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Cong Toan Tran, Derek Noël, Philippe Rivière, Christine Arzano, Dominique Marchio. In-situ method for air-to-air heat pump seasonal performance determination including steady-state and dynamic operations. *International Journal of Refrigeration*, 2021, 127, pp.239-249. 10.1016/j.ijrefrig.2021.03.001 . hal-03508107

HAL Id: hal-03508107

<https://hal.science/hal-03508107>

Submitted on 24 May 2023

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Title

In-situ method for air-to-air heat pump seasonal performance determination including **steady-state** and dynamic operations

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Abstract

There is no reliable in situ measurement method of air-to-air heat pump seasonal heating performances. Several methods were previously developed but they were only validated in steady-state conditions. In real life, dynamic sequences such as defrost cycle and starting phases occur, which are not considered in these methods. The goal of this work is to complete one of the existing methods so that it can address both steady-state and dynamic phases. The compressor energy balance method is used here: in the standard heating phase, refrigerant fluid measurements determine the refrigerant flow rate by using the compressor thermal balance. Modelling hypothesis are formulated regarding performance evaluation during defrost and start-up sequences. An algorithm is developed in order to identify the operating mode (standard heating phase, stop, start-up, defrost). Default parameters for this algorithm are proposed. The developed method is assessed over a long term test by comparison with a reference method based on intrusive measurement on the refrigerant. The integrated heat pump performance measured with the in-situ method is within 1 % of the reference measurement. Although large errors may occur in dynamic phases, their occurrence is low and some errors may compensate between the different phases. Leads on further work needed to bring this method closer to deployment are discussed. Better knowledge of the occurrence of the different operating modes is required. Further adaptation is also required for heat pumps using liquid injection at compressor suction.

Keywords

Air-to-air heat pump; seasonal performance; in situ measurement; non-intrusive method; dynamic operation

Nomenclature

Symbols

| | |
|---------------------|---|
| A | Area in m^2 |
| C_g | Oil mass fraction in % |
| D | Compressor shell diameter in m |
| f | Compressor motor frequency in Hz |
| h | Enthalpy in $kJ.kg^{-1}$ |
| k | Air thermal conductivity in $W.m^{-1}.K^{-1}$ |
| L | Compressor shell height in m |
| \dot{m} | Mass flow in $kg.s^{-1}$ |
| p | Pressure in bar |
| P | Electric power input in W |
| \dot{Q}_{cond} | Heating power at the condenser in W |
| \dot{Q}_{amb} | Compressor heat losses to the ambient environment in W |
| T | Temperature in K |
| t | Time in s |
| U_{com} | Compressor internal energy in J |
| W | Wattmeter in (figure 1) |
| x | Refrigerant vapour mass fraction |
| Δ | Difference |
| ΔT_{dis} | Compressor discharge superheat in K |
| ΔT_0 | Compressor discharge superheat lower value, below which compressor inlet is probably diphasic, in K |
| ΔT_{SC} | Subcooling in K |
| ΔT_{SH} | Suction superheat in K |
| η | Isentropic efficiency |
| σ | Stefan-Boltzmann constant in $W.m^{-2}.K^{-4}$ |
| $\sigma^{\Delta T}$ | Uncertainty of contact temperature measurement in K |

Subscripts and acronyms

| | |
|-------------|---|
| <i>amb</i> | ambient |
| <i>cd</i> | condensation |
| <i>com</i> | compressor |
| <i>cond</i> | condenser |
| <i>COP</i> | Coefficient of performance |
| <i>dis</i> | discharge |
| <i>ev</i> | evaporation |
| <i>HP</i> | Heat pump |
| <i>in</i> | inlet |
| <i>isen</i> | isentropic |
| <i>liq</i> | liquid saturated phase |
| <i>nom</i> | nominal value |
| <i>oil</i> | oil circulating in the refrigeration circuit |
| <i>out</i> | outlet |
| <i>r</i> | refrigerant |
| <i>ref</i> | stands for reference : results based on intrusive measurement |
| <i>SCOP</i> | Seasonal coefficient of performance |
| <i>tot</i> | total (total area : sum of lateral and bottom and top compressor shell areas) |
| <i>vap</i> | vapor saturated phase |

1 Introduction

Heat pumps (HPs) experience a strong development today in Europe and in particular air-to-air units, as HPs start replacing fossil fuel boilers (EurObserver, 2018). The success of electric HPs is due to their potential to lower energy consumption and greenhouse gas (GHG) emissions, which requires both highly efficient HP systems and low electricity generation mix CO₂ content. HPs are also part of heating means contributing to the achievement of renewable heating targets of EU member states (EU, 2018) and higher efficiency also means higher renewable energy contribution. HP efficiency is measured according to its coefficient of performance (COP), which is the ratio of useful heat delivered to electrical energy consumed. In Europe nowadays, HP performances are characterized by a seasonal coefficient of performance (SCOP) according EN14825 standard (CEN, 2018a). In real life, HP efficiency will significantly differ from the standard SCOP values with local climate, specific building insulation, real sizing, possible secondary or backup heater and controls, indoor temperature set points, quality of installation, degradation of performance over time including potential faults in installation or machine components (Nordman et al., 2012). It thus seems highly relevant to measure HP seasonal performances in situ to evaluate their real contribution in cutting energy consumption and GHG emissions, and their renewable energy contribution.

(Tran et al., 2011) made a review of all possible methods for both internal (measurement on refrigerant side) and external (measurement on air side) heating capacity measurement, in order to identify possible in situ methods and also laboratory methods to check in situ methods. The main issue for air-to-air HPs is the continuous measurement of the heating capacity. Measurement on the air side with hot wire velocity sensors for the air flow and temperature sensors at inlet and outlet has been done in several research studies (Ichikawa et al., 2007) (Rivière et al., 2010b) (Tran et al., 2013) (Tran et al., 2019). However, the deployment of these methods in situ is hampered by unknown accuracy, high sensor costs and likely sensor accuracy degradation in time with dust and maintenance. Regarding internal methods, the heating capacity is computed as the product of the refrigerant mass flow and the enthalpy difference between condenser inlet and outlet. Direct refrigerant mass flow measurement using non-intrusive techniques is not feasible today: flow meters as ultra sound techniques are not adapted to small pipe size and refrigerant plus oil mixtures (Tran, 2012). Other techniques are however feasible. For instance, any manufacturer could use the performance map of the compressor or expansion valve to deduce the refrigerant mass flow from the measurement of low and high pressures (Woohyun and Braun, 2012). This method was for instance applied for commercial size variable refrigerant flow HPs for various sites across Europe (Vandaele et al., 2016). However, our objective is to measure the in situ performance independently from the manufacturer and thus we cannot use this type of method.

This was the reason to develop an alternative internal method based on the energy balance of the compressor (Tran et al., 2013). The mass flow rate is computed through an assessment of the compressor heat losses, the measured electric compressor power and the enthalpy difference between the refrigerant suction and exhaust of the compressor following Eq. 1. In the original method, η , the ratio between compressor heat losses and power consumption was fixed and depended only on the type of compressor (Falhén, 2004). In the developed method, heat losses are modelled as a function of compressor shell temperature and outdoor temperature, using natural convection correlations. Impact of oil circulation is also accounted for explicitly. With the exception of the high pressure measurement which can be intrusive, as a high pressure plug is in general

present for refrigerant charging purpose on these machines, all other measurements are non-intrusive.

$$\dot{m} = \frac{(1 - \eta)P_{com}}{h_{com,out} - h_{com,in}} \quad (Eq. 1)$$

This method has been validated extensively in steady-state conditions for standard cycles with rolling piston compressors (Tran et al., 2013; Niznik, 2017; Zhang et al., 2019) and showed a very good accuracy. Niznik (2017) showed that the root-mean-square deviation of the in-situ method was 2.36 % from the reference heating capacity values for a few steady-state testing points. Noël et al. (2018a) experimentally showed that the average difference between the compressor energy balance method and the direct water-side measurement on an air-to-water HP is below 5 % in various steady-state heating phases. The measurement uncertainty was estimated to be about 5 % taking into account the sensors accuracy (Noël et al., 2018b). But the main sources of error in the COP estimation in standard heating phases are actually the uncertainty about the oil mass fraction circulating through the cycle and the accuracy of the compressor heat losses estimate. Niznik (2017) estimates that the latter are responsible for 27 % of the total uncertainty of the heating capacity calculation. Zhang et al. (2019) adapted the compressor energy conservation method for variable refrigerant flow (VRF) systems and compared it with an air-side measurement. They obtained a difference of 13 % in heating capacity between the two methods and concluded that the compressor energy conservation method can be accurate for calculating the capacity of VRF systems on field.

