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Non-Intrusive Performance Assessment Method for Heat Pumps: Experimental Validation and Robustness Evaluation Facing Faults

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ABSTRACT

Thanks to their high theoretical efficiency, residential heat pumps (HP) are a promising technology when attempting to reduce the energy consumption of heating in dwellings. Evaluation of their real performances on-field is thus crucial to guarantee their efficiency. However, measuring accurately real heating capacity of air-to-air HP is not easy, since measuring air enthalpy and mass flow rate is challenging. A previously developed non-intrusive internal method based on the compressor energy balance has been improved. It can calculate the coefficient of performance (COP) of different HP types, including air-to-air, on field, thanks to real-time measurements, without interfering with normal operation of the system, and without technical data of the specific heat pump. In this study, a complete validation of this method has been led on a test bench, using an air-to-water HP in order to compare the results of the method with the water-side measurements. This internal refrigerant method was tested for various climatic conditions and heating needs, in stationary and dynamic conditions, including defrosting phases. Different faults were simulated to analyse the behaviour of the method in these conditions, including refrigerant charge faults and heat exchanger fouling. The analysis could be extended to identifying which parameters need to be observed to early detect these faults. The method proves to be robust and its uncertainty to remain low, although it varies with the different working phases. The precise knowledge of real-time performances obtained with this method can help to assess the performance impact of faults and thus to improve associated fault detection and diagnostic methods. On a longer-term scale, the comparison of measured field performances and performances obtained via simplified models, such as regulatory models for instance, could give interesting indications to improve these models.

1. INTRODUCTION

The first step of improving energy efficiency of buildings and their energy equipment is to know precisely the performance data, taking into account the real functioning conditions. To achieve the reduction of energy consumption and carbon emissions due to heating, domestic hot water production and air conditioning in dwellings, the technology of heat pumps is a key solution since their theoretical efficiency is very high. In order to help the development of residential HP, it is necessary to gain transparency on their real on-site performances. The rated COPs that are provided through regulatory test benches are estimated only in standard conditions and thus are not depicting a good representation of reality, since real performance does not only depend on the inherent efficiency of the system, but also on the climate, the sizing, the quality of installation, the control, or the maintenance of the system for example. This large number of influent parameters makes it difficult to have a very precise way to predict the performances, which is why an on-field measurement is needed. Accurate knowledge of the refrigerant mass flow rate in real time could also lead to improve fault detection and diagnosis methods, making maintenance operations easier and avoiding the efficiency loss of HP systems due to several common faults.

A measurement method has been developed (Tran, 2012) in order to calculate the performance of the system continuously on-field without interfering with the correct functioning of the heat pump. It was then validated for HP systems with an internal heat exchanger by Goossens et al. (2016b), but its uncertainty remained high particularly because the evaluation of the heat losses of the compressor was not accurate enough. Niznik (2017) reduced its uncertainty thanks to a better evaluation of these heat losses. The on-field performance measurement method is based on a virtual mass flow rate sensor using the energy balance of the compressor, without using any manufacturer data or compressor map. It is thus applicable to any heat pump technology, including air-to-air. In this study, the experimental validation of this method for a simple cycle was extended to a large number of functioning conditions, in stationary state and also including changes in operating conditions, starting and defrosting phases. The method was then tested in faulty conditions, such as exchanger fouling and refrigerant undercharging and overcharging, to evaluate its robustness. The test bench was an air to water prototype in a climatic chamber. The heating capacity measured with the internal refrigerant method was compared with the reference values obtained through the water-side measurements for validation.

2. NON-INTRUSIVE PERFORMANCE ASSESSMENT METHOD

2.1 Method principle

Based on the refrigerant thermodynamic cycle, the in-situ refrigerant method described in Tran et al. (2013) is applicable to any type of residential heat pumps, including air to air. It uses only refrigerant-side measurements, avoiding the difficulty of measuring the air flow rate on field.

The coefficient of performance of the heat pump is defined as:

$$COP_{HP} = \frac{\dot{Q}_{cond,m}}{\dot{W}_{HP}} \tag{1}$$

With $\dot{W}_{HP}$ the measured total electric power consumption of the system and $\dot{Q}_{cond,m}$ the method heating capacity of the heat pump, calculated as follow:

$$\dot{Q}_{cond,m} = \dot{m}\Delta h_{cond,in\rightarrow out} \tag{2}$$

$$\dot{W}_{HP} = \dot{W}_{comp} + \dot{W}_{aux} \tag{3}$$

$\dot{W}_{aux}$ represents the electrical power due to auxiliaries (fans, control, power electronics), $\dot{m}$ the mass flow rate of the working fluid and $\Delta h_{cond,in\rightarrow out}$ the variation of enthalpy between the inlet and the outlet of the condenser.

