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Performance assessment method for air-to-air heat pumps: on-field implementation and hardware-in-the-loop testing

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Abstract

Due to their high efficiency, heat pumps (HP) are a key technology as part of the necessary decarbonisation of the building sector. Evaluation of their real performances is thus crucial to guarantee their efficiency. Real performances depend on many factors that cannot always be predicted, making in-situ measurement necessary for each installation. However, for air-to-air HP, measuring accurately the real heating capacity is not easy, since measuring the air enthalpy and mass flow rate is challenging. In that context, several thesis projects have led to the development of a HP performance assessment method. It is a non-intrusive internal method based on the compressor energy balance. It can calculate the coefficient of performance (COP) for different types of HP, including air-to-air systems, thanks to real-time measurements, without interfering with normal operation of the system. In this study, a first on-field implementation, on a monosplit air-to-air heat pump, is presented. Complementary results, obtained from laboratory tests, are then presented, allowing the validation of the method under real operating conditions. The first part of the paper presents the principle and the necessary instrumentation of the performance assessment method. The second part details its implementation on a real installation in a meeting room of an office building in the Paris area. The same HP is also tested in a controlled environment using a hardware-in-the-loop set-up in the EDF R&D laboratory, in which real weather conditions are reproduced, and additional reference measurements are performed. Finally, the results, in terms of HP behaviour and performances, are presented and discussed, highlighting the relevance of this in-situ approach for reliable performance assessment.

Keywords: Air-to-air heat pumps, performances, on-field assessment, hardware-in-the-loop test bench

1. Introduction

Heat pumps are one of the keys to reducing greenhouse gas emissions in buildings, thanks to their high performance. However, in the face of this growing market, the installation and maintenance sector is struggling to keep pace, in particular in Europe [1]. There is a lack of skilled workers in the field of residential installations. To prevent the situation described above from undermining the quality of installations and the development of heat pumps for residential application, it is crucial to develop "smart" heat pumps, capable of assessing their performance in real time and identifying any operating faults in advance. In this process, knowledge of the HPs performance is essential. It is the first building block required to build these advanced functions. However, this performance depends on numerous parameters, including outdoor and indoor temperatures, as well as the characteristics and quality of the installation. Consequently, performances determined a priori in the laboratory, with fixed temperatures and particular installation conditions, cannot be used, as they are not representative of the diversity of situations encountered in the field. The implementation of advanced heat pump functions requires real-time knowledge of in-situ performance. However, performance measurement in the field is difficult to implement discreetly, reliably, accurately and inexpensively. The challenge is all the greater if it requires to measure performance of a heat pump that has already been installed and/or a heat pump that uses air as a heat distribution medium. Although laboratory equipment meets the requirements of reliability and precision, it is not at all suitable in terms of discretion and cost.

In this context, a series of thesis were carried out between 2009 and 2020 [2-3-4], culminating in the development of a method for measuring heat pump performance in real time, suitable for all technologies for residential applications.

A first thesis [2] developed an on-board performance measurement method based on compressor energy balance, using non-intrusive refrigerant temperature measurements at different points of the thermodynamic cycle. This method has been validated for pseudo-stationary operation. It also highlighted the fact that the calculation of compressor heat losses is the main source of uncertainty in the method. Based on these results, a second thesis [4] finely modeled the two compressor technologies used in residential heat pumps to derive a simple method for calculating compressor heat losses, based on one or two temperature measurements depending on the compressor type, rotary or scroll. This calculation was incorporated into the performance measurement method, which was then validated on several stationary laboratory test points. Finally, the third thesis [3] aimed at extending the method, to adapt it to any operating condition, including dynamic conditions, and any refrigerant. This improved method has been validated in climatic chambers. Moreover, in her thesis related to a refrigerant charge assessment method [6], Jounay used this method to calculate the refrigerant mass flow rate of an air-to-water HP filled in with R32 and validated it on a semi-virtual test bench (see paragraph 4.1).

The study presented in this paper is the first part of the COPACAIR project, funded by French Agency ADEME and dedicated to the on-field implementation of this performance assessment method. In a first part, the heat pump performance assessment method is presented, with its original principle and its necessary modifications to deal with thermodynamic cycles without any superheat at the evaporator outlet. In a second part, the on-field implementation of the method on an air-to-air heat pump installed in an office building is described. In a third part, a hardware-in-the-loop test bench, dedicated to air-to-air systems and installed in EDF R&D laboratory, is presented. Finally, results obtained on the test bench, based on measurement or calculated through the developed method are compared and analyzed.