This method in principle could also work for scroll compressors. However, experiments by Niznik (2017) have shown that the shell temperature of scroll compressors is less homogeneous than the one of rotary compressors. (Niznik, 2017) also gives sensor positioning and weighting factors allowing to represent the average compressor shell surface temperature and to compute heat losses.

The method was also adapted to injection cycles (Singer et al., 2016) and proved accurate experimentally in steady-state conditions (Goossens et al., 2016a). Its relative error compared to the reference method was below 8 % for tests led on a partial injection HP cycle with internal heat exchanger and scroll compressor.

Its main drawback is that it becomes inaccurate in case of diphasic refrigerant entering the compressor and/or leaving the condenser, both conditions that occur in dynamic conditions, when the HP starts, stops or is in defrost operation mode. In some cases, it may also occur in steady-state conditions (Tran et al., 2012), diphasic fluid enters the compressor and measured performance may be flawed.

The main objective of this article is to develop a complete methodology to assess the in-situ seasonal performances. To that purpose, and because of the intrinsic limits of the method and also because the method should be independent from the HP manufacturers, we had to develop an algorithm which allows, first to identify the operational mode of the HP based on measured operating variables, and second to build better estimates of the HP performance during phases for which the method estimate is flawed by specific operating conditions. This article is organized as follows. Part 2 describes the methodology, including the test bench, the calculation method for each operational mode and the algorithm to identify the operational mode. In part 3, results regarding the

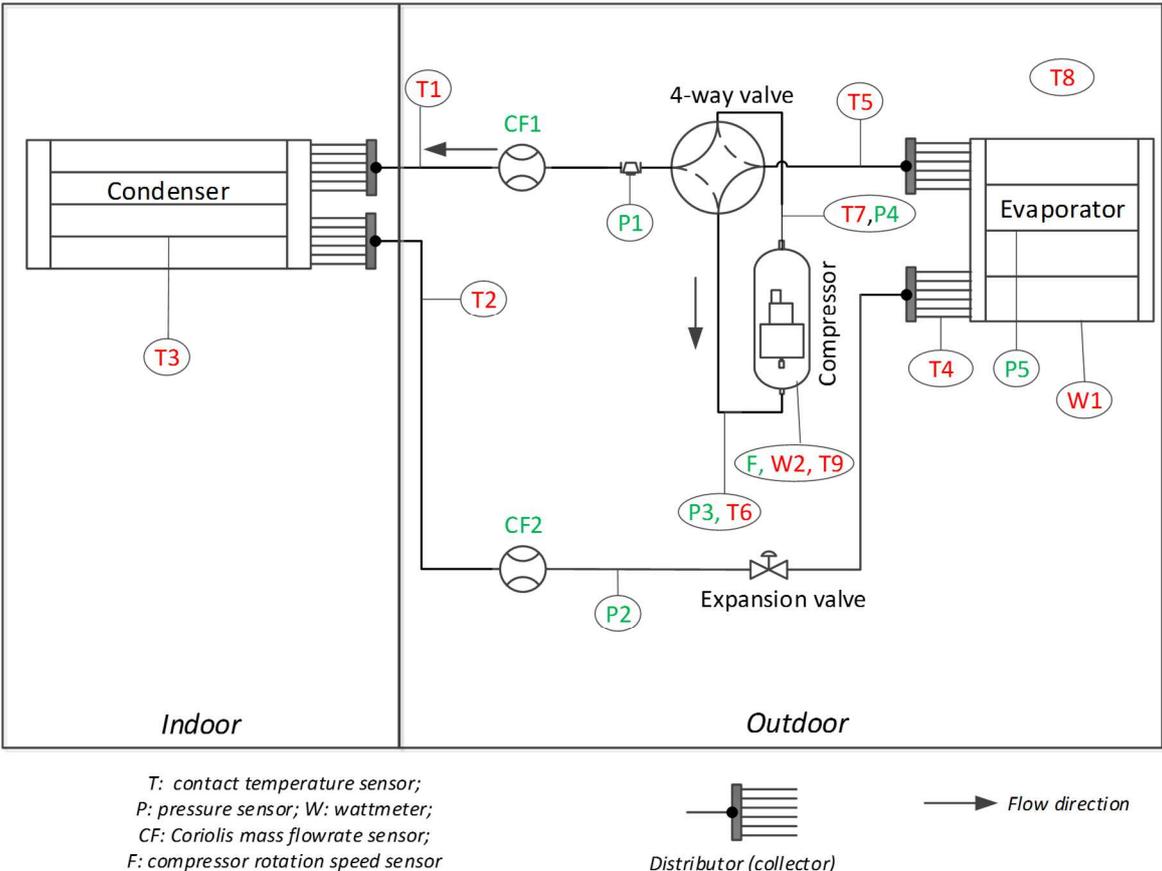
experimental verification of the main hypotheses of the calculation method and the global validation of the SCOP are presented. In part 4, results are quantitatively and qualitatively discussed in order to evaluate how far they can be generalized and what future work is required.

2 Methodology

2.1 Metrology

The in-situ method was developed (Tran et al., 2013) and improved (Niznik, 2017) in order to measure accurately the performances of HPs with the lightest metrology as possible, without any impact on the normal operation of the machine.

To measure the in-situ HP performances, the compressor energy balance method requires the enthalpy values at the inlet and outlet of the compressor and condenser, the compressor power input and its heat losses. It uses only one surface temperature on the compressor shell. Electrical power measurement of the whole unit is also necessary. To avoid any fluid-intrusive measurement, the mass flow rate is deduced from the steady-state energy balance of the compressor in standard heating phases. Figure 1 represents the needed instrumentation.



In red → In-situ method ; In green → Intrusive measurement

Figure 1 : Required instrumentation to measure the HP performances for the in-situ method (in red) and additional intrusive measurement (in green) for the experimental verification (part 3)

Evaporating and condensing pressures are computed from temperature measurements. The evaporating temperature is measured at the evaporator inlet (T4). The condensing temperature measurement (T3) is made with a surface Pt100 sensor placed on a bend of the tube of the exchanger, where a sensor is generally already set by the manufacturer to ensure the heat pump control. High pressure measurement can sometimes be made directly from the high pressure plug. Temperature is also measured at the inlet and outlet of the condenser and compressor. Enthalpy values are then deduced from the obtained temperature and pressure data.

For the mass flow rate calculation, compressor energy input and outputs need to be measured. The measured electrical power input (W2), is converted into an enthalpy difference between the compressor suction and discharge, and the rest of it into heat losses. In order to estimate these losses, the ambient temperature is measured (T8). Niznik (2017) numerically and experimentally showed that for a rotary-type compressor, the compressor shell temperature is relatively uniform regardless of the operating condition. Therefore, only one sensor is used to measure this temperature (T9). These sensors locations, for a rotary-type compressor, are defined by Niznik (2017) at mid-height of the compressor shell, and at the outdoor unit air inlet for ambient temperature. The author tested 9 different sensor ambient temperature locations and showed that considering that the compressor ambient environment is always at the outdoor air inlet temperature gave the best heating capacity results and was the easiest place to set up a sensor. The appropriate heat transfer correlations are then used to get the compressor heat loss values (Niznik, 2017).

To get the HP performances, the total electrical power consumption of the heat pump (W1) is also measured.

2.2 Calculation method

The main calculation method is used for standard heating phases, under which the machine runs most of the time. Starting and stopping phases, as well as cycle inversions during defrosting cycles, are taken into consideration separately to get the seasonal coefficient of performance.