The working fluid is majorly composed of refrigerant fluid, but there is also a small part of oil, necessary to ensure the good working of the compressor, that migrates into the refrigerant cycle. The oil fraction, C_{oil}, is considered to be equal to 0.5 % of the total working fluid (Goossens, 2017). Thus, in steady state, the variation of enthalpy can be expressed as:

$$\Delta h_{cond,in\rightarrow out} = (1 - C_{oil})\left(h_{r,cond,in} - h_{r,cond,out}\right) + C_{oil}.c_{p,oil}\left(T_{cond,in} - T_{cond,out}\right) \tag{4}$$

With $c_{p,oil}$ the specific heat capacity of the oil, r stands for refrigerant.

The mass flow rate is indirectly measured through an energy balance at the compressor:

$$\dot{W}_{comp} = \dot{m}\left[(1 - C_{oil})\left(h_{r,comp,out} - h_{r,comp,in}\right) + C_{oil}.c_{p,oil}\left(T_{comp,out} - T_{comp,in}\right)\right] + \dot{Q}_{losses} \tag{5}$$

Where $\dot{W}_{comp}$ the compressor power input measured, and $\dot{Q}_{losses}$ the compressor heat losses. Therefore, the refrigerant mass flow rate can be expressed as:

$$\dot{m} = \frac{\dot{W}_{comp} - \dot{Q}_{losses}}{(1 - C_{oil})\left(h_{r,comp,out} - h_{r,comp,in}\right) + C_{oil}.c_{p,oil}\left(T_{cond,in} - T_{cond,out}\right)} \tag{6}$$

To have the enthalpy values needed, pipe surface temperature measurements are monitored, and if there is no pressure sensor, low pressure and high pressure can be determined from the temperature measurements where the fluid state is undoubtedly diphasic: the evaporator inlet for low pressure and the center of the condenser surface for high pressure. The pressure measurement error that could be caused by the temperature glide of zeotropic fluids classically used in residential HP (R410A, R407C) is very limited (Tran, 2012).

2.2 Heat losses measurement

Goossens (2016b and 2017) developed a method to estimate the heat losses of the compressor, based on CFD modelling of different types of compressors and experimental study. This work established the best locations for temperature sensors for the compressor shell and the ambient air in order to estimate the compressor heat losses.
For rotary compressors, only one surface temperature sensor is necessary on the compressor shell, and only two for scroll compressors (one is used to check the validity of the correlation domain). The surface temperature is referred as T_{surf}. Goossens (2017) concluded that the best hypothesis for the ambient air temperature T_{amb} as seen from the compressor is to set $T_{amb} = T_{ext}$, with T_{ext} the air temperature outside the HP unit.
For example, the correlation used to calculate the compressor heat losses for a rotary type compressor is:

$$\dot{Q}_{losses} = \left(\frac{\overline{Nu}_L k}{L} A_L + \frac{\overline{Nu}_{D,1} k}{D} A_D + \frac{\overline{Nu}_{D,2} k}{D} A_D\right)\left(T_{surf} - T_{amb}\right) + \sigma A_{tot}\left(T_{surf}^4 - T_{amb}^4\right) \quad (7)$$

Where Nu is the Nusselt number for the different sides of the compressor, k is the thermal conductivity of the air, L, D and A stand for the different characteristic dimensions of the compressor, respectively length (height), diameter (1 and 2 for the top and the bottom surface respectively) and area for the lateral side, the top and the bottom of the compressor, σ is the Stefan-Boltzmann constant for radiative heat transfer.

2.2 On field instrumentation

In summary, the method requires the instrumentation described on figure 1.

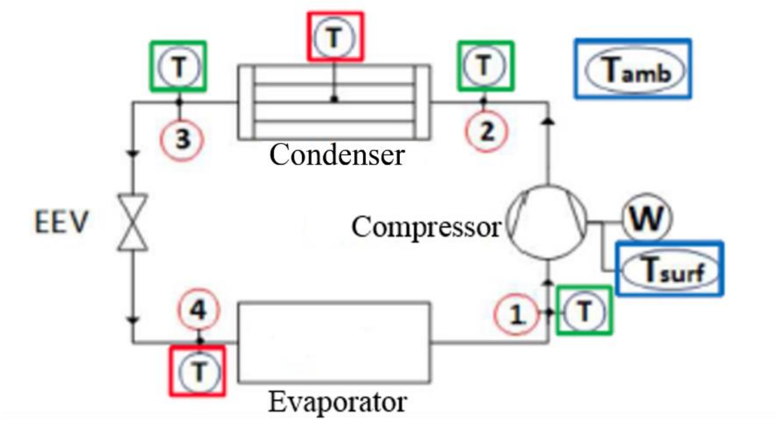

Figure 1: Required measurements for the in-situ performance assessment method

The temperature measurement in green are for measuring enthalpy, in red for low and high pressure, and in blue for the compressor heat losses.