2. Heat pump performance assessment method

2.1. Principle

Based on the refrigerant thermodynamic cycle, the in-situ refrigerant method described in [2] uses only non-intrusive 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 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, pumps, 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 mass flow rate is indirectly measured through the energy balance of the compressor:

$$\dot{W}_{comp} = \dot{m}\left(h_{comp,out} - h_{comp,in}\right) + \dot{Q}_{losses} \tag{4}$$

Where $\dot{W}_{comp}$ is the compressor power input measured, $h_{comp,out}, h_{comp,in}$ the enthalpies at the outlet and inlet of the compressor respectively, and $\dot{Q}_{losses}$ the compressor heat losses. Therefore, the refrigerant mass flow rate can be expressed as:

$$\dot{m} = \frac{\dot{W}_{comp} - \dot{Q}_{losses}}{h_{r,comp,out} - h_{r,comp,in}} \tag{5}$$

Niznik [4] 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}. Niznik [2] 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 compressor is:

$$\dot{Q}_{losses} = \left(\frac{\overline{Nu}_L k}{L} A_L + \frac{\overline{Nu}_{D,1} k}{D} A_D + \frac{\overline{Nu}_{D,2} k}{D} A_D\right)(T_{surf} - T_{amb}) + \sigma A_{tot}(T_{surf}^4 - T_{amb}^4) \tag{6}$$

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. Instrumentation

Besides temperature sensor(s) on the compressor shell to calculate the heat losses (T_{surf}), this method needs temperature and/or pressure sensors to obtain enthalpy values at various locations of the thermodynamic cycle. For that, surface temperature sensors are used (in green, figure 1). If no pressure sensor is available in the machine, low pressure and high pressure can be determined from the temperature measurements where the refrigerant is undoubtedly in a two-phase state: the evaporator inlet for low pressure and the center of the condenser surface for high pressure (in red, Fig.1). It should be noticed that the pressure measurement error that could be caused by the temperature glide of zeotropic fluids is very limited [2]. Finally, the electrical consumption of the compressor (W) must be measured. When the refrigerant at the inlet and outlet of the compressor is in a gas phase, temperature and pressure measurements allow to determine the enthalpy, then the mass flowrate via the compressor energy balance.

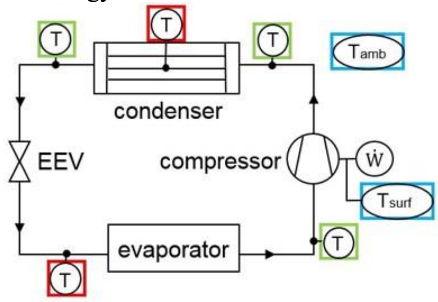

Figure 1: *Instrumentation for the performance assessment method*

2.3. Method's adaptation to the thermodynamic cycles without superheat

The method presented above therefore relies on non-intrusive temperature measurements on the refrigerant side to determine enthalpies at various points in the cycle, in particular at the compressor inlet and outlet. Knowing the enthalpies enables us to determine the refrigerant flow rate by means of an energy balance at the compressor. However, certain refrigerants such as R32, widely used in residential heat pumps, induce very high compressor discharge temperatures. When these refrigerants are used, the expansion valve opening does not control evaporator superheat, but rather the compressor discharge temperature, either directly or via discharge superheat. The consequence of this control logic is that the refrigerant is very often in a two-phase state at the compressor suction. Under these conditions, the instrumentation described above is not sufficient to determine the enthalpy of the refrigerant at this point in the cycle, as the vapour quality is unknown.

Noël [3] therefore proposed an adaptation of the method for two-phase suction. It is based on the expression of the refrigerant flow rate as a function of the compressor efficiencies, volumetric or global ones (equations 7 and 8), and according to equation 5.

$$\dot{m}_{\eta_{vol}} = \eta_{vol}.\rho_{suc}.C.F \tag{7}$$

ρ_{suc} is the refrigerant suction density, C the displacement volume of the compressor, and F the compressor rotating frequency. To calculate ρ_{suc}, the suction vapour quality is needed.

$$\dot{m}_{\eta_g} = \frac{\eta_g.\dot{W}_{comp}}{h_{comp,out,is} - h_{comp,in}} \tag{8}$$

$\dot{W}_{comp}$ is the measured compressor power input. The suction vapour quality is needed to calculate the suction enthalpy, $h_{comp,in}$, and the corresponding discharge isentropic enthalpy, $h_{comp,out,is}$.

