Design And Validation Of A Heat Pump Digital Twin And Its Control Strategy Towards The Development Of Flexibility Focused Controllers

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ABSTRACT

In the context of global warming, the European Union (EU) is setting increasingly ambitious targets to phase down CO2 emissions. The designated path towards carbon neutrality especially includes the electrification of residential building sector thanks to massive deployment of Heat Pump (HP) in replacement of fossil fuel-based heating systems. Buildings, being widely electrified, could become a major actor of the electrical grid stability through the development of flexibility services. To that extent, many studies focused on flexibility provided by HPs especially in simulation environment. To do so, a virtual test bench is generally created to represent the system {HP + Building}. However, those studies tend to consider simplified representation of HPs and their embedded control strategy that might lead to some discrepancies when applying flexibility schemes.

In this paper, a complete methodology for the creation and validation of a specific HP digital twin on the Dymola software is proposed. This methodology relies on multiple test sequences realized in climatic chambers with an Air to Water HP. Firstly, manufacturer data completed by a series of static conditions tests are used to calibrate the HP model. Secondly, scenario-based Hardware-In-the-Loop (HIL) experimental tests offer complete validation of both HP dynamic behaviour and its control strategy. Thirdly, other test series reproducing off-grid periods are carried out in order to compare how the developed HP model performs compared to literature models while being subject to flexibility orders. Finally, this paper gives insight on the impact of the modelling accuracy of the HP and its internal control rules when accounting for flexibility; and therefore, for the development of flexibility designed controllers.

1. INTRODUCTION

HPs emerge as a promising solution in the objective of electrification of the building sector thermal uses. Indeed, HPs offer efficient means of utilizing electricity for heating and cooling purposes, thereby reducing reliance on fossil fuels and mitigating carbon emissions. The integration of heat pumps into buildings also highlights a significant opportunity to apply for flexibility services in order to help optimize energy production and consumption on the grid. This flexibility can be expressed in different ways either implicit - through time varying signals such as price - or explicit - through direct orders coming from a grid manager (Renté et al., 2022). Both approaches lead in the end to a time translation of the end user electrical demand. Previous studies explored the role of HPs in enhancing grid flexibility. In general, applying for flexibility relies on the creation of a specific higher-level flexibility designed controller in relation with the lower-level inner control system of the equipment. The development of such tools is primarily addressed through simulation-based approaches, as it greatly facilitates the development phase. Some papers went further and tested specific controllers on real machines in testing laboratories or in real buildings (Kuboth et al., 2020). In that context, the simulation step appears to be essential. In the literature, the proposed {HP + Building} models - virtual test benches - are generally simplified especially when considering inner HP control strategies. Indeed, (Clauß & Georges, 2019) proposed a review of different studies addressing implicit flexibility with HPs highlighting the main choices made when modelling HP and internal control rules. In the different studied papers, the HP dynamics are mostly oversimplified such as simplistic compressor control or idealistic Electrical Back Up Heater (EBUH). Whereas at the same time, the authors highlight a shift in performance when considering more and more complex approaches.

The same observation is brought by (Evens & Arteconi, 2021) in which a detailed HP control strategy is step by step designed to showcase its modelling importance in the context of flexibility leading to significant discrepancies between the different control formulations. In the end, control strategies commonly employed in simulation do not accurately represent behaviour which induces performances mismatch.
In this paper, the goal is to address these limitations by examining the importance of accurate modelling and realistic control strategies with an air-to-water HP in the context of explicit flexibility. Indeed, implicit flexibility has been extensively studied, therefore our work will be done in the context of explicit flexibility. This type of demand side management may result in a different behaviour of the HP. Thus, the primary objective of this work is to showcase a complete method to build and validate a dynamic heat pump model through lab experiments in a HIL test bench as seen in the literature (Evens & Arteconi, 2022; Tejeda De La Cruz et al., 2017). Then, the second goal of this work lies in an evaluation of three distinct levels of modelling in order to determine the importance of detailed modelling and fine-tuned control strategy. Through this analysis, we seek to assess the critical role of advanced modelling of HP performances and control techniques for the design and test of HP flexibility controllers.

2. METHODOLOGY

The goal of this work is to propose a methodology to parametrize and validate a fully working virtual test bench of a building and its embedded HP. To do so, the work presented in this paper relies in its core on multiple lab tests used to parametrize HP models and sub models. Those tests have been especially conducted in a HIL environment. Thus, the lab setup and tests scenarios will be presented first. Then, 2 different types of HP models (simplified and detailed approaches) will be shown. Finally, 2 different control strategies ruling the whole system, one simple and one detailed will be presented.