3. STEADY STATE EXPERIMENTAL VALIDATION

The performance assessment method described above has been widely validated in the climatic laboratory of EDF Lab Les Renardières.

3.1 Experimental setup

The test bench was an air-to-water prototype with a rotary-type compressor, installed in adjacent climatic chambers. Six different ambient temperatures were tested between 15 and -10 °C in the climatic chamber where the outdoor unit was installed. Four condensing temperatures between 30 and 60 °C, and three compressor frequencies (30, 60 and 90 Hz) were set. Each combination of these values was tested, thus representing 72 test points. The laboratory was automated to ensure the changes of conditions without the need of human intervention in order to save time. For each point, the acquisition lasted for one hour and thirty minutes in order to be sure that the steady state was reached and to have a sufficient amount of data to analyze. The experimental set-up is described in figure 2.

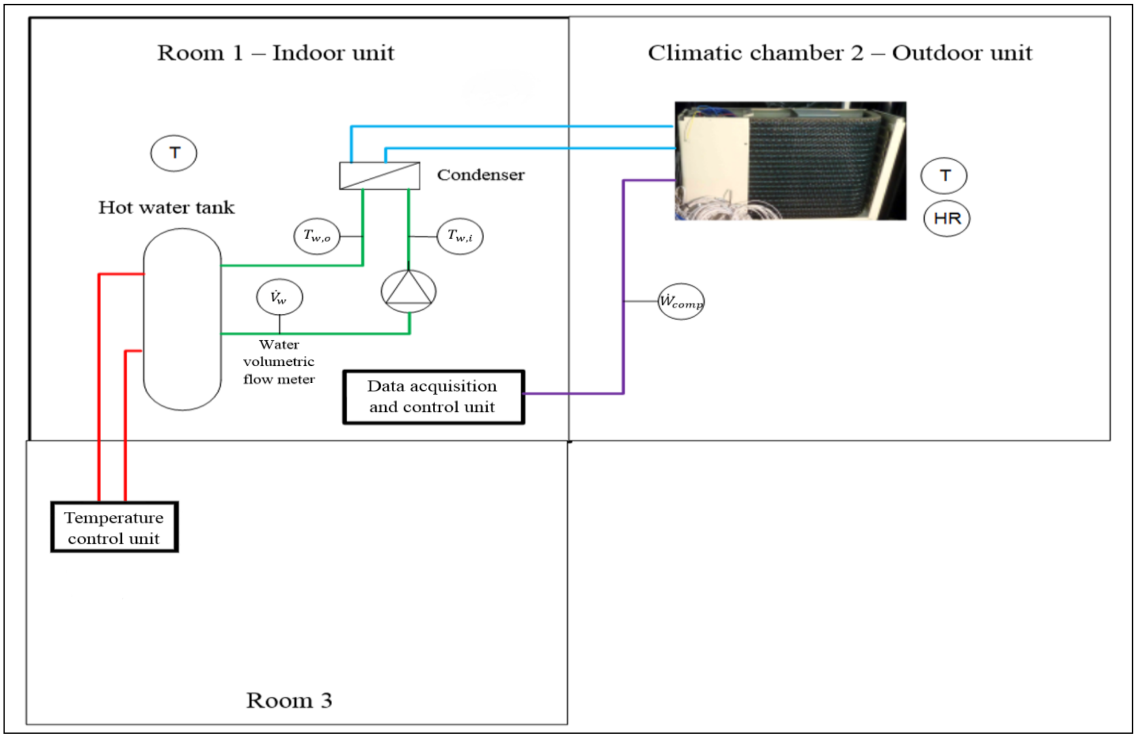

Figure 2: Air-to-water HP prototype installed in climatic chambers

Besides the instrumentation dedicated to the performance assessment method, an additional instrumentation was installed. In order to measure the heating capacity from the water side $\dot{Q}_{cond,w}$, the inlet and outlet water temperature (respectively $T_{w,in}$ and $T_{w,out}$) were measured, as well as the volumetric flow rate $\dot{V}_w$. The heating capacity is then calculated as follow:

$$\dot{Q}_{cond,w} = \rho_w.\dot{V}_w.c_{p,w}.(T_{w,out} - T_{w,in}) \qquad (8)$$

With ρ_w the water density and $c_{p,w}$ the water specific heat capacity. $\dot{Q}_{cond,w}$ is taken as the reference for the validation of the performance assessment method.