It was showed [5], that using the volumetric or global efficiency with the hypothesis that the refrigerant is saturated underestimates the refrigerant mass flow rate. Using the energy balance (equation 5), and the same hypothesis of saturated refrigerant, the mass flow rate is then overestimated.Thus, the real mass flow rate value is necessarily between the efficiency and the compressor energy balance results. Therefore, iterations can then be made by progressively decreasing the vapour quality value to make the efficiency and the compressor energy balance calculation converge.

This improved performance assessment method has been implemented and tested in-situ.

3. On-field implementation of the method

3.1. Description of the installation

The method has been implemented on a monosplit 3.5 kW air-to-air heat pump, with R32 as refrigerant. It has been installed in a meeting room of an office building. This building is located at about 80 km from Paris. The following figure shows the installation of the indoor and outdoor units of the HP, the supervision equipment and the acquisition unit. One can observe that the measurements are almost invisible in the indoor side.

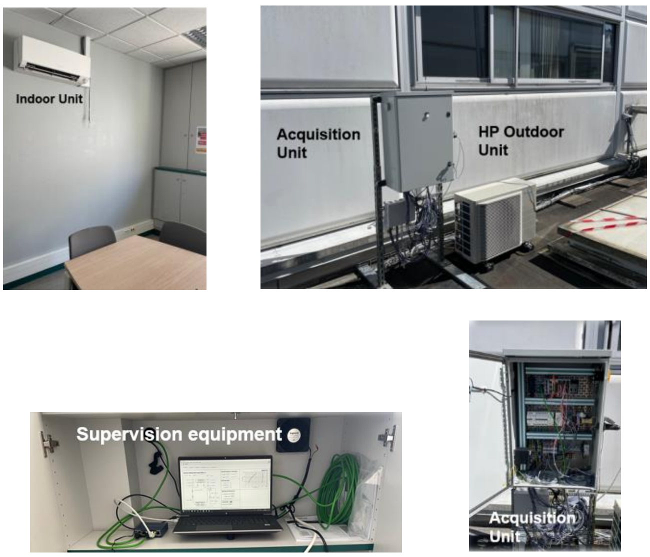

Figure 2: Main components of the installation

3.2. Measurement architecture

Fig. 3 describes the measurement architecture. Most of the sensors are installed in the outdoor unit (except the condensing temperature measurement). They are connected to the acquisition unit, which is itself connected to the supervision equipment.

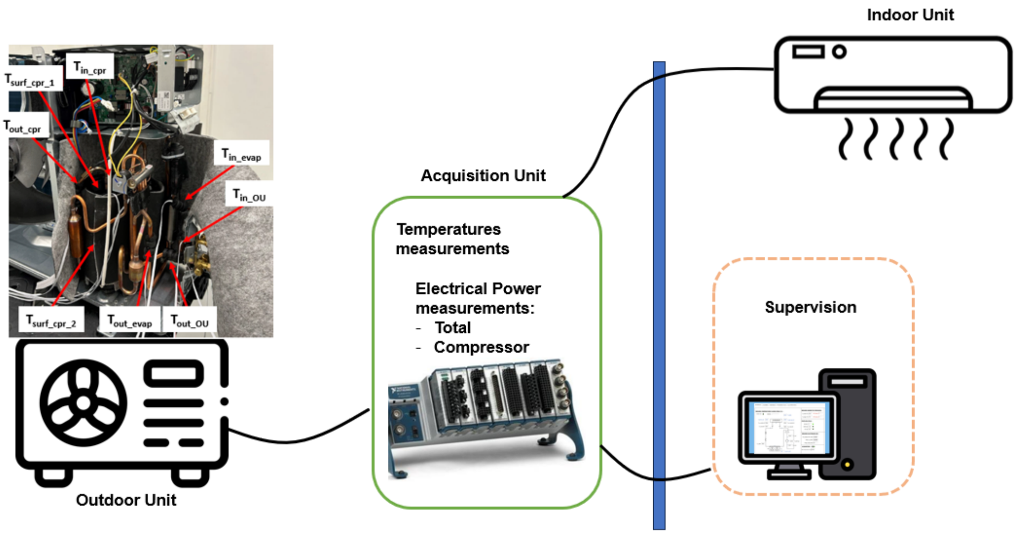

Figure 3: Measurement architecture

The sensors used are listed in Table 1 with the corresponding measurement uncertainty.