2.2.1 Standard heating phase

A few minutes after starting, the HP operation stabilizes. Considering that the steady-state is reached, the performances are calculated with the refrigerant-side energy balance method.

From a literature review and considering the difficulty of measuring the mass fraction of oil circulating through the cycle, Tran (2012) considered that it is constant and represents about 2 % of the total circulating fluid. However, (Niznik, 2017) experimentally showed that taking a constant oil concentration value of 0.5 % gave better heating capacity results. The oil enthalpy values are calculated from the temperature measurements and the oil thermal capacity (Liley and Gambill, 1973). The miscibility of the oil in the refrigerant is neglected and their properties are considered separately.

The pressure losses in the refrigerant lines, as well as in the heat exchangers, are neglected.

The fluid is supposed to be superheated vapor at the compressor inlet, so the enthalpy is calculated from the temperature and pressure values. However, the compressor inlet temperature is sometimes very close or even lower than the evaporating temperature. That means that the fluid is

diphasic at compressor suction and in a first approximation, enthalpy is calculated considering the fluid as saturated vapor.

The steady-state energy balance of the compressor gives the refrigerant mass flow rate (Eq. 2):

$$\dot{m} = \frac{P_{com} - \dot{Q}_{amb}}{(1 - C_g)(h_{r,com,out} - h_{r,com,in}) + C_g(h_{oil,com,out} - h_{oil,com,in})} \quad (Eq. 2)$$

Where P is the electrical power, C_g the mass fraction of lubricating oil, h the specific enthalpy and \dot{Q}_{amb} the compressor heat losses. The subscripts com, r, in, out and oil respectively stand for compressor, refrigerant, inlet, outlet and oil.

Compressor heat losses are composed of convection and radiative heat losses (Niznik, 2017). For a rotary-type compressor, they are calculated from Eq. 3:

$$\dot{Q}_{amb} = \left(\frac{Nu_L k}{L} A_L + \frac{Nu_{D,1} k}{D} A_D + \frac{Nu_{D,2} k}{D} A_D \right) (T_{shell} - T_{amb}) + \sigma A_{tot} (T_{shell}^4 - T_{amb}^4) \quad (Eq. 3)$$

where Nu_L is the Nusselt number for the lateral side, $Nu_{D,1}$ and $Nu_{D,2}$ are respectively the Nusselt numbers for top and bottom surfaces, k is the air thermal conductivity at film temperature, L is the height of the compressor shell, D its diameter, A_L is the lateral side area, A_D is the top or bottom plate area and A_{tot} is the total shell area. T_{shell} and T_{amb} are respectively the compressor shell and the ambient air temperatures in Kelvin. σ is the Stefan-Boltzmann constant and the compressor surface emissivity is considered to be equal to unity.

Then, the heating capacity at the condenser is calculated using the enthalpy values at the condenser inlet and outlet (Eq. 4):

$$\dot{Q} = \dot{m} [(1 - C_g)(h_{r,cond,in} - h_{r,cond,out}) + C_g(h_{oil,cond,in} - h_{oil,cond,out})] \quad (Eq. 4)$$

In some cases, it is possible to have no subcooling at the condenser outlet. This is not common but is more likely to happen if the HP is undercharged (because of faulty charging or refrigerant leak). In that case, the fluid is considered as saturated liquid. Finally, the coefficient of performance can be obtained by dividing the heating capacity by the total heat pump electrical power P_{HP} , measured with the wattmeter W_1 (Eq. 5):

$$COP = \frac{\dot{Q}}{P_{HP}} \quad (Eq. 5)$$

The calculation process in standard heating phase is summarized in figure 2.

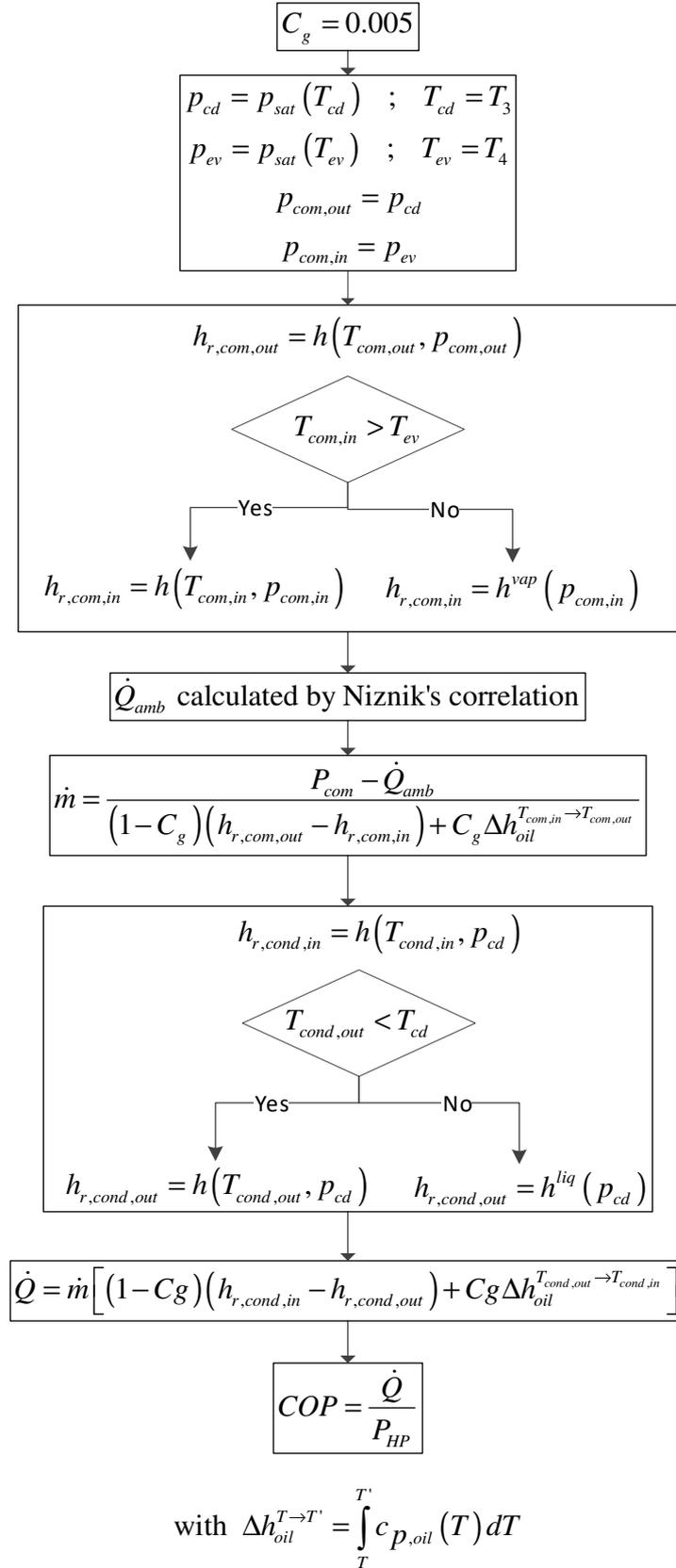


Figure 2 : Calculation of the performances under standard heating phase

This calculation method for standard heating phases is adapted only in stabilized conditions as the energy balance equation of the compressor and condenser assume that there is no variation of the internal energy of the respective components. After a brutal change in compressor frequency like

after a defrosting cycle, at startup of the machine, or with high variations of operating conditions, the HP operates under dynamic conditions during a few minutes. However, it has been shown that the error made neglecting the variation of the internal energy of the compressor and condenser (the main term being the compressor) is very low if integrated over a long enough time period (Tran et al., 2013) and thus not significant for the seasonal performance factor estimate. The validity of this hypothesis will be confirmed in the next part. Therefore, the calculation method described in figure 2 can be used for the heating phase, even when not exactly under steady-state operations.

2.2.2 Starting phase

When the compressor is stopped, high and low pressures converge to reach the same value and the fluid becomes diphasic in all the cycle. When the compressor starts, the compressor first vaporizes the diphasic fluid and the compressor discharge temperature is very close to saturation. During a starting phase, the compressor suction and discharge superheat are very close to zero. In that case, the calculation method for standard heating phases is not appropriate. Indeed, as the refrigerant vapor quality at compressor suction is not measured and may be much lower than 1, it is impossible to know precisely the enthalpy variation at the compressor.