The relative uncertainty of the volumetric flow rate measurement was 0.5%, and the uncertainty of the inlet and outlet temperature sensors were 0.2°C. The absolute uncertainty of $\dot{Q}_{cond,w}$ was calculated with the uncertainty propagation formula:

$$\sigma_{\dot{Q}cond,w} = \sqrt{\sum_i \left(\frac{\partial \dot{Q}cond,w}{\partial x_i}\right)^2 \sigma_{x_i}^2} \qquad (9)$$

where x_i is the measured variable and σ_{x_i} is the absolute uncertainty of each measured quantity. The absolute uncertainty of the water-side heating capacity was almost independent of the operating condition and around 123 W.

The electrical power consumed by the compressor and by the entire HP system are measured with two wattmeters. High and low refrigerant pressures were measured with pressure sensors.

3.2 Results

Among the 72 operating conditions that were set, 58 stationary phases could be studied. During each phase, the stabilization lasted one hour, and the 30 minutes left of data were actually analyzed in this part. It has been impossible to reach the stationary state in all the 72 operating conditions because some were interrupted by defrosting at -5° C and -10° C, and the most extreme conditions (very high compression ratio with low frequency) were impossible to stabilize.

For each stationary phase, the mean heating capacity calculated with the refrigerant method and with the water-side measurements were calculated. The relative error and the normalized root mean squared deviation were then calculated as follow:

$$\varepsilon_r = \frac{|\dot{Q}_{cond,m,avg} - \dot{Q}_{cond,w,avg}|}{\dot{Q}_{cond,w,avg}}, \qquad N-RMSD = \frac{1}{\dot{Q}_{cond,w,avg}}\sqrt{\frac{\sum(\dot{Q}_{cond,m} - \dot{Q}_{cond,w})^2}{n}} \qquad (10), (11)$$

Where $\dot{Q}_{cond,m,avg}$ is the mean heating capacity calculated with the method during 30 minutes of a stationary phase, $\dot{Q}_{cond,w,avg}$ is the mean reference heating capacity over the same period, and n the number of measurements made.

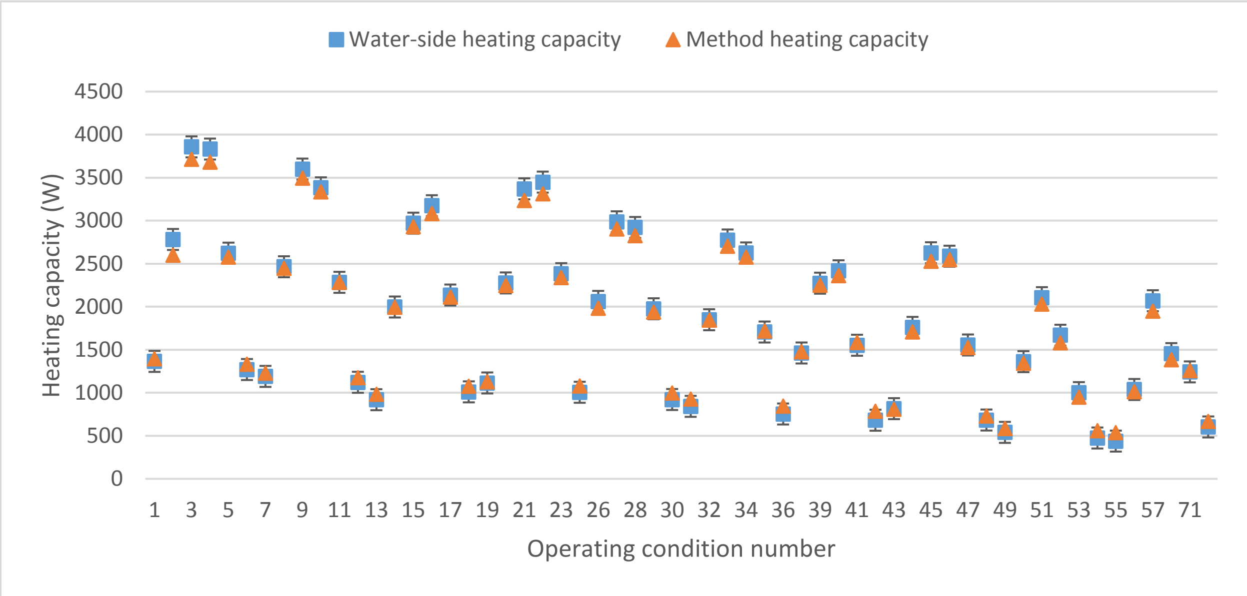

Figure 3: Mean heating capacity for each operating condition, calculated with the method and from the water-side measurements, with measurement uncertainty on the water-side.