Table 1: Instrumentation for field test

Measured Parameter	Sensor	Precision
Refrigerant temperature	Contact PT100 1/10DIN	± 0.8 K
Electrical power	Wattmeter WT1800	± 1%

3.3. On-field results

3.3.1. Implementation of the method

To implement the performance assessment method, it is necessary to have the heat pump compressor mapping. This mapping describes the volumetric, isentropic, and global efficiencies of the compressor as a function of evaporation and condensation temperatures and its frequency. However, as the mapping of the heat pump compressor under test was not available, the mapping of a compressor used in another study was used [6]. This compressor uses the same twin-rotary technology and the same R32 refrigerant. It should be noted that this mapping was carried out for a compressor frequency between 30 and 90 Hz.

As specified, the refrigerant of the tested heat pump is R32. It implies a control of the expansion valve based on discharge superheat, and, therefore, no superheat at the evaporator outlet in current operation. As the compressor displacement was known, the improved method (iteration on the evaporator outlet vapor quality) based on volumetric efficiency was implemented to determine the refrigerant mass flow rate. For frequencies below 30 Hz, the volumetric efficiency at 30 Hz is used. In the same way, for frequencies above 90 Hz, the volumetric efficiency at 90 Hz is used.

3.3.2. Results

Data have been registered from March 2025, in heating mode. Fig. 4 shows indoor and outdoor temperatures, compressor frequency, heating capacity and integrated COP (ICOP on Fig.4), over a day. It highlights the capacity of HP to respect the setpoint indoor temperature of 21°C, by varying its operation via compressor frequency. However, the compressor frequency struggles to stabilize and is very low, constantly below 30 Hz, which is slightly outside the validity range of the mapping used. Relatively frequent compressor on/off cycles are observed when the outside temperature is above 7°C. This shows that the heat pump is significantly oversized for the room in which it was installed: a meeting room that is constantly closed with a lot of internal heat gain.

Fig.5 shows the COP calculated using the assessment method over two months of monitoring (March and

April 2025). The values are consistent and show a slightly positive trend in relation to the outdoor temperature. However, as described above, for outdoor temperatures close to or above 7°C, the compressor began to cycle on and off, which impacted the COP values.