The mass flow rate calculation error using the standard heating phase calculation method during a starting phase depends on the operating conditions: high and low pressures, compressor discharge superheat (difference between the compressor outlet temperature and the condensing temperature) ΔT_{dis} , vapor quality at compressor suction, and compressor isentropic efficiency η_{isen} , defined in Eq. 6.

$$\eta_{isen} = \frac{h_{com,out}^{isen} - h_{com,in}}{h_{com,out}^{real} - h_{com,in}} \quad (\text{Eq. 6})$$

This compressor isentropic efficiency is theoretical as it is defined for an adiabatic compression. It does not take into account the compressor heat losses in order to make it easier to compare it to the compressor isentropic efficiency values given by the manufacturer or in the literature, which are obtained in a controlled environment (typically at 35 °C ambient temperature) with reduced heat losses.

Figure 3 shows the error in mass flow as a function of refrigerant vapor quality at compressor suction and discharge superheat, for R410A with an evaporating temperature of 0 °C, a condensing temperature of 40 °C and isentropic efficiency values η_{isen} between 0.45 and 0.75. This range represents a good estimation of the isentropic efficiency of residential HP compressors (Cuevas et al., 2009) (Yang et al., 2009).

At low compressor discharge superheat, we can be sure that the compressor is in diphasic state because if not, it would mean that the isentropic efficiency would be unrealistically high. So at start-up, while compressor discharge superheat is lower than 25 K, it is very likely that the refrigerant is diphasic at compressor suction. When the superheat is higher than 25 K, and up to 40 K, the compressor suction may be in a two-phase state or not, depending on the real isentropic efficiency of the compressor. Choosing 25 K as a threshold is a conservative choice, making sure that for lower compressor discharge values, the refrigerant at compressor suction is certainly in a two-phase state. Hence, it can be used to detect starting phases.

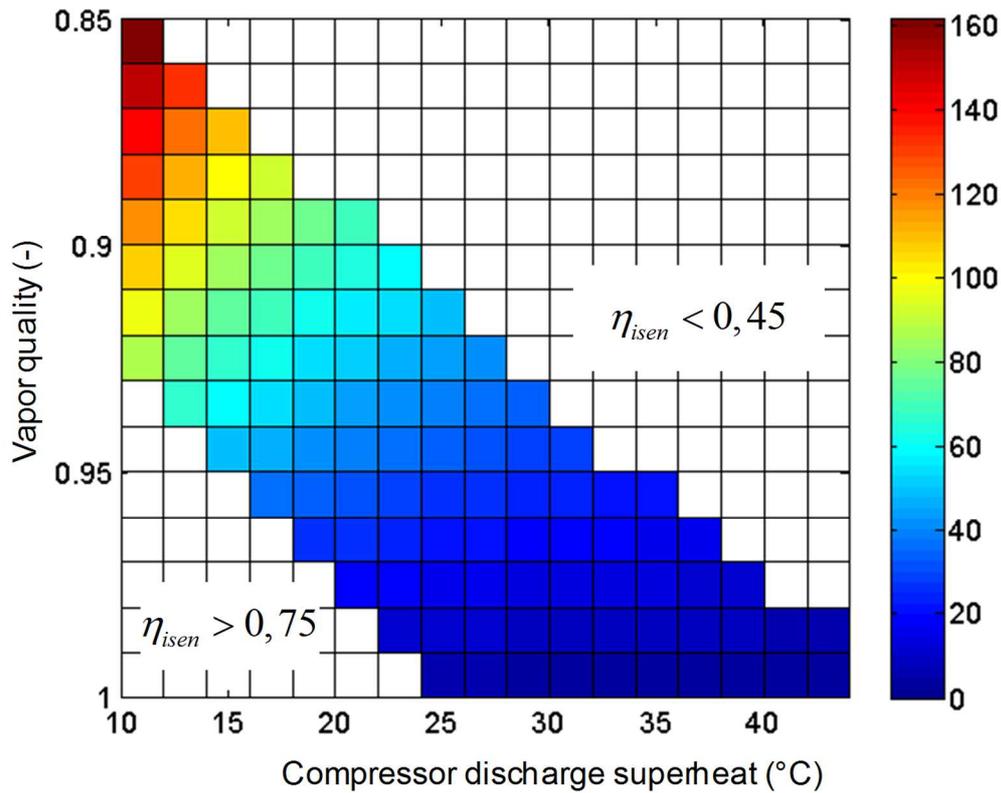


Figure 3 : Calculation error (in %) on the mass flow rate using the standard heating calculation method as a function of compressor discharge superheat and vapor quality.

The two-phase state at the compressor suction leads to highly overestimating the mass flow rate because the vapor quality may be far from the saturation value. Since it is not possible to measure the heating power at the condenser if the fluid is highly diphasic at the compressor inlet, it is supposed to be equal to the compressor electrical input. This assumption will be justified in section 3.1.3.

To summarize, if the compressor discharge superheat is below a certain value ΔT_0 , it is considered that the system is in the starting phase and the coefficient of performance is supposed to be equal to 1. This generally lasts for a few minutes. In the case of R410A and for the discussed working conditions, ΔT_0 is equal to 25 °C. For other cases, the same method can be applied to identify an appropriate value of ΔT_0 .

2.2.3 Defrosting cycle

When the outdoor coil temperature gets close to 0 °C and is lower than the air dew point temperature, frost can appear on the evaporator fins. This reduces the heat exchange and leads to a drop in evaporating temperature and pressure. A defrosting cycle starts: the compressor slows down then stops, it restarts with a reversed cycle to get hot fluid into the outdoor exchanger and get rid of the frost. The compressor stops again and a new starting phase begins before reaching stabilized heating operation.

During the cycle inversion, the indoor fan is stopped to avoid getting cold air into the room. This is thought to be the standard behavior for mini-split air-to-air heat pumps (Rivière et al., 2010), which in general do not have an auxiliary heater. During the defrost cycle, the compressor power is duly

accounted for in the energy consumption, but the heating capacity is supposed to be zero. This phase can be detected as the saturation temperature of the outdoor unit gets higher than the one of the indoor unit and the heating power is supposed to be equal to zero.

2.3 Algorithm

In order to automatically take into account every type of operating phase through the heating season, an algorithm is developed to calculate the seasonal coefficient of performance (SCOP). It is composed of the following steps:

- If the measured electrical power of the compressor is less than 1% of its nominal power, $P_{com} < 0.01P_{com}^{nom}$, it is not working and the heating capacity is zero, $\dot{Q} = 0$. The threshold value of 1 % is chosen because, even when the compressor has stopped, the electrical power is not null due to the stand by consumption of the HP.
- If the compressor is working, the comparison of evaporating and condensing temperatures tells if the cycle is inverted. If $T_{cd} > T_{ev}$, the HP is in heating mode. If not, the cycle is inverted and the heating power is zero, $\dot{Q} = 0$.
- During heating mode, the starting phase can be identified looking at compressor discharge superheat ΔT_{dis} . If $\Delta T_{dis} \geq \Delta T_0$ is not satisfied, it is a starting phase and $\dot{Q} = P_{com}$. The value of ΔT_0 depends on the fluid and the working conditions.
- If the starting phase is over, the heating power and COP can be computed with the calculation method presented in Figure 2.
- Singular measurement points can be detected when the calculated COP is excessively high. If $COP > 3COP^{nom}$, the heating power is supposed to be equal to the compressor electrical power $\dot{Q} = P_{comp}$.
- Among the correct measurements, the diphasic fluid phenomena can be detected. If the suction superheat is lower than the uncertainty of temperature measurement, $\Delta T_{SH} < \sigma^{\Delta T}$, the fluid is in two-phase state at the compressor suction. Similarly, if the subcooling is small enough, $\Delta T_{SC} < \sigma^{\Delta T}$, the fluid is in two-phase state at the condenser outlet. In other cases, the HP operates under standard heating phase.
- SCOP can then be calculated from the integration of the heating power delivered and the electrical power consumption of the HP throughout the season.