For these 58 operating conditions, with different outdoor temperatures, condensing temperatures and compressor frequencies, the refrigerant measurement method gives the same order of magnitude as the water-side measurements for heating capacity (figure 3). The mean relative error is 4.4 %, the minimum is 0.04 %, for T_{ext} = 15 °C, T_{cond} = 60 °C and Freq = 60 Hz (operating condition 11) and the maximum is 22.6 %, for T_{ext} = -5 °C, T_{cond} = 50 °C and Freq = 30 Hz (operating condition 55). In 91 % of the cases, the relative error is less than 10 %, in 71 % of the cases it is less than 5 % and in 43 % of the cases it is less than 2.5 %. The relative error is higher than the water-side measurement relative uncertainty in only five cases:

Operating condition	T_{ext}	T_{cond}	Freq	Relative error	Water-side measurement uncertainties
2	15 °C	30 °C	60 Hz	6.6 %	4.4 %
3	15 °C	30 °C	90 Hz	3.8 %	3.2 %
4	15 °C	40 °C	90 Hz	4 %	3.2 %
21	10 °C	40 °C	90 Hz	4.1 %	3.6 %
22	10 °C	30 °C	90 Hz	4 %	3.6 %

Operating condition 2 was actually not stationary because the condensing temperature was varying due to the limitation of the temperature control unit. For the other cases, the measurement uncertainties are among the lowest because the heating capacity is high, but the relative error remains below 5 %.

The mean N-RMSD is 6.3 %, thus higher than the mean relative error. The N-RMSD represents the deviation between the two methods at each instant, while the relative error is the global error during the 30 minutes, taking into account some compensation effects.

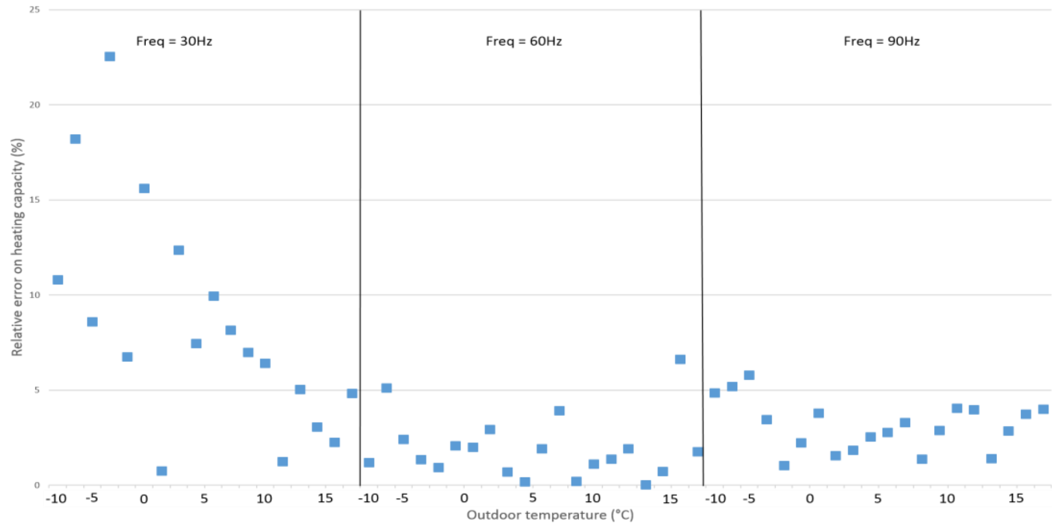

Figure 4: Relative error for each operating condition, sorted by compressor frequency and outdoor temperature.

It can be observed that the relative error is higher when the heating capacity is low, thus when the compressor frequency and the outdoor temperature are low, but the relative uncertainty of the water-side method is also the highest for these cases. In reality, the HP is unlikely to be working at low frequencies for high compression ratios.

4. INTEGRATION OF DYNAMIC PHASES

The compressor energy balance method is rigorously valid only in steady state conditions. However, the method was tested under dynamic phases in order to conclude about the acceptability of the error made. These dynamic phases include evolution in temperature conditions and compressor frequency, starting and defrosting phases.

For the instantaneous heating capacity, there is an instability of the result of the calculation by the method when there is a brutal change in the operating conditions, as can be seen in figure 5. But when the result is integrated over a certain duration, the performance calculated with the method and from the water-side measurements seem to converge, as can be observed on figure 6, which represents the COP integrated during all the time that precedes.