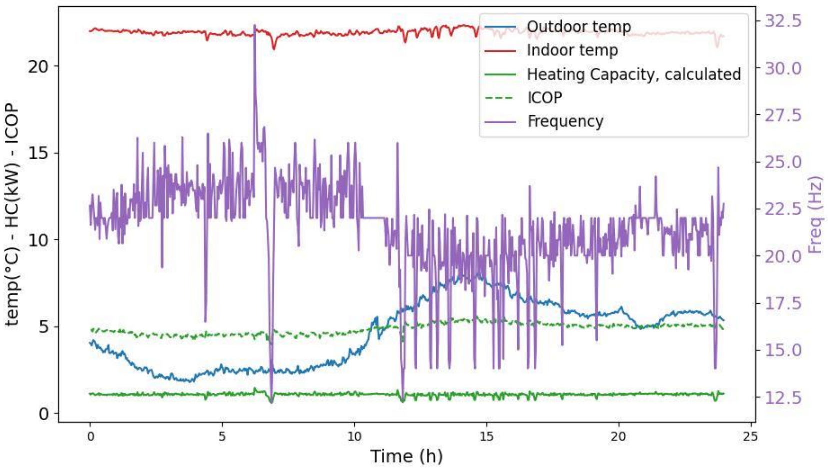

Figure 4: Evolution of system's parameters during a day

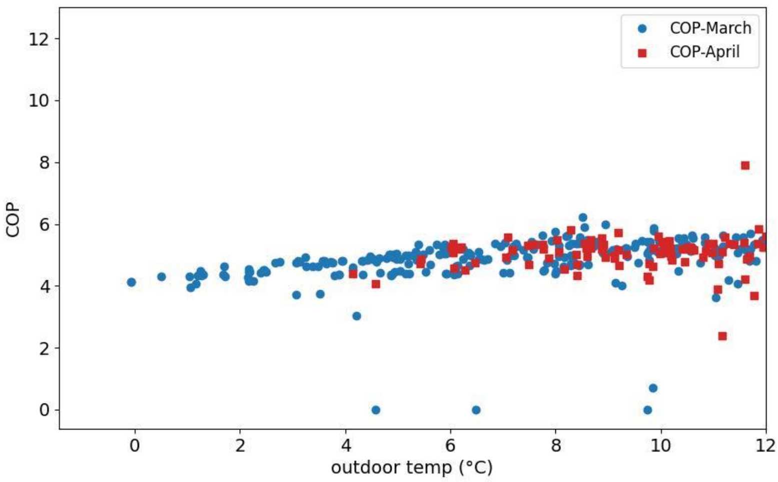

Figure 5: calculated COP as a function of outdoor temperature

This field monitoring therefore demonstrates that the method is entirely feasible to implement, as it is possible to visualize the heat pump operation and to calculate its performance using just a few sensors on the refrigerant side and compressor side. Moreover, the calculated performances seem to be coherent with the observed operation. However, this method deserves to be validated precisely. In particular, the impact of the default compressor mapping must be evaluated, particularly for low frequencies, that can be frequently met in case of oversizing.

This validation process has been realized on a semi-virtual test bench.

4. Evaluation on a semi-virtual test bench

4.1. Principle and installation of a semi-virtual test bench for air-to-air HP

The objective of semi-virtual test bench experimentation is to operate the equipment being tested in an environment that is close to real life, while still being controlled [7]. The principle of this test bench, installed in climatic chambers, is described in Fig. 6. A controller controls an outdoor weather scenario (temperature and humidity) applied in a test chamber inside which the outdoor unit of the heat pump is installed. This same controller sends temperature and humidity information to a numerical model of a building, which then calculates the indoor temperature of the building based on its heat loss and on the actual heating capacity delivered by the heat pump being tested. This temperature is applied in the test chamber where the heat pump's indoor unit is installed.

This principle assumes that it is possible to measure the power delivered by the air-to-air heat pump being tested. This measurement is carried out using the enthalpy method, by means of an air flow duct plenum connected to the indoor unit (Fig. 7). This includes a sleeve for measuring dry and dew temperatures, pressure, and air flow. The plenum is also equipped with a pressure compensation system to maintain atmospheric pressure at the outlet of the indoor unit. A fan located at the end of the plenum is controlled by the difference between the pressure in the plenum and atmospheric pressure. On the refrigerant side, the instrumentation required for the performance calculation method (Fig. 1) is supplemented by a mass flow meter and pressure sensors to obtain data to validate the heating capacity and refrigerant flow rate.