Figure 4 illustrates this algorithm.

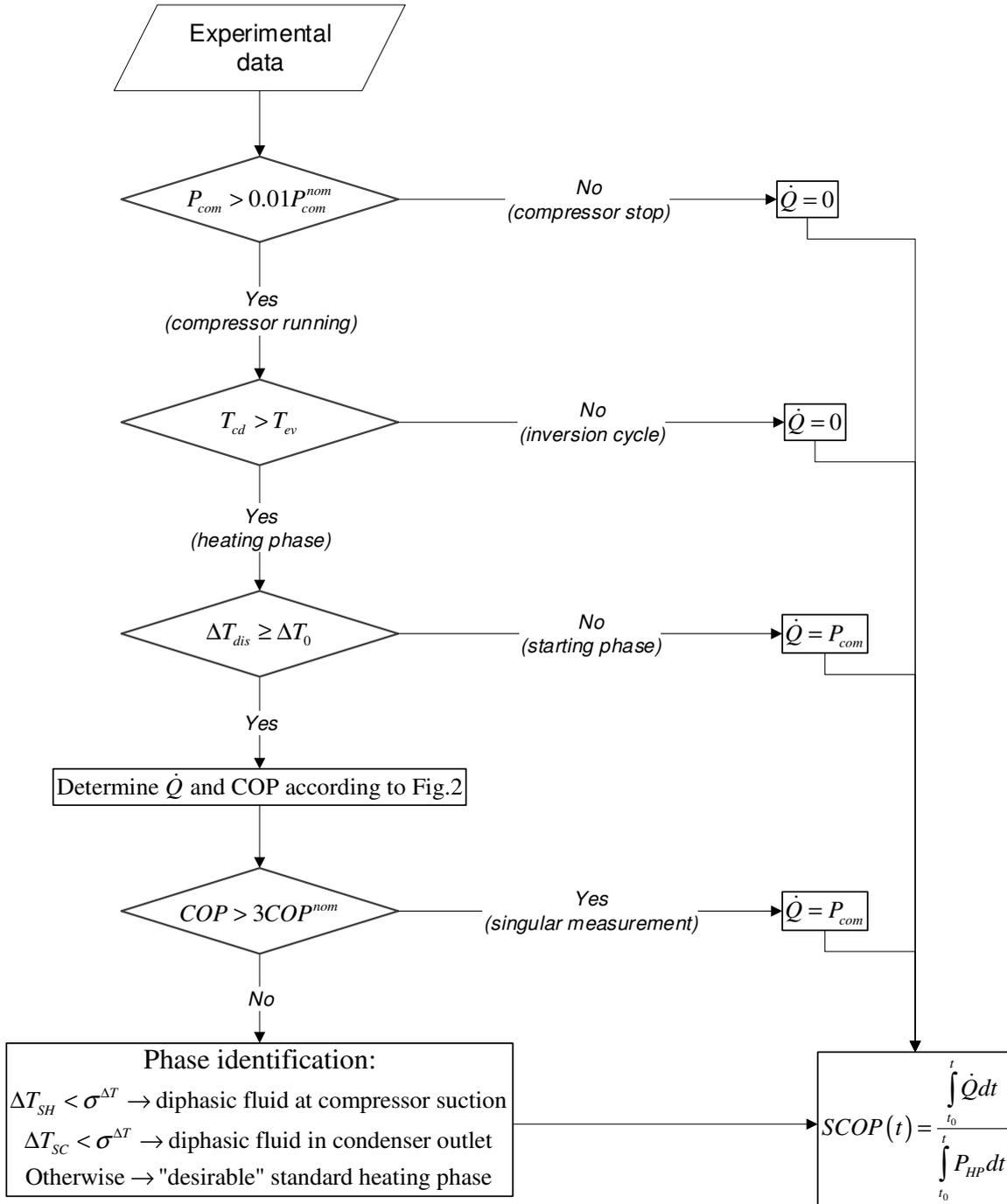


Figure 4 : Seasonal performance measurement algorithm

3 Experimental results

A commercial split air-to-air HP was tested in laboratory conditions, allowing to use intrusive measurements of which the uncertainty is well controlled. Including essentially pressure and mass flowrate measurements, the intrusive sensors are used as reference to compare with the in-situ method. The whole instrumentation of the test bench is presented in Figure 1.

3.1 Validation of the hypothesis

In this section, we will study in detail a defrosting cycle to check the validity of the various assumptions made in the Methodology part.

3.1.1 Detection of the working phases

Figure 5 shows the temperature and compressor frequency measurements. Note that the frequency measurement is not necessary for the in-situ measurement method, but used only to better analyze the working conditions of the HP. Different phases can be identified:

- Decreasing in the rotation speed of the compressor (zone 2)
- Stopping phase (zone 3)
- Inversion cycle (zone 4)
- Stopping phase (zone 5)
- Starting phase (zone 6)

During the inversion cycle, the evaporation temperature (measured in the external unit) is higher than the condensation temperature (measured in the internal unit). Then, this criterion can be used to detect the inversion cycle.

Based on the criterion $\Delta T_{dis} \geq \Delta T_0$, it is possible to identify the moment when the starting phase (zone 6) terminates and the standard heating phase begins (zone 7). Figure 5 show that during the starting phase, the condensation temperature increases gradually and reaches a stable value in the end of this phase. The variation of the compressor frequency has also a similar trend. These observations confirm the validity of the proposed criterion. It should be noted that the evaporation temperature remains roughly constant during the starting phase, so that it is quite hard to use this measurement to separate the zones 6 and 7. Finally, note that the standard heating phase is not necessarily in steady state condition (the frequency is not perfectly constant as shown in Figure 5) because of various reasons like the variation of heat load.

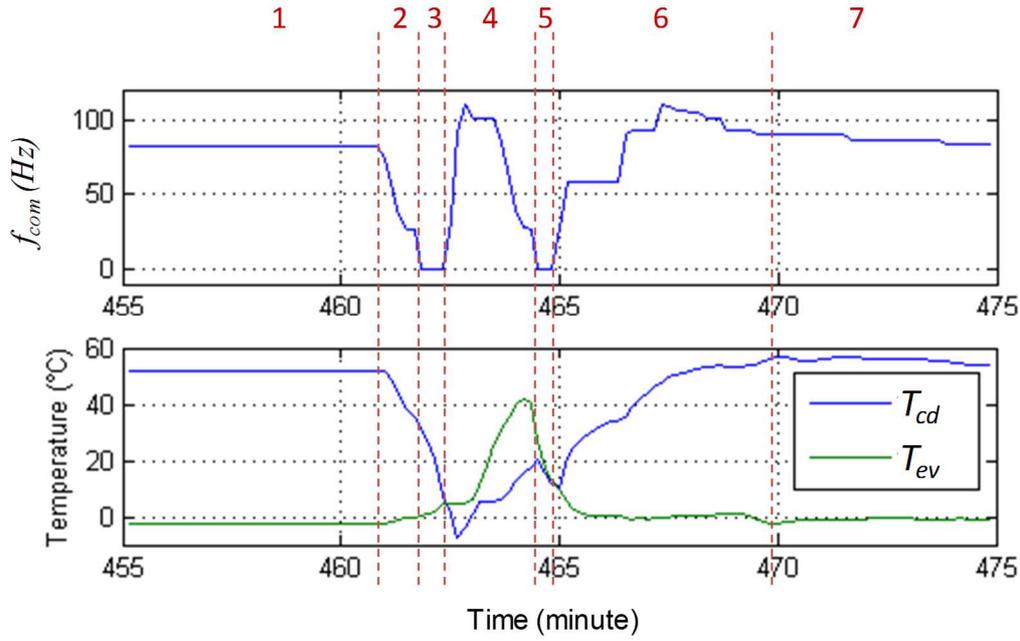


Figure 5: Different phases during a defrosting cycle

3.1.2 Compressor internal energy negligible

The COP calculation method (Figure 2) is adapted only to steady-state conditions as the energy balance equation of the compressor assumes that the HP operates under steady-state conditions. In practice, the HP may work under dynamic conditions where the true energy balance of the compressor is shown in Eq. 7:

$$\int_{t_0}^t P_{com} dt = \int_{t_0}^t \dot{m} \Delta h_{com} dt + \int_{t_0}^t \dot{Q}_{amb} dt + [U_{com}]_{t_0}^t \quad (\text{Eq. 7})$$

where $[U_{com}]_{t_0}^t$ is the variation of the compressor internal energy during a given period. For the results presented in figures 6 to 9, t_0 is set at the 455th minute.