2.1 Experimental set-up

To calibrate and parametrize models, different kinds of test have been conducted in a HIL test bench. The global set-up is seen on Figure 1. The tested machine is composed of a monoblock outdoor unit: NIBE Air to Water HP S2125 – 8 kW and an indoor unit: NIBE VVM S320. The outdoor unit is mainly composed of the refrigeration circuit and its related components such as compressor and heat exchangers. The existing sensors, included in the machine, have been completed by contact temperatures sensors at key points of the refrigerant cycle. The compressor, fan and pump electrical consumption, air and water temperatures and flowrates are also measured. The precision of each sensor type is presented in Table 1.

Table 1 : Lab measurement uncertainties

Type	Precision
Intrusive temperature sensor PT100 1/10 DIN	± 0.08K
Contact temperature sensor PT100 1/10 DIN (LNE, 2004)	± 0.8K
Wattmeter WT1800	± 10 W
Flowmeter	± 18 m3/h

The outdoor unit is linked to the indoor unit through hydraulic pipes. The indoor unit consists of a Domestic Hot Water (DHW) tank, the EBUH, the circulation pump and the HP controller. It should be noted that the DHW is not included is this study. After the indoor unit, supply water flow is sent to a buffer tank. The temperature and flow value at the outlet of the indoor unit are measured and used as inputs for the virtual part of the test bench (Figure 1). Indeed, a model of a house and its distribution system is simulated in real time on Dymola. The case study is a 100m² detached house with hydronic radiator system which is modelled as a monozone using EDF BuildSysPro Dymola library (Plessis et al., 2014). The model can be parametrized in terms of building materials and emitter types. The model uses the flow information collected previously as inputs and calculates the resulting return temperature and flow. A thermal device is then installed to actively match the return temperature with the calculated value.

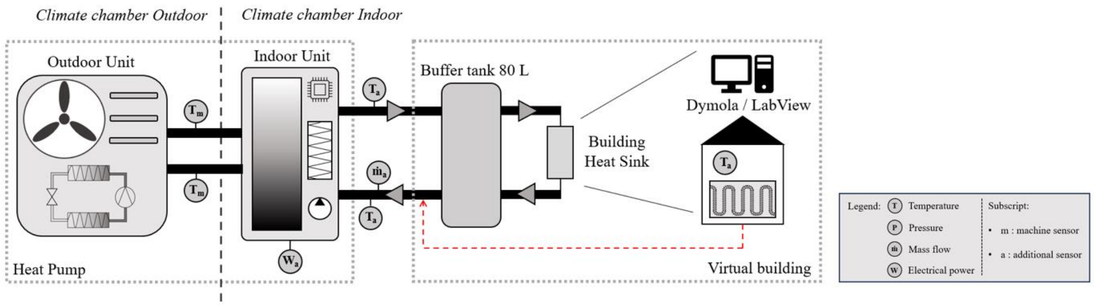

Figure 1: Schematic of the HIL test bench

A first series of tests has been conducted in order to determine static HP performances without the virtual part activated. The objective was to characterize the HP performances and operation parameters in various operating conditions, described in Table 2.

Table 2: Static test operating conditions

Outdoor air temperature	Frequency	Return water temperature
{12, 7 , 2, -2, -7, -12} °C	{20, 40, 60, 80} Hz	{20, 30, 40, 50, 55} °C

To evaluate and validate our model, additional dynamic tests on the HIL test bench have been conducted. Those tests used 2 weather scenarios from French regulation RT2012 (CSTB, 2012) of 1 week, a warm one and a cold one, as presented in Figure 2.

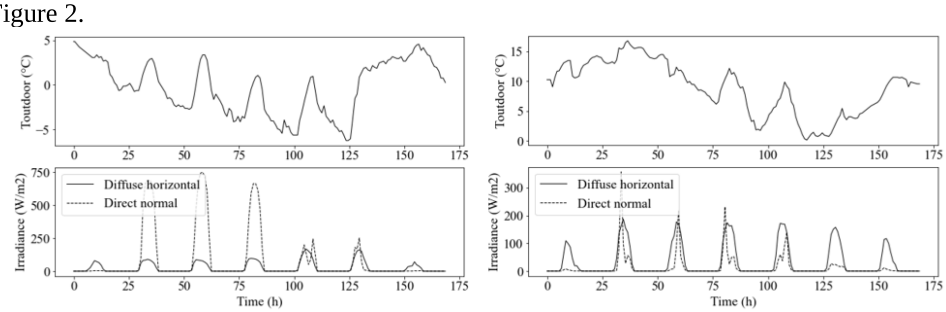

Figure 2: Cold (on the left) and warm (on the right) weather sequences

During the warm test, the indoor sensor control was disabled. The HP was only relying on the heat curve to calculate the setpoint for the supply water temperature. During the cold test, the indoor correction was activated with a user setpoint constant at 21°C over the seven days to better characterize the indoor sensor influence on the controller.