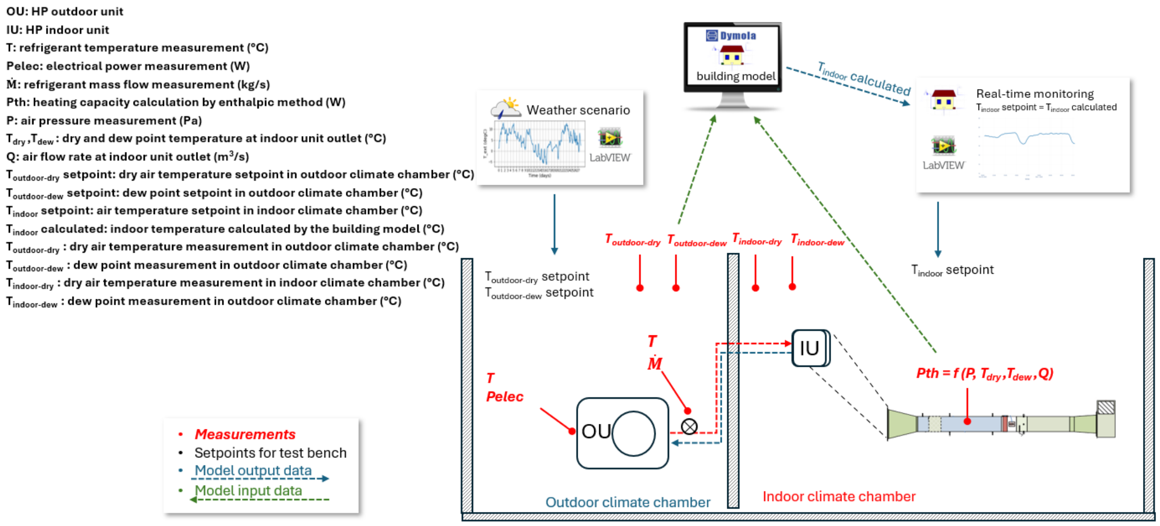

Figure 6: Semi-virtual test bench for air-to-air heat pumps

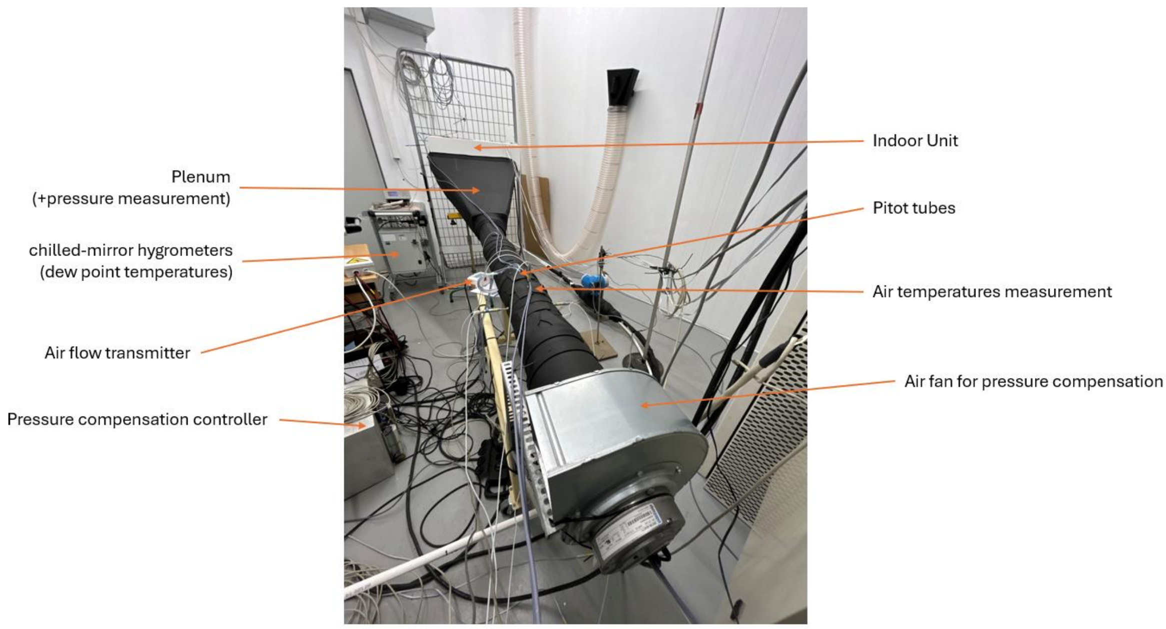

Figure 7: Air-flow duct plenum and indoor unit in the semi-virtual test bench

The semi-virtual test bench is equipped with a heat pump identical to the one used for field test. The building model is a simple mono-zone one, with heat losses adapted to the HP heating capacity, to avoid oversizing and to explore the entire compressor frequency range.

The sensors used are listed in Table 2 with the corresponding measurement uncertainties.

Table 2: Instrumentation

Measured Parameter	Sensor	Precision
Refrigerant temperature	Contact PT100 1/10 DIN	± 0.8K
Dry air temperature	PT100	± 0.1K
Dew point temperature		± 0.2K
Air flow rate		± 2 to 6%
Electrical Power	Wattmeter WT1800	± 1%

Due to the precision of the sensors showed in table 2, the reference heating capacity P_{th}, measured on air side, has a global uncertainty comprised between ±2 and ±8% depending on the air flow rate value (the uncertainty decreases when air flow rate increases).