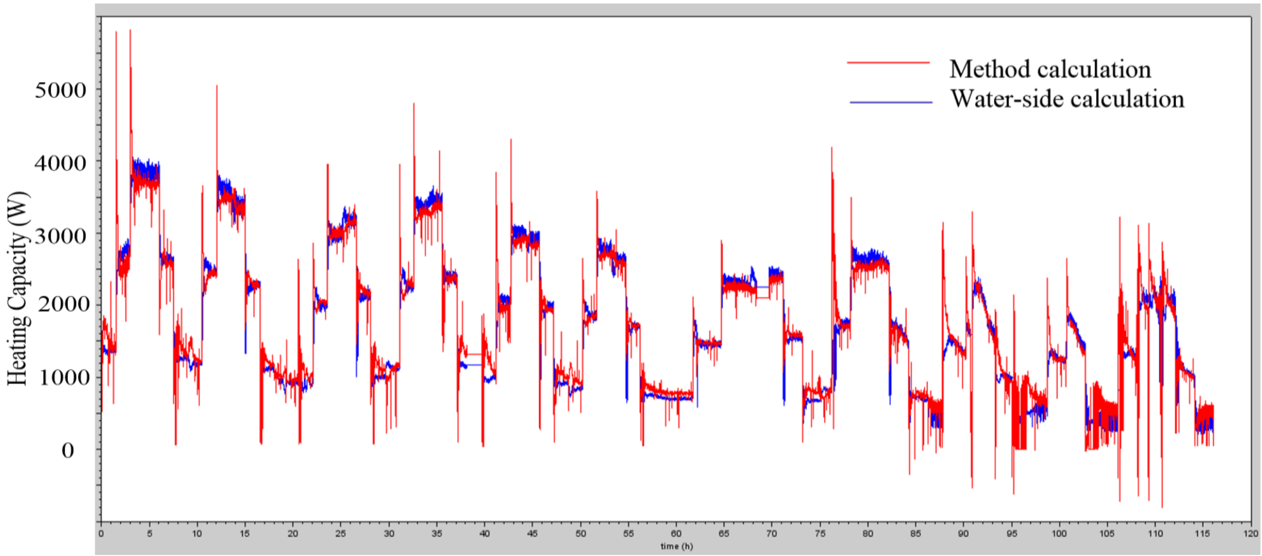

Figure 5: Instantaneous heating capacity (W) over the 72 operating conditions tested.

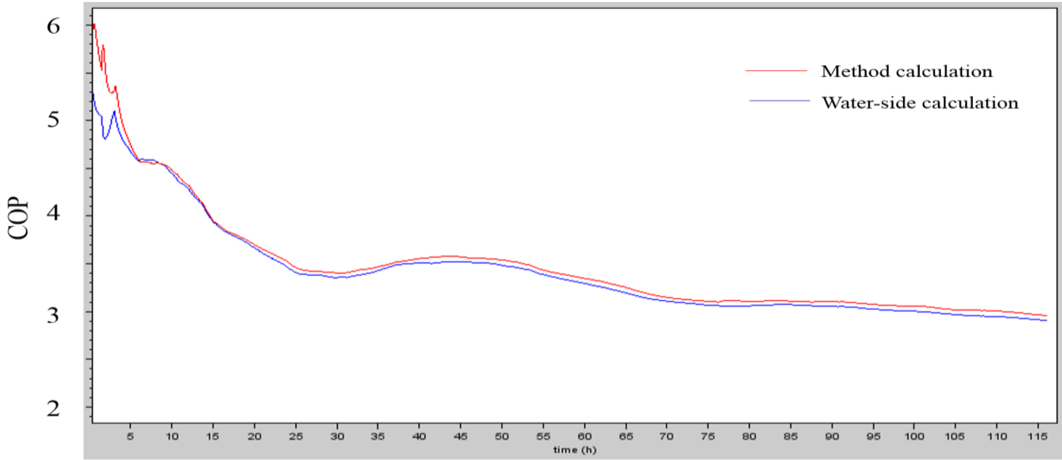

Figure 6: "Sliding" COP integrated at each moment over all the time that precedes, over the 72 conditions tested

The method is less stable at the end of the tests on figure 5 because there are defrosting phases and more extreme conditions the outdoor temperature being below 0° C. It is also sensitive to brutal compressor frequency changes that are not representative of a normal functioning. However, when integrating the heating capacity to calculate the COP over a longer time, the two calculations converge with a relative error of 1.4 % at the end of the 72 conditions. The dynamic phases that are integrated are short enough to be negligible over the entire period of time. The figure 6 also shows that 5 hours of integration are enough to have less than 2 % of relative error (1.5 %).