2.2 Heat Pump models

2.2.1 Detailed HP model: To build the detailed model of the HP, the choice has been made to adopt a component-based approach by use of the TIL library (Gräber et al., 2010). The modelling of heat exchangers relies on mass, energy, and momentum balance equations. This modelling approach uses 1D finite volume elements. This library provides templates to fill with real geometrical parameters of heat exchangers. One should note that these templates do not allow to deal with specificities such as asymmetrical condensers or complex evaporator piping structures. The evaluation of the fluid states relies on heat transfer correlations. Concerning the evaporator, the choice has been made to work with the "Shah Chen" correlation (Collier et al., 1996) on the refrigerant side, and the "Haaf" correlation (Haaf, 1988) on the air side. Concerning the condenser, the choice has been made to use the "Longo" correlation (Longo et al., 2015) on the fluid side and "VDI" correlation (Martin, 2010) on the water side. To determine the discretization value a short study shows that above a given value, the results tend to converge. Therefore, a discretization step of 5 has been chosen for both evaporator and condenser to balance between precision and computational burden. The electronic expansion valve is modelled thanks to Bernoulli equation and is controlled by a PI controller adjusting the superheat refrigerant value at evaporator outlet. The superheat setpoint is calculated according to the manufacturer rules. Finally, to complete the cycle, a compressor model has been created. This model relies on correlations for the isentropic efficiency (η_{is}), the volumetric efficiency (η_{vol}) and the global efficiency (η_{glob}) defined in (Försterling et al., 2005). The data used to calibrate those correlations comes from the static test conditions without the defrost sequences to better map compressor performances during heating periods. It should be noted that the refrigerant mass flow had to be deduced from measurements by using energy balance at the condenser as there is no flowmeter on the refrigerant side. The efficiency equations (1) are presented as follows:

$$\eta_{is} = \frac{h_{out_{is}} - h_{in}}{h_{out} - h_{in}} \qquad \eta_{vol} = \frac{\dot{M}}{V_{displ} * f * \rho_{in}} \qquad \eta_{glob} = \frac{\dot{M}\left(h_{out_{is}} - h_{in}\right)}{\dot{W}} \tag{1}$$

To determine the correlation structures, the choice has been made to work with the AHRI 20-coefficient polynomial expression (2) (Ossorio & Navarro-Peris, 2023) as follows:

$$\begin{aligned}\eta_i = a_1 + a_2T_e + a_3T_c + a_4f + a_5{T_e}^2 + a_6T_eT_c + a_7T_ef + a_8{T_c}^2 + a_9T_cf + a_{10}f^2 + a_{11}{T_e}^3 \\ + a_{12}{T_e}^2T_c + a_{13}{T_e}^2f + a_{14}{T_c}^3 + a_{15}{T_c}^2f + a_{16}{T_c}^2T_e + a_{17}f^3 + a_{18}f^2T_e + a_{19}f^2T_c + a_{20}T_eT_cf\end{aligned} \tag{2}$$

With T_e (K) being the evaporating temperature, T_c (K) being the condensing temperature, f (Hz) being the frequency and a_i being the polynomial coefficients. The proposed approach led to the results presented in Figure 3.

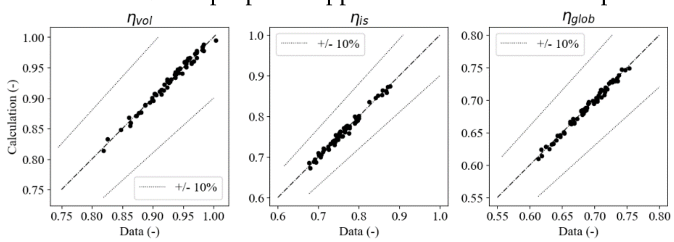

Figure 3: Correlation fitting of compressor efficiencies.

2.2.2 Simplified HP model: On the other side, a simplified polynomial approach to model the whole HP has been developed. For this purpose, the choice has been made to work with another dataset provided by the HP manufacturer consisting of 55 points for: outdoor temperature in [-25°C, 15°C], supply water temperature in {25, 35, 45, 55, 65 °C} at low, medium, and high capacities (values depending on compressor envelopes). The reason that pushed the use of an alternative dataset lies in the experimental procedure used (*EN 14511-2*, 2018) that ensures a better replicability of test conditions, in particular with the initial frosting state at the evaporator. Polynomial expressions (3) for heat capacity and electrical power according to 3 different inputs have been fitted:

$$X = b_1 + b_2f + b_3T_{out} + b_4T_{supply} + b_5f^2 + b_6{T_{out}}^2 + b_7{T_{supply}}^2 + b_8fT_{supply} + b_9T_{out}T_{supply} + b_{10}fT_{out} \tag{3}$$

One should note that the dataset used inherently includes the impact of frost/defrost in the registered performances. The resulting fitting performances are shown in Figure 4.