In this part, we aim to quantify the error made when $[U_{com}]_{t_0}^t$ is not considered in the energy balance. To do so, we use Eq. 7 to calculate $[U_{com}]_{t_0}^t$ during the defrosting phase, knowing that the electric consumption, the mass flowrate and the enthalpy variation can be precisely measured via the intrusive sensors and the heat losses calculated via Eq. 3.

The results are presented in Figure 6. In the beginning (zone 1), $[U_{com}]_{t_0}^t$ is close to zero, and that is coherent because the HP is in steady state. When the compressor slows down (zone 2), $[U_{com}]_{t_0}^t$ goes down to -20 kJ which is equivalent to the energy delivered by the HP during 3 seconds under the nominal working condition. In terms of the variation, $[U_{com}]_{t_0}^t$ is changed with a rate of about 330 J/s, which corresponds to 10 % of the electrical power of the compressor under nominal condition. Comparing the calculated heating power with the reference measurement, the instantaneous error may be significant, but the overall error can be negligible because it occurs during a very short period

(about 1 minute) and under a working condition where the heating power is reduced. In this case study, not considering $[U_{com}]_{t_0}^t$ in the energy balance underestimates the energy delivered by 56 kJ during the whole studied phase (zone 2), and that corresponds to the energy produced during 7 seconds under the nominal working condition.

These values represent the upper bounds of $[U_{com}]_{t_0}^t$ for any working conditions because they are measured in a defrosting phase where the compressor state changes abruptly. In other words, when the change in the working condition is more moderate (e.g. in case of a variation due to a frequency change for load follow-up), $[U_{com}]_{t_0}^t$ is expected to be lower, so negligible in the energy balance. In our tests, $[U_{com}]_{t_0}^t$ during a follow-up period is expected to be small (because the outdoor temperature varies slowly) and it cannot be calculated accurately due the uncertainties of the sensors and the compressor heat loss determination (see part 3.3 Uncertainty analysis for more information).

During the inversion cycle (zone 4), it is not possible to correctly calculate $[U_{com}]_{t_0}^t$ because the fluid at compressor suction is diphasic. This is the reason why we present only the results for the zones 1 and 2 in Figure 6.

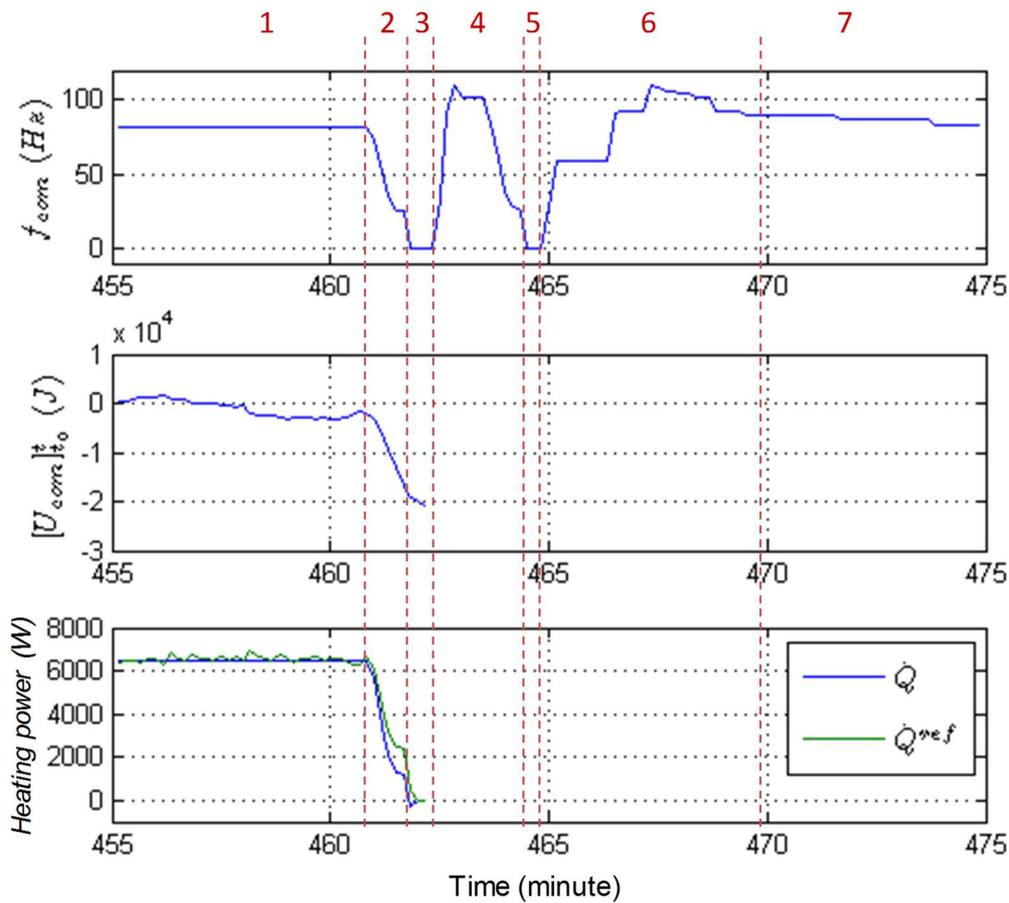


Figure 6: Variation of the compressor internal energy

3.1.3 COP close to 1 during the starting phase

During the starting phase, the in-situ method cannot be used to determine the COP correctly and it is assumed that the COP is equal to 1. We present in Figure 7 the COP obtained by the intrusive measurements. The COP is indeed close to 1 during the starting phase (zone 6) and this confirms that our assumption is reasonable. Although the observed COP value is a bit higher than 1, it is not expected anyway to have a very high COP during the starting phase and the duration of the starting phase is relatively short (about 5 minutes) and so the impact of this limited error is thought to be small.

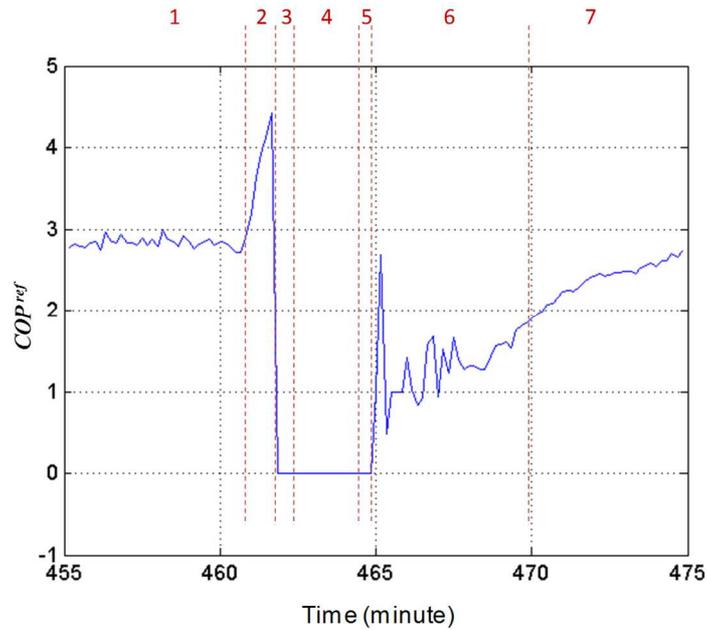


Figure 7: COP determined via intrusive sensors during the defrosting phase

3.2 Validation of the SCOP algorithm

Figure 8 shows the heating power calculated by the proposed algorithm, in comparison with the reference values. Under steady state condition (zone 1), the heating power calculated with the in-situ method is in good agreement with the reference one. It is underestimated during the starting phase (zone 6), and that is because the real COP is a little higher than 1. Then, the heating power is overestimated before to reach the same level of the reference values (zone 7). The overestimation can be explained by the fact that the fluid at compressor suction is not totally in the vapor state, and that results in overestimating the mass flowrate. Indeed, the measurements indicate that the criterion $\Delta T_{SH} \leq \sigma^{\Delta T}$ is satisfied in zone 7. It should be noted that, during our tests we observed the problem of diphasic fluid at compressor suction not only in starting phases, but also, sometimes, under steady state conditions. That is due to the control logics of the tested HP.

