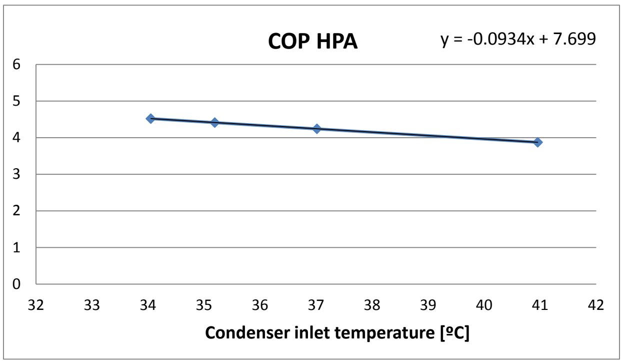
LOAD 80% (heating mode)

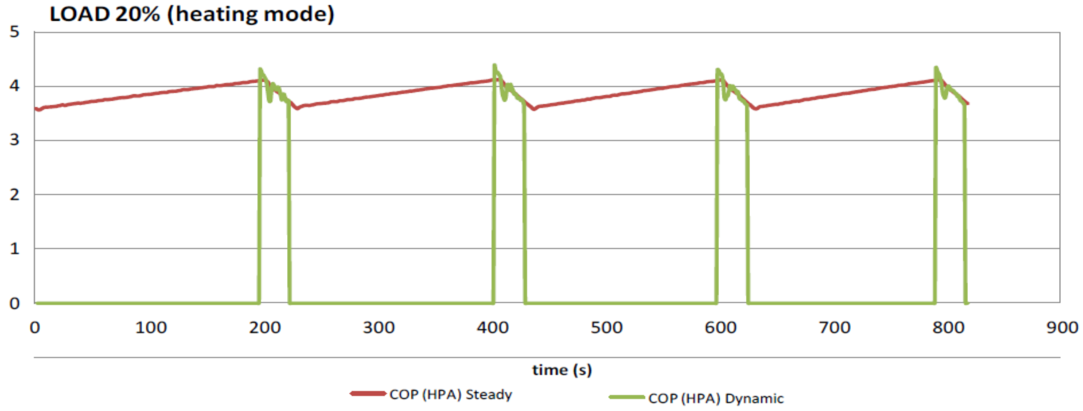


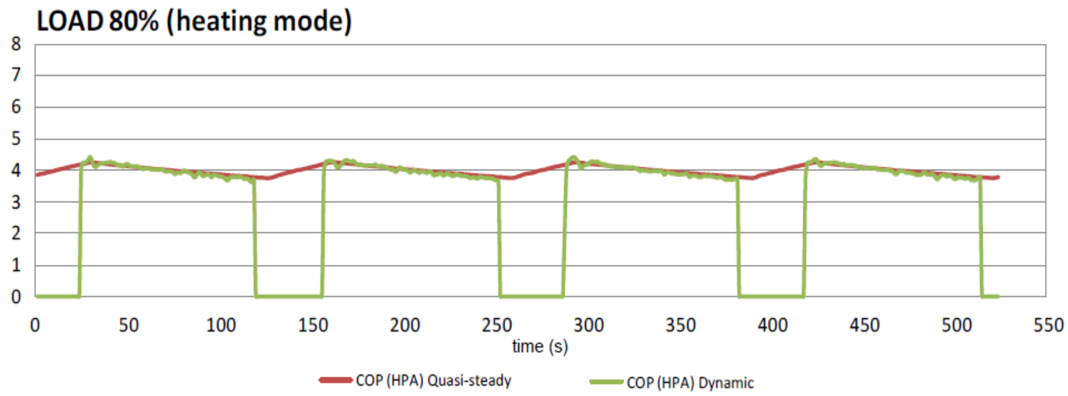


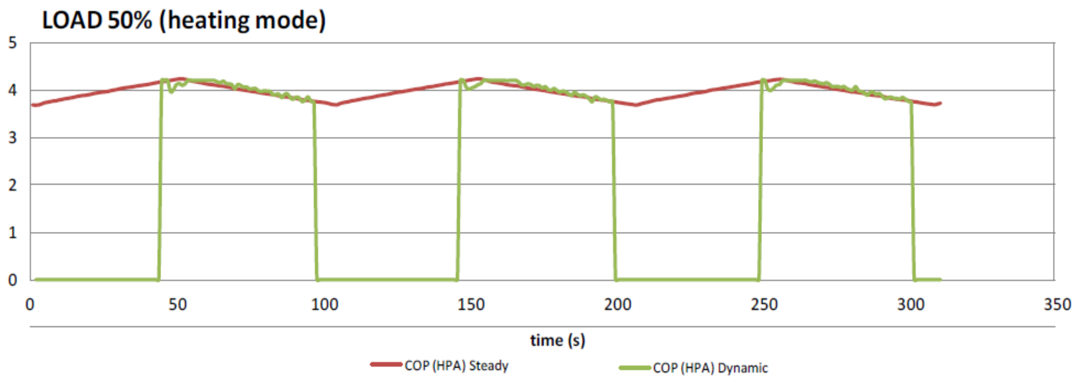
Time [s]
2000
1800
1600
1400
1200
1000
800
600
400
200
0
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1
α
toff
ton
tcycle
Min(t_{on})=2min