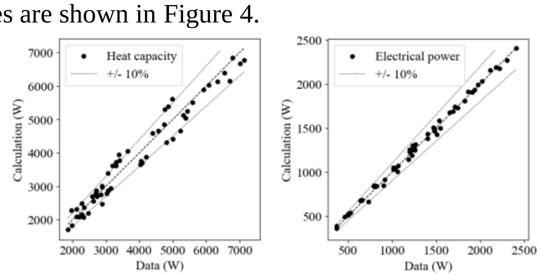

Figure 4: Correlation of HP electric power and heat capacity

2.3 Control Strategies

2.3.1 Detailed strategy: One of the main parts of the {HP + Building} model is the control strategy ruling the whole system. The control strategy is based on 3 principal inputs, the outdoor air temperature, the supply water temperature, and the indoor air temperature. Knowing that, a specific effort has been made to develop in Dymola a close representation of the real control strategy implemented in the controller. The developed control especially accounts for:

- Degree Minutes Logic (Supply water temperature difference with supply setpoint integrated through time)
- Preheating phase of the compressor and start-up procedure
- Compressor frequency controls (PI, compressor envelopes, filtering)

- EBUH stages control
- Indoor temperature correction
- Defrost strategy (only possible with the detailed HP model): see section 2.4

2.3.2 Simple strategy: As highlighted in introduction, HP control strategies in literature tend to follow straightforward approaches. Indeed, in practice, manufacturers do not generally provide details of how their controller work. In this paper, we would like to be able to compare a detailed control strategy as presented above with a simplified strategy. To do so, a simplified approach is defined as follows:

- Supply temperature setpoint calculated through manufacturer heat curve
- HP regulating between 20 and 80 Hz + 15 min start up procedure
- Hysteresis based on user setpoint and indoor air temperature : Band [-1,+1]°C
- EBUH 5kW modulating from 0 to 100% & Hysteresis based on user setpoint - 1.5°C : Band [-0.5,+0.5]°C

2.4 Defrost events modelling

In addition, to the detailed control rules explained above, a method to include dynamic defrosting sequences in the virtual test bench is proposed. Accounting for a physical detailed frost and defrost model is a time-consuming and challenging topic. On the one hand, the aforementioned modelling approach has some limits. The 1D approach may be too simplified to represent frosting, as frost on the evaporator fins is not a homogenous phenomenon. Moreover, the defrosting relies on a cycle inversion which is a dynamic phenomenon very difficult to model. On the other hand, according to lab measurements, it appears that in most of the tested conditions the impact of frost is not detectable. This might be due to the manufacturer's choice to push defrosting sequences in a preventive way before extensive ice accumulation. Additionally, the aim in this work is to account for frosting and defrosting effect on HP performances from an energy point of view. Thus, modelling the refrigerant behaviour during defrost events seemed out of the scope. To that extent, the choice has been made to develop a simplified defrosting model matching manufacturer defrosting method using only few sensors accessible from the controller measurements. Through the different tests made, a dataset of 181 defrost sequences has been selected. The goal of the developed model is to propose a representation of the defrosting energy burden on water distribution system by use of a limited number of variables. To do so, we aim to develop a model giving the duration Δt, the HP electrical power P_e and the negative heat capacity P_h at the condenser according to environmental inputs.

At the same time, we expect to know when defrost is happening as we implemented the defrosting strategy within the detailed controller. The data used for this part is shown in Figure 5. The electrical power and heat capacity are assigned averaged values over the entire defrosting sequence. For some defrosting events, the heat capacity is higher than 0 W which is counter intuitive. However, this can be explained by the fact that the overall defrost sequence is actually wider than the proper cycle inversion which can in the end lead to an averaged positive heat capacity.

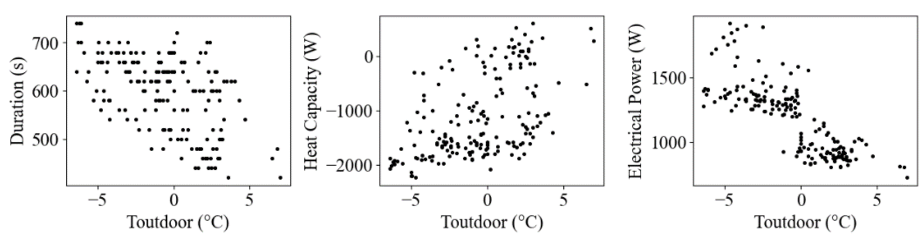

Figure 5: Experimental results of main outputs characterizing HP defrost in function of outdoor temperature

To create our model, identification of relevant inputs is needed. In this paper, we preselected 8 different possible variables presented as follows (Table 3).