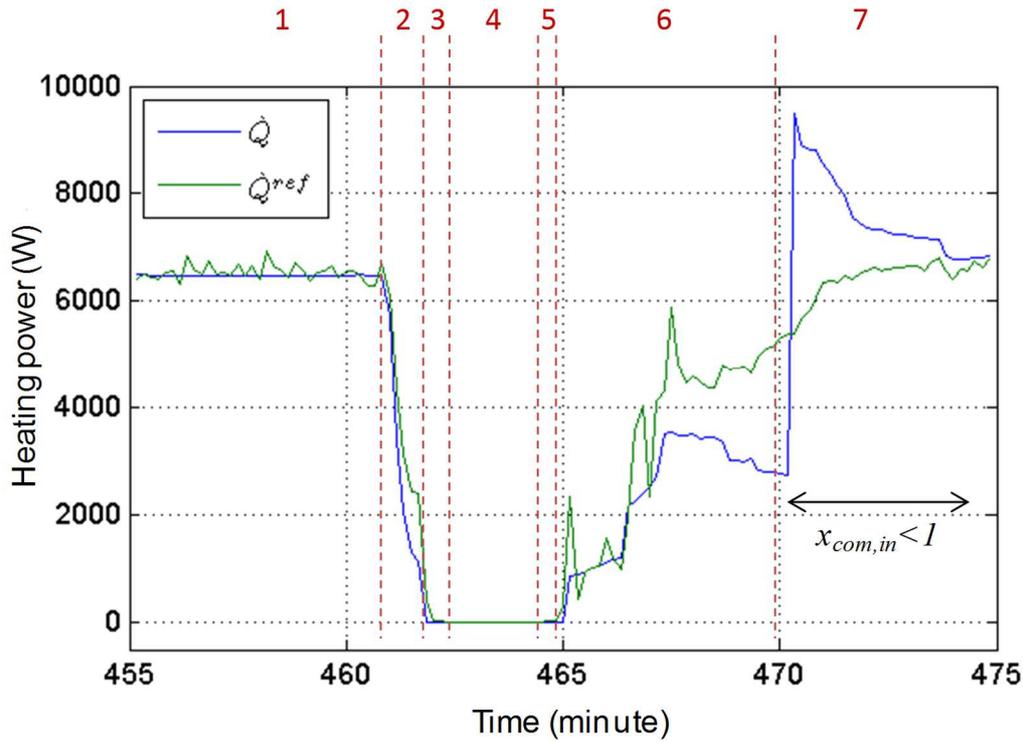


Figure 8: Heating power calculated by the in-situ method, in comparison with the reference values

Figure 9 presents the SCOP (time integrated value) over a period of about 10 hours. Each peak corresponds to a defrosting cycle. At the beginning of the test, the values of SCOP represent the instantaneous coefficient of performance. After the first defrosting cycle, SCOP decreases significantly, and then goes up again when the HP reaches the standard heating phase. The impact of the defrosting cycle on SCOP decreases progressively, and after a while SCOP reaches a stable value corresponding to the overall performance of the HP.

The difference between the two methods is lower than 0.1 SCOP point at all times, which corresponds to a relative error of about 3 %. At the end of the test, the difference is very small (about 0.01 SCOP point). We also observe that the defrosting cycle degrades the performance by about 0.2 point (from 2.9 to 2.7) in these test conditions.

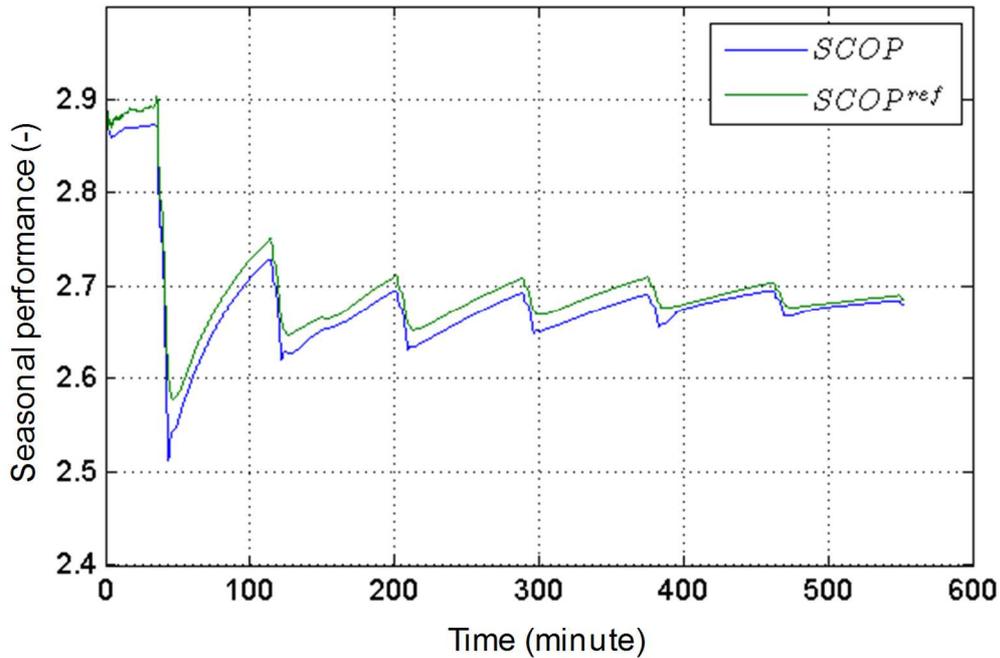


Figure 9: SCOP calculated by the in-situ method, in comparison with the reference values

Table 1 summarizes the heating power difference between the in-situ method and the reference method, according to the various working phases. The compressor stop and inversion cycle occur for 1.4 and 2.4 % of the test duration, respectively. The starting phase occurs during 5.7 % of the time and contributes to 3.4 % of the total energy delivered. For these reasons, even if the error during this phase is high (-29.2 %), the error reported to the whole test remains small (-1 %). The occurrence of singular measurement is negligible (0.06 %). The HP spends most of the time in the heating phase, which includes two cases: a) diphasic fluid at compressor suction (6 % of the time), and 2, without diphasic suction, so-called desirable phase (84.4 % of the time). Considering the whole test, the method overestimates the heating power during the heating phase, but underestimates it during the starting phase. The overall error of the energy produced is 0.2 %, and this explains the very small SCOP error after 10 hours as observed in Figure 9.

| Phase | Occurrence (%) | Energy share (%) | Error (%) | Error reported to the total energy produced (%) | |
|-----------------------------|--------------------------------------|------------------|-----------|---|------|
| Compressor stop | 1.4 | 0 | . | . | |
| Inversion cycle | 2.4 | 0 | . | . | |
| Starting phase | 5.7 | 3.4 | -29.2 | -1.0 | |
| Singular measurement | 0.06 | 0.02 | -97.8 | -0.02 | |
| Heating phase | Diphasic fluid in condenser outlet | 0 | 0 | . | |
| | Diphasic fluid at compressor suction | 6.0 | 6.5 | +5.7 | +0.4 |
| | “Desirable” phase | 84.4 | 90.1 | +0.5 | +0.4 |

Table 1: Errors of the energy delivered according to the working phases (errors compared to the reference method)

3.3 Uncertainty evaluation

The uncertainty of COP is determined with the uncertainty propagation formula. The partial derivatives are calculated using numerical differentiation. The uncertainties of the measured variables are shown in Table 2. The pressure is not directly measured in the in-situ method, but calculated from the saturation temperature measurement. Beside the error due to the uncertainty of sensors, an uncertainty of 0.1 bar is added to the pressure calculation, in order to take into account the pressure losses through the heat exchanger. Finally, the uncertainty of the mass oil concentration is supposed to be equal to $\pm 0.5\%$ (i.e. the concentration ranges between 0 and 1%).