4.2. Results – Behavior of the test bench and the HP

Several days of testing were conducted on the test bench described above. Fig. 8 shows the evolution of the outdoor temperature (in the outdoor climate chamber) and indoor temperature (in the indoor climate chamber), compressor frequency, and heating capacity, over one day of test. This figure shows the test bench's ability to:

- simulate a climate scenario in the outdoor climate chamber
- operate the heat pump realistically, according to its own control system, to maintain a stable temperature in the indoor climate chamber that is close to the setpoint.

Fig. 9 shows that the HP is able to stabilize its operation and varies the compressor frequency over a wide range of values (25-100 Hz).

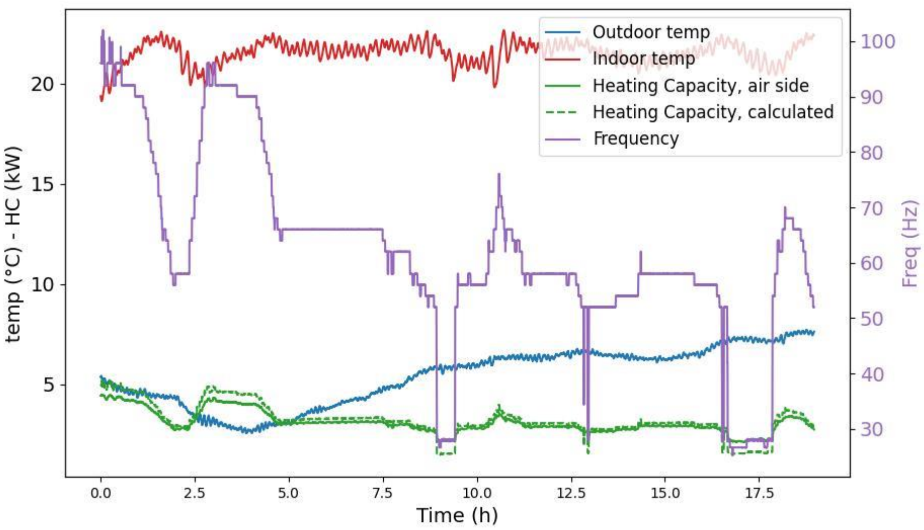

Figure 8: Temperatures, frequency and heating capacity over one day of test

4.3. Validation of the method on the semi-virtual test bench

Fig.8 also highlights the thermal capacity calculated using the performance evaluation method (green dotted curve) and measured on the air side (solid green curve). The values are very close, except for periods when the compressor frequency is below 30 Hz or above 90 Hz, i.e., when it is outside the validity range of the mapping.

However, at low frequencies, calculating the volumetric efficiency using a poorly adapted mapping is not the only source of error. In fact, significant noise was observed on the test bench for measurements below 30 Hz, affecting the accuracy of the frequency measurement.

Fig.9 and fig.10 show the values of heating capacity and COP, calculated using the method based on volumetric efficiency, as a function of the values measured on the air side and according to the compressor frequency, for the test illustrated in the Fig.8. These figures show similar results in terms of trend and Mean Absolute Percentage Error (MAPE) because the electrical power value comes from the same measurement.

These figures highlight the validity of the performance evaluation method, with good agreement between reference and calculated values (MAPE of 3.3%). Indeed, the average error is of the same order of magnitude as the uncertainty of the reference heating capacity measurement. However, these figures show a significant error during the few periods when the compressor frequency was very low. As already mentioned, this error is certainly due to the lack of precision in the volumetric efficiency calculation, as the compressor mapping is not well suited to these frequencies. The error may also be due to the problem of measuring low frequencies encountered on the test bench and mentioned above.

It should be noted that the error also increases for frequencies above 90 Hz, but to a much lesser extent than for low frequencies.