Table 3: Preselected variables for defrost modelling

Tout	Outdoor temperature	TevapIn	Refrigerant temperature at evaporator inlet
Tsupply	Supply water temperature	TevapOut	Refrigerant temperature at evaporator outlet
HR	Relative humidity	DtEvap	Refrigerant temperature difference between evaporator outlet and inlet
Freq	Compressor frequency	LP	Low pressure

Those are expected to have a significant impact on defrosting dynamics. Additionally, those variables are registered at the start of defrosting. To determine what are the most important features for our 3 different outputs, we conduct a sensitivity analysis (Figure 6).

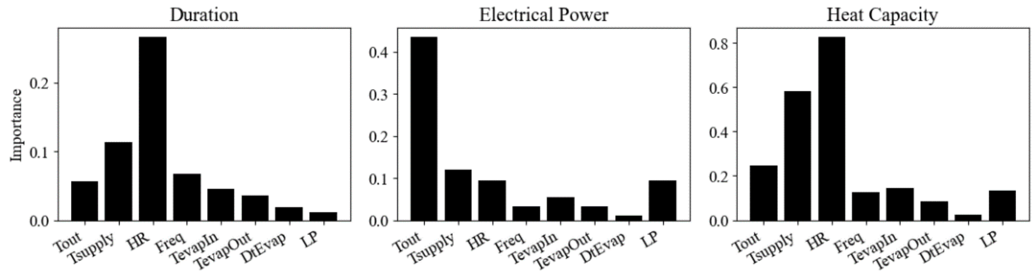

Figure 6: Features sensitivity analysis for the 3 different outputs

To choose the pool of parameters, the selection focuses on the most sensible ones with a trade-off according to their inherent dependencies. In the end, all this procedure leads to this set of equations (4):

$$\Delta t = f(HR, Tsupply, Freq) \qquad Pe = f(Tout, LP, HR) \qquad Pq = f(HR, Tsupply, Tout) \tag{4}$$

The model structure for each of them is a quadratic 10-polynomial expression as previously highlighted in equation (3). To evaluate the model, the complete dataset is split into learning and validation subsets with a 70/30 ratio. The resulting RMSE, the NRMSE and the standard deviation are computed over the validation dataset. The results are presented in Table 4. In the end, validation results show that the model succeeds in predicting the energy impact of defrost operations according to limited and accessible inputs conditions. Furthermore, other selections for the learning and validation datasets showed comparable performances.

Table 4: Defrost model performances

	RMSE	NRMSE	Standard deviation
Heat Capacity	412 W	14.5 %	217 W
Electrical Power	180 W	15.1 %	106 W
Duration	40 s	12.6 %	16.5 s

3. RESULTS & DISCUSSION

3.1 Heat Pump Model Validation

The detailed heat pump model presented in section 2.2.1 will be validated using the cold weather scenario (Figure 2). To complete the validation process, the needed inputs are collected from the lab measurements and then applied in the simulation, including: compressor frequency, defrost state, return water temperature, humidity… In the end, only the HP inherent performances and heat capacity are being evaluated without any control strategy involved. The corresponding results are shown in Figure 7.

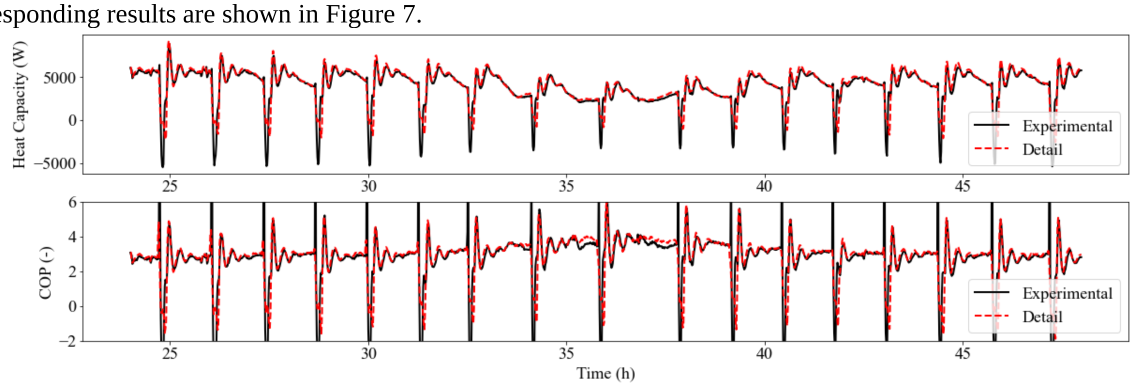

Figure 7: Dynamic detailed model comparison experimental data: focus on 24h

When comparing dynamically the heat capacity and the COP, the detailed HP model reproduces the dynamic of the real machine thanks to an accurate modelling of the hydraulic inertia between indoor and outdoor units. Moreover, when looking at the overall energy consumption over the whole week sequence, the HP model produces 446 kWh of heat and consumed 160 kWh of electricity. In comparison, the lab sensors registered overall energies of 427 kWh and 159 kWh for respectively thermal and electrical dependencies. In the end, less than 5% deviation in the obtained results are considered satisfactory.
Concerning the simplified HP model presented in section 2.2.2, it was validated in static conditions (Figure 4). However, this model is not validated in dynamic conditions as it could not handle dynamically real defrosting sequences.