5. VALIDATION IN FAULTY-CONDITIONS

It has been reported that many faults can occur during the lifetime of the heat pump systems (Madani, 2014). The method was thus tested under faulty conditions to estimate if it still can give an accurate estimation of the performance while being confronted to faults, and to evaluate the range of fault intensity under which the method is still relevant. The goal here is to determine whether the measurement method developed can be used to measure a performance degradation caused by faults, but not to analyze the behavior of the HP in these conditions.

For each fault intensity, 6 operating conditions were tested to have different compression ratios and compressor frequencies.

Operating condition	T_{ext}	T_{cond}	Freq
1	10 °C	40 °C	60 Hz
2	10 °C	40 °C	45 Hz
3	10 °C	40 °C	30 Hz
4	-5 °C	50 °C	90 Hz
5	-5 °C	50 °C	75 Hz
6	-5 °C	50 °C	60 Hz

For certain fault intensities, some operating conditions could not be properly stabilized. For example, at -5 °C for evaporator fouling, the fan could not be stopped during the defrosting phases making them inefficient.

5.1 Evaporator fouling

Since the evaporator is in the outdoor unit, it is very common that dirt accumulates on it and reduces the heat exchange between the air and the refrigerant. To simulate this fault, the rotation speed of the fan was reduced to 90 %, 70 % and 50 % of the nominal speed. It has to be noted that for condition 4 at 70 % and conditions 4 and 5 at 50 %, defrosting phases disturbed the tests. Like in the previous parts, the calculation with the method was compared to the water-side measurements.

For every fault intensity and operating condition, the method proves to give results close to those obtained from the water-side measurements. The relative error remains low and there is no obvious correlation between fault intensities and relative errors. When comparing the mean relative error for each fault intensity, the highest mean relative error is for 50 % of nominal speed (2.8 %) and the lowest for 90 % of nominal speed (1.2 %). For every fault intensity and operating condition, the relative error is lower than the relative water-side measurement uncertainty.

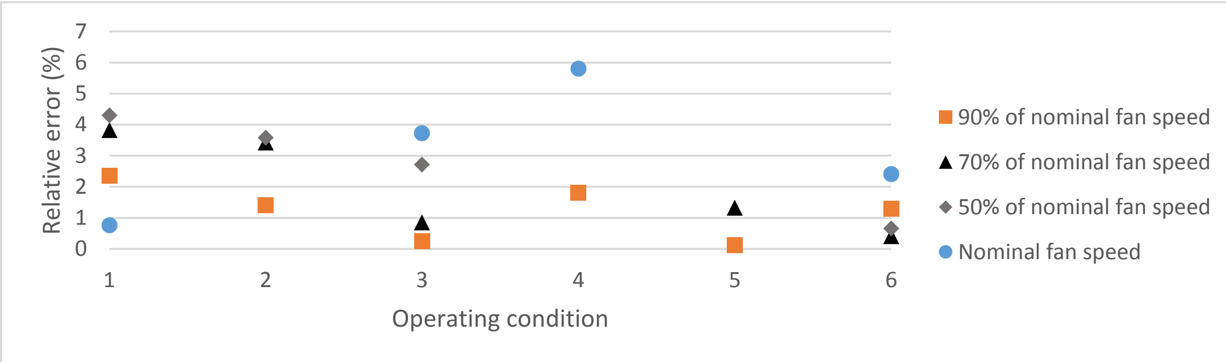

Figure 7: Relative error on heating capacity for different operating conditions and evaporator fault intensities

5.2 Refrigerant charge faults

HP systems can involuntarily work with a refrigerant charge that differs from the nominal charge. It can be explained by an initial mistake at the installation or, in the case of an undercharge, by a leak. An incorrect charge can be the source of a performance degradation. The method was thus tested on the HP on which different refrigerant charge faults were voluntarily set: 120 %, 70 % and 50 % of the nominal charge.

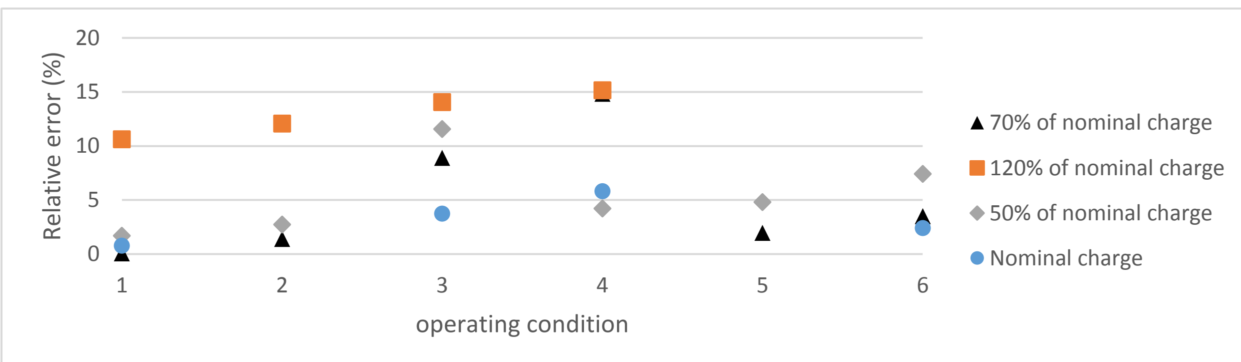

Figure 8: Relative error on heating capacity for different operating conditions and charge fault intensities