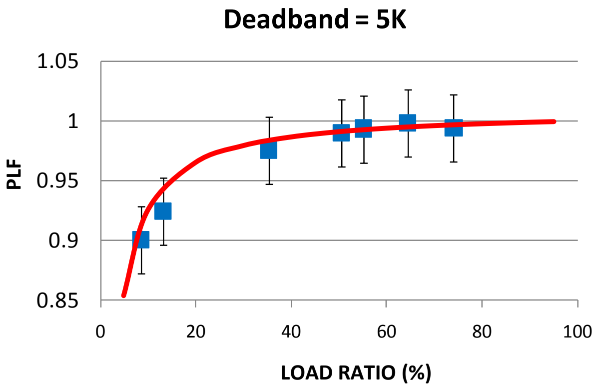

HEATING CAPACITY [kW]
y = -0.1081x + 30.978
28
26
24
22
20
18
16
14
12
32 33 34 35 36 37 38 39 40 41 42
Condenser inlet temperature [ºC]


COP HPA
y = -0.0934x + 7.699
6
5
4
3
2
1
0
32 33 34 35 36 37 38 39 40 41 42
Condenser inlet temperature [ºC]


LOAD 20% (heating mode)
time (s)
COP (HPA) Steady
COP (HPA) Dynamic


LOAD 80% (heating mode)
time (s)
COP (HPA) Quasi-steady
COP (HPA) Dynamic


LOAD 50% (heating mode)
time (s)
COP (HPA) Steady
COP (HPA) Dynamic


Deadband = 5K
1.05
1
0.95
0.9
0.85
PLF
0
20
40
60
80
100
LOAD RATIO (%)

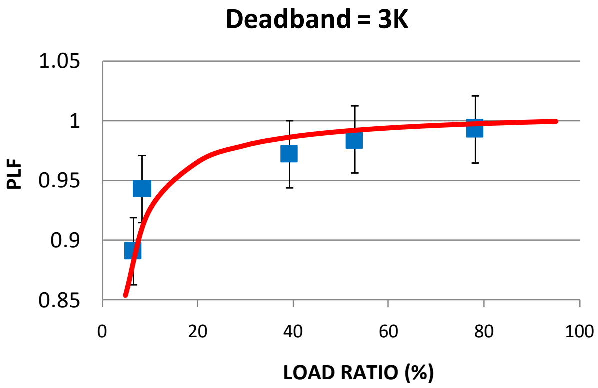

Deadband = 3K
1.05
1
0.95
0.9
0.85
PLF
0
20
40
60
80
100
LOAD RATIO (%)

Figure 1. Layout of the test rig

Figure 2. Inlet and outlet water temperatures at the evaporator and condenser at different thermal loads

Figure 3. Zoom of Figure 2: inlet and outlet water temperatures at the condenser for a 50% load ratio

Figure 4. Example of condensation and evaporation pressure evolution

Figure 5. ON, OFF and cycle periods as a function of the load ratio α

Figure 6. HP performance at 18ºC at the inlet of the evaporator and different water temperatures at the inlet of the condenser

Figure 7. Instantaneous COP and quasi-steady COP at different thermal loads

Figure 8. Measured values of the Part Load Factor (PLF) for two different values of the thermostat deadband. The continuous line is the plot of expression 12 with null start-up losses in both cases.

Equipment	Type	Range	Precision
Thermocouples	T	-270-400ºC	±0.5K
Thermoresistance	PT-100	-220-850ºC	±0.1K
Mass Flow meters (Coriolis)	Danfoss	0-5500 kg/h	±0.05%
Pressure transducers	Fisher Rosemount	1000 kPa	±0.2 %
	Fisher Rosemount	1000-2500 kPa	±0.2 %
	Fisher Rosemount	0-5500 kPa	±0.2 %
Power meter	SINEAX CAM	30kW	±0.2%

Table 1. Specifications of the instrumentation used in the laboratory test rig.

Compressor	Inner Loop HX	Outer Loop HX	Expansion device	Refrigerant
Scroll Danfoss SH90 15,3 $m^3 \cdot h^{-1}$ @ 2900 rpm, 50 Hz	BPHE Swep V80HPx26 Evaporator in cooling mode	BPHE Swep V80HPx26 Condenser in cooling mode	Thermostatic Expansion Valve TRE10	R410A

Table 2. Prototype components.

$\alpha = \frac{\dot{q}}{\dot{Q}}$	t_{on}(s) experimental	t_{on}(s) theoretical	t_{off}(s) experimental	t_{off}(s) theoretical	t_{cycle}(s) experimental	t_{cycle}(s) theoretical
0.16	140	144	870	754	1010	898
0.52	285	251	241	232	526	483
0.72	454	431	150	167	604	598

Table 3. Comparison between experimental and theoretical characteristic periods at different load ratios.