3.2 Virtual test bench performances

To study the importance of detailed modelling and fine-tuned control strategy, 3 different assemblies of virtual test bench are tested:

- Simple model and simple control (A1)
- Simple model and detailed control (A2)
- Detailed model and detailed control (A3)

It should be noted that control strategy in A2 and A3 still differs due to the defrost strategy that is explicitly included in A3 and implicitly included in A2 through global efficiency. More specifically, while there are defrosting sequences in A3, the HP in A2 runs in the heating mode all the time. Moreover, the model initialization does not fully match the experiments which inevitably leads to some dynamic and energy differences in the results. Dynamic behaviours of the different assemblies are compared to the experimental measurements on the cold sequence in Figure 8. In regard to section 3.1, it should be noted that here the frequency is no longer an input to the model, but it is set by the controller model.

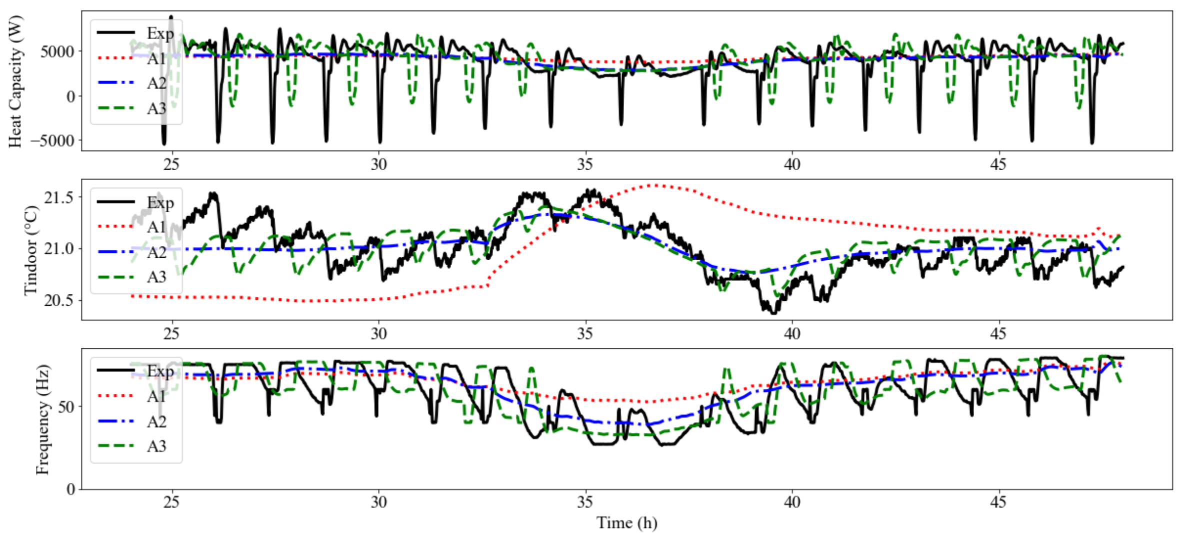

Figure 8: Dynamic comparison of the different assemblies : focus on 24h simulation/test

As A3 takes into account defrost sequences, it is the most accurate assembly for following and reproducing the real-life behaviour. A2 globally follows the different profiles from lab tests but does not follow the defrost dynamics and then misses their impact on indoor temperature. A1 shows the least accurate dynamic response with a clear deviation for indoor temperature in some occurrences. To complete this analysis, Table 5 highlights an energy analysis as well as the ability to account for dynamic events such as On-Off cycles and defrost cycles for both weather profiles presented in Figure 2. All assemblies give good energy performances with A3 providing the best results when considering both weather conditions. Additionally, the best scenario at accounting for dynamic events is also A3 with better matching for On-Off cycles and good estimation of defrosting sequences. The mismatch between experimental measurements and both A1, A2 assemblies could be explained by the precision of the statistical HP model used. Indeed, as shown in Figure 4, the obtained model often reaches significant inaccuracies especially at low heating load. Thus, it could lead to the 12.9% and 10.4% relative errors observed for the COP calculation during the warm winter week.