It was impossible to set operating conditions 5 and 6 for 120 % of nominal charge because high compression ratios combined with high frequencies and a refrigerant overcharge lead to discharge temperatures and pressures over the security limits.

For the three first operating conditions, the method is clearly more disturbed by overcharging than by undercharging, which is a less common fault. For 70 % of the nominal charge, the relative error remains very acceptable except for operating condition 4 where it is higher than the measurement uncertainty on the water-side, but it is explained by the fact that it includes a defrosting phase.

For 50 % of the nominal charge, the fluid was almost always diphasic at the condenser outlet, thus the hypothesis that the fluid was saturated in these cases was made in order to obtain the enthalpy values. Even with this approximation, the relative error remains low, it is higher than for 70 % of the nominal charge but it follows exactly its trend.

However, for the overcharged condition, the relative error is always higher than the water-side measurement uncertainties. There is no obvious explanation of this, but the best guess is that the compression ratio is higher thus the difference between suction and discharge temperatures is more significant, and the compressor heat losses are miscalculated.

For all the evaporator and refrigerant charge faults tested, the method tends to over-estimate the heating capacity when it is high (over 1500 W) and it tends to underestimate it when it is low (below 1500 W), like it was also the case in normal conditions, as we can see on figure 5.

6. CONCLUSIONS

To have a good idea of the real on-site performances of heat pumps it is necessary to make precise measurements on the long term on field, without disturbing the normal functioning of the system. The internal refrigerant method, which is a promising solution for this purpose, was tested in a large number of operating conditions, and proves to reach a satisfying error level in steady state conditions, even for very low outdoor temperatures and high heating needs. In almost all the conditions tested, the relative error of the method when comparing to the water-side measurement method is lower than the measurement uncertainty of the latter. It can thus be used to assess the performances of a HP system with a satisfying accuracy, especially for air to air heat pumps where measuring the heating capacity from the air-side is very constraining.

Even though this method does not give satisfying instant results for brutal changes in the operating conditions, the integration of the heating capacity over a certain period of time including dynamic phases gives a precise estimation of the COP when compared to the water-side measurement. It can thus be a good tool to measure the seasonal performance of heat pump systems on field, because it is very little invading for the user and does not disturb the HP functioning. Moreover, the cost of such a solution should be very reasonable since it is mainly based on temperature measurements. It will then be implemented on field to have a better knowledge of real performances of residential heat pumps.

The accuracy of the method was evaluated in evaporator fouling and refrigerant charge faults conditions to assess its robustness. For the conditions tested, the relative error remains satisfying, but it is relatively high for an over-charge fault. A further analysis must be made to have a better understanding of the behavior of the method facing faults. The method can then be used to detect a performance loss caused by faults, but the virtual mass flow sensor developed could also be very useful to early detect faults, associated with a fault detection and diagnosis method.
A more precise analysis of the fault impacts on the performance and functioning parameters of inverter heat pump systems must be conducted over a larger range of faults and fault intensities.

NOMENCLATURE

A	area	(m²)
C_{oil}	oil concentration	
$c_{p,oil}$	oil specific heat	(J/(kg.K))
D	compressor diameter	(m)
h	enthalpy	(J/kg)
k	fluid thermal conductivity	(W/(m.K))
L	lateral compressor length	(m)
$\dot{m}$	mass flow rate	(kg/s)
$\overline{Nu}$	Nusselt number	
$\dot{Q}$	thermal power	(W)
T	temperature	(K)
$\dot{W}$	electric power	(W)
ε_r	relative error	
σ	Stefan-Boltzmann constant	(W/(m².K^4))

Subscript

Amb	ambient air
avg	average
cond	condenser
comp	compressor
D,1	top surface of the compressor
D,2	bottom surface of the compressor
in	inlet
L	lateral side of the compressor
m	calculated with the method
out	outlet
r	refrigerant
surf	surface of the compressor
tot	total
w	calculated from the water-side measurements

Abbreviations

COP	coefficient of performance
CFD	computational fluid dynamics
EEV	electronic expansion valve
Freq	compressor frequency
HP	heat pump

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