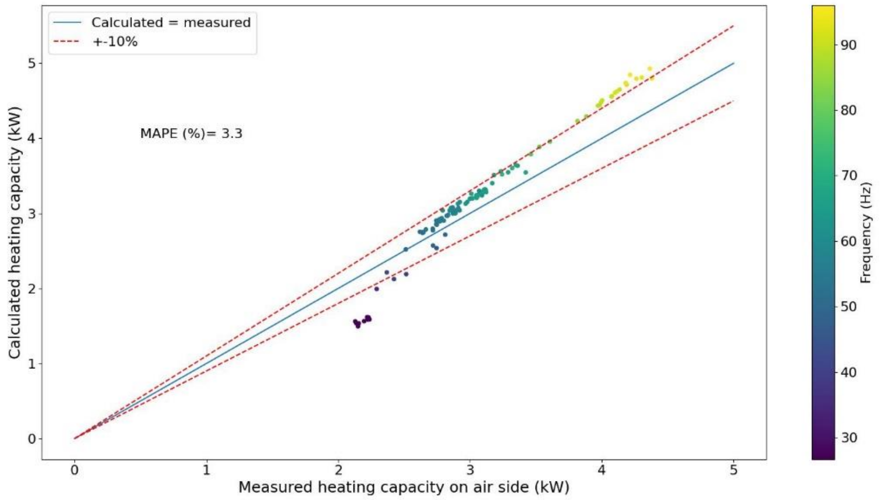

Figure 9: Reference and calculated heating capacities over one day of test

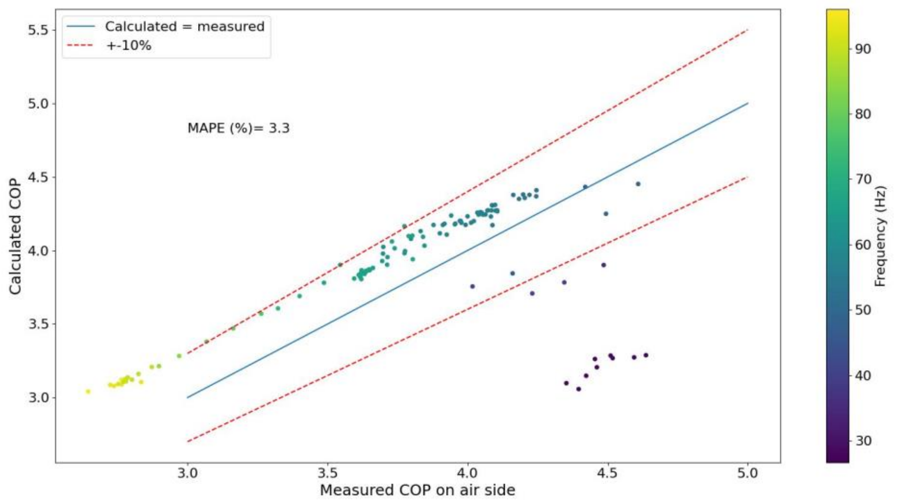

Figure 10: Reference and calculated COP over one day of test

5. Conclusion and perspectives

This study presents a method for evaluating the performance of heat pumps in real time. This method is based on energy balance at the compressor and involves only non-intrusive temperature measurements on the refrigerant side and measurement of the electrical power absorbed by the compressor.

This method was implemented during field monitoring in a meeting room in an office building near Paris, starting in March 2025. The results were perfectly consistent and interesting but showed that the heat pump was oversized compared to the heat losses of the room where it was installed. As a result, the compressor often operated at frequencies below 30 Hz. However, 30 Hz was the validity limit of the compressor mapping used to determine the volumetric efficiency expression required for the method.

These field results were then supplemented by an experimental study conducted on a semi-virtual test bench. This made it possible to carry out tests in near-real conditions, with the heat pump subjected to a real weather scenario reproduced in a climate chamber, and was operating based on its own control system to compensate for the heat losses of a virtual building. The building model was calibrated so that the heat pump was perfectly sized. Tests were then carried out to operate the heat pump over a very wide range of operating conditions in terms of compressor frequencies.

These tests demonstrated the validity of the method for evaluating heat pump performance in real time, with an average error in COP of around 3.5% over a one-day test.

These tests also revealed a deterioration in the results of the method at very low or very high frequencies, due to the lack of accuracy of compressor mapping at these frequencies.

Compressor mapping is therefore a key contributor to the accuracy of the proposed method and will require particular attention in future developments. The present results suggest that a generic mapping can be applied within its validity range, provided that the compressor belongs to the same technology class and uses the same refrigerant. Despite using a rotary R32 compressor from a different manufacturer than the one tested in the laboratory, the predicted performance remained highly consistent, indicating the potential for a more “universal” mapping applicable to compressors of the same category. Nevertheless, additional validation—especially on other heat pump units and under extreme operating frequencies—is necessary to fully confirm this conclusion. Besides this complementary validation, it could be useful to develop a library of compressor mappings, either for direct use or to calibrate a “universal” one.

Acknowledgements

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