Table 5: Simulation and experimental results for warm and cold weather sequences

	Cold week					Warm week				
	Electrical (kWh)		COP (-)		Defrost Cycles (-)	Electrical (kWh)		COP (-)		On-Off Cycles (-)
Exp	159		2.69		70	69		4.72		23
A1	169	+6.3%	2.60	-3.3%	-	75	+8.7%	4.11	-12.9%	6
A2	167	+5.0%	2.61	-3.0%	-	74	+7.2%	4.23	-10.4%	36
A3	156	-1.9%	2.80	+4.1%	76	67	-2.9%	4.70	-0.4%	32

From the presented results on both weather conditions, A3 seems to clearly outperform the two other assemblies both in term of energy calculation and especially in term of dynamic response. When comparing A1 and A2 results, we see that the detailed control modelling is not improving the energy consumption that much but better replicates the HP behaviour. In the end, A1 with its simple modelling of HP and its control strategy provides good energy estimation. Therefore, this kind of model is well adapted to estimate energy consumption and production weekly or even annually, especially as it is fast to compute. However, when the goal is to closely represent the HP dynamic behaviour, A2 and A3 better match the indoor climate profile. Nonetheless, the A2 approach with the simple HP model cannot include all control laws especially lower-level ones as the model does not represent the refrigerant states within the cycle. This is especially true for the defrost strategy that relies on direct refrigerant measurements. Therefore, the A3 detailed HP modelling is needed to address replication of defrost dynamics and other in-depth HP specificities.

3.3 Flexibility test comparison

This section focuses on testing the three assemblies presented above in the context of explicit flexibility. To do so, an additional lab test has been conducted with the HIL test bench over 5 days. This time, the heat pump is parametrized to stop the compressor and EBUH between 8 to 9 a.m. and 6 to 8 p.m. This mimics an active order coming from the grid operator to achieve electrical load reducing. The resulting lab test is presented in Figure 9. Interesting HP dynamics can be outlined especially the compressor dynamic during the recovery phase after the shutdown order. Indeed, when focusing on 1 day, the compressor does not start directly at the end of the order. In fact, it starts its preheating phase – compressor crank heater is heating up the oil to ensure proper viscosity for optimal lubrication. In the end, the HP only relies on the EBUH to provide heat for a significant period.

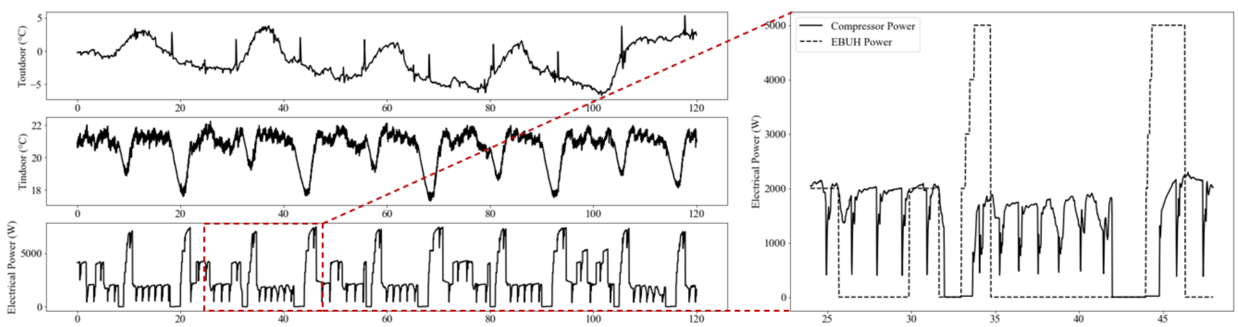

Figure 9: Explicit flexibility lab test results (left) and an electrical energy analysis focus on 24h (right)

The flexibility scenario is then applied to the 3 different assemblies. The dynamic profiles over 1 day are shown in Figure 10 and the weekly energy analysis is made in Table 6. Significant differences between simplified and detailed approaches appear in the context of explicit flexibility with the A1 assembly underestimating by more than 27% the total electrical consumption. On the contrary, A2 and A3 assemblies showed less than 10% deviation compared to experiments. When looking specifically at the rebound effect after the shutdown, the A1 assembly has a shorter recovery phase because there is no preheating in the controller, so the compressor starts right at the end of the order as it shown on the frequency profile. This behaviour leads to overestimate the HP capability to recover from the shutdown. A2, A3 and experimental test are subject to the compressor preheating and show a bigger electricity consumption during that phase due to the additional work made by the EBUH. It should be noted that the controller rules are not perfectly implemented in the controller model for A2 and A3 which could lead to the observed differences with the experimental scenario. A gap between the different assemblies is highlighted here with the simple control approach providing poor performances both in terms of dynamic response and energy. Therefore, detailed control modelling seems key in applying flexibility measures in simulation as they account for complex HP dynamics. The results underline the need to implement reality-based control strategies for creating virtual test benches in order to ensure proper translation of simulation results into real life experiments.