| Variable | Uncertainty (compared to measured values) |
|---|---|
| Refrigerant flow rate (intrusive measurement) | 0.2 % |
| Pressure (intrusive measurement) | 0.2 % |
| Refrigerant temperature | 1 K |
| Electric power | 1 % |
| Data acquisition system | 0.1 % |

Table 2. Instrumental uncertainty

The uncertainty of COP varies with the working conditions. When using the intrusive sensors, it ranges between 2.2 % and 2.9 % in our tests. Concerning the in-situ method, the uncertainty of the heating capacity during our steady state tests is about 5.1 % taking into account the sensor accuracy, which corresponds to the same order of the uncertainty presented in (Derek et al., 2018). If the compressor heat loss determination is responsible for 27 % of the total uncertainty of the heating capacity according to Niznik (2017), the uncertainty of COP is estimated at 7.1 %. However, it seems difficult to quantify exactly the uncertainty of the seasonal performance. The main reason is that the error of the assumptions made for the starting phase and the specific working condition (two phase state at compressor suction) may vary depending on the specific heat pump and installation. If these working phases are short compared to the standard heating phase, it is believed that the overall COP uncertainty is close to the one of the standard heating phase. Further studies are required on different heat pump installations to confirm this assumption.

4 Discussion

The in-situ performance assessment method is accurate in standard heating phases, as the deviation between its performance results and those of the reference method is only 0.5 % for these tests. The seasonal COP evaluation also shows very good results, even though dynamic phases are included. The algorithm presented in this paper can identify the different working phases, and the overall error of the energy produced is only 0.2 %, thanks to a compensation effect of overestimation during standard phases and underestimation during starting phases. This is confirmed by Noël et al. (2018a), where the integrated values of COP of an air-to-water HP using the method and the direct measurement on the water-side converge just after a few hours, including several defrosting cycles and compressor start-ups.

On some real installations, dynamic working phases could be more frequent, especially when the HP is oversized or used under very small part load ratios, as it could reach minimum compressor frequency and would need to stop regularly to control the room temperature. This could lead to greater errors in the seasonal performance estimations as the method is less accurate in dynamic conditions, mostly because after a start-up, the fluid is diphasic at the compressor inlet. Future work could thus be devoted to qualifying the robustness of the method in such operating conditions.

Also, the occurrence of the two-phase suction phenomenon could be greater for some HPs, even in standard heating conditions. In particular, allowing two-phase fluid to enter the compressor lowers the discharge temperature, which can be necessary for systems that use R32 as a working fluid. This control method was patented (Taira et al., 2003), (Yamada, 2017) and is widely used for most recent air-to-air systems in Europe. In that case, the suction vapour quality is unknown and the error made can be significant. Using the saturated vapour enthalpy value instead of the real diphasic value highly overestimates the heating capacity, even if the vapour quality is close to 1. Taira et al. (2003) asserts that it is safe for the compressor to allow a two-phase fluid with a vapour quality of 0.75 to enter its suction port. This two-phase fluid case is another possible area of development for future work on this method.

5 Conclusions and perspectives

HP system performances are one of the key features to reduce the energy consumption and the greenhouse gas emissions of the building sector. However, it is challenging to measure their real-time heating capacity, particularly for air-to-air HPs, without using a heavy, expensive and invasive laboratory instrumentation. The compressor energy balance method has many advantages for in-situ performance assessments as its metrology is adapted to an on-field implementation and it can be used independently from manufacturer information or measurements. It is accurate enough in standard heating phases to avoid the use of a flow-meter or an air-side measurement that would be very invasive for the end-user. When standard heating phases represent a large part of the working time, the SCOP algorithm calculation proves to be very satisfying, attenuating the error made in dynamic conditions.

However, in order to allow its deployment, field studies are required to evaluate the real weight of very small part-load ratio operating conditions over a season for a representative number of HPs. The occurrence of these working conditions could be caused by oversizing and could increase HP on-off cycling. If the dynamic and two-phase suction working phases appear to be more frequent than what was supposed, some improvements have to be made to have a better estimation of the heating capacity in order to decrease the error made in these cases. Moreover, even in standard heating phases, some HP logics can allow two-phase fluid to enter the compressor. In that case, the standard in-situ method needs to be improved or completed. It seems impossible to measure the suction vapour quality on field, so it could be helpful to have a better understanding of the vapour quality control according to the operating conditions. It could also be necessary to consider alternative performance assessment methods to have better accuracy on the heating capacity estimation under this condition. For example, it could be interesting to try using a simplified, universal compressor performance map, either to get the suction vapour quality or directly the mass flow rate, as it would

be impossible to get the performance map of each compressor; in many cases indeed for air-to-air conditioners, the compressor performance map is sensitive proprietary information.

Funding

This work was supported by EdF R&D, TREE department - Technologies & Research for Energy Efficiency.

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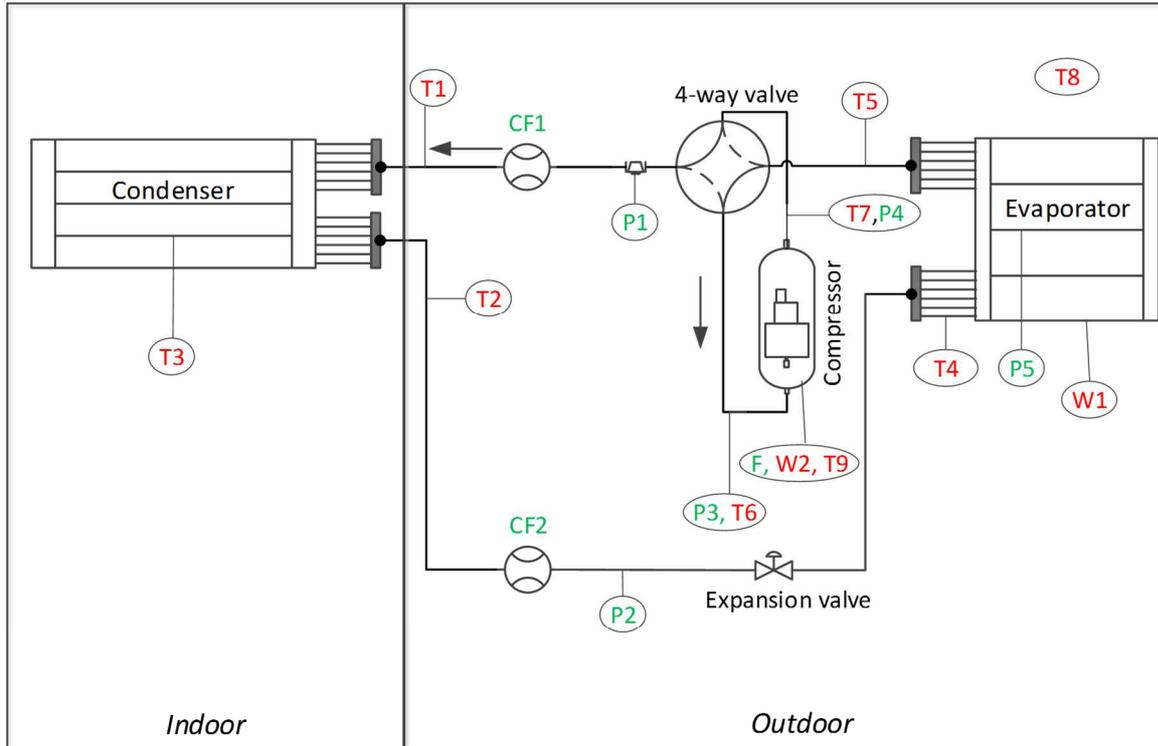
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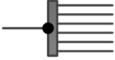
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APPENDIX



T: contact temperature sensor;
P: pressure sensor; W: wattmeter;
CF: Coriolis mass flowrate sensor;
F: compressor rotation speed sensor

 Distributor (collector)

 Flow direction

In red → In-situ method ; In green → Intrusive measurement

Figure 10: Complete instrumentation of the test bench to compare the in-situ method with the reference one