Additionally, even if A2 and A3 do not show significant differences in term of energy, it should point out that the defrosting dynamics are important to consider because it has a specific impact on water temperature and therefore on heating regulation and the triggering of EBUH (as shown in Figure 8, the variation of the indoor temperature during the defrosting periods is perceptible). This aspect is all the more important to consider when coupling with other dynamic events such as DHW charge because the combined impact of DHW production and defrosting on water temperature within the hydraulic loop can be massive.

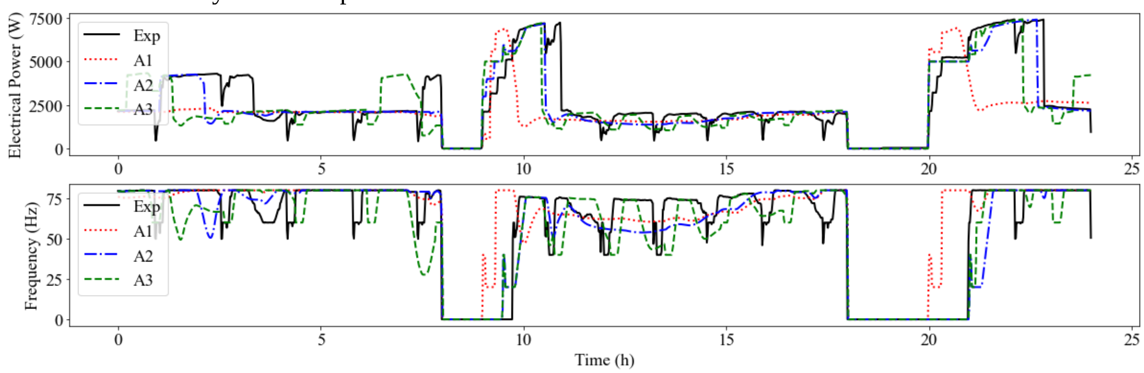

Figure 10: Flexibility scenario electricity power and frequency profiles: focus on 24h

Table 6: Flexibility test/energy results over the whole test sequence

	Electrical (kWh)		COP (-)	
Exp	319		1.72	
A1	231	-27.6%	2.14	+24.4%
A2	295	-7.5%	1.78	+3.5%
A3	289	-9.4%	1.82	+5.8%

Furthermore, one should note that the compressor preheating problem highlighted in this section especially applies for air sourced HPs. This behaviour might restrain flexibility potential from existing HP. Therefore, this preheating strategy must be discussed with regard to flexibility needs. Multiple solutions such as continuous heating of compressor crank case or minimum working frequency during shut down orders might be sufficient to help HP better answer to grid needs. In addition, the energy overconsumption during the recovery phase might also be mitigated through a better consideration of the incoming order. For example, one should be able to control the HP to preheat the building before the order. These considerations highlight the need for creating specific flexibility designed controllers able to address the presented difficulties.

4. CONCLUSION

This paper discussed the importance of detailed modelling of HP performances and their inherent control strategies in the context of explicit flexibility. Three different model assemblies {HP model + Control} of various accuracies have been developed with the help of extensive lab testing - especially in a HIL environment. Those models have then been tested under different weather scenarios and especially under an explicit shut down order scheme. In-depth comparison between simulation and experimental results highlighted the importance of precise modelling of HP and more importantly its control strategy when addressing explicit flexibility. Moreover, only the detailed HP model is able to account for defrosting sequences, which can have an impact on comfort and EBUH triggering. Additionally, on air sourced HPs, it seems necessary to adapt the HP response to an explicit order to avoid the negative rebound effect resulting in a delay in HP restart while the oil warms up, compensated by the lower performance of the EBUH. In further work, the goal would be to also include DHW production within the developed model assemblies as it might have a significant impact on flexibility potential. As perspective for future research, this study underlines the need for proper flexibility designed controllers to increase HP capability to take part in flexibility schemes.

NOMENCLATURE

h_{in}	compressor inlet fluid enthalpy	(J.kg^{-1})	T_e	evaporating temperature	(K)
h_{out}	compressor outlet fluid enthalpy	(J.kg^{-1})	T_c	condensing temperature	(K)
$h_{out_{is}}$	compressor outlet isentropic fluid enthalpy	(J.kg^{-1})	η_{is}	compressor isentropic efficiency	(-)
V_{displ}	compressor volume displacement	(m^3)	η_{vol}	compressor volumetric efficiency	(-)
ρ_{in}	compressor inlet density	(kg.m^{-3})	η_{glob}	compressor global efficiency	(-)
P_e	HP electrical power	(W)	f	compressor frequency	(Hz)
P_h	HP heat capacity	(W)			

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