



Impact of heat pump controller in the performance: experimental comparison of EN14825/EN14511 and compensation method of four different air-to-water heat pumps

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Abstract

In this study, we present the impact of the controller on the performance of various heat pumps by comparing the results evaluated using two distinct test methods: the first is the standard EN14825:2022+EN14511:2022 and the second is the two-mass model of the compensation method.

The standard prescribes the test method and the calculation procedure of the seasonal performance factors. With this method, the laboratory controls the temperature supply according to the tables defined for the test cases, and the controller of the heat pump is not included in the test boundary. Instead, the controller is included in the test boundary of the two-mass model of the compensation method, which emulates the load: the return temperature depends on the emulation result, and the unit controls the supply temperature. The laboratory adjusts the heating curve to obtain the same average supply temperature as foreseen in the standard. The standard method considers steady-state performance (with the only exception of defrost) while the second method considers dynamic working conditions.

Four different heat pumps were tested under different part load points considering the design capacity declared by the manufacturer. The test considered average climate conditions and low- and medium-temperature applications.

The comparison done with the two methods reveals that the influence of the controller is different between the tested units and the test points. Those differences can be seen in the defrost control and the capacity control in part-load conditions. The results indicate a COP difference in the test points ranging from 0% to 10% for one unit and up to 30% for another unit. The implications of these findings demonstrate the importance of including the control in the test boundary for the development of heat pumps.

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Selection and/or peer-review under the responsibility of the organizers of the 15th IEA Heat Pump Conference 2026.

Keywords: Compensation method ; Two mass model; Impact of heat pump controller; EN 14511

1. Introduction

In Europe, the building sector is responsible for approximately 40% of primary energy consumption and 36% of related greenhouse gas emissions [1]. Heat pumps are recognized as an effective solution for decarbonizing the building stock.

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It is relevant to promote the market penetration of energy-efficient systems. In this direction, local or national incentive programs are often based on the heat pump energy efficiency calculated at nominal conditions or at seasonal level. The heat pump efficiency is estimated from laboratory testing following standard test methods. The current standards for residential heat pumps are based on steady-state tests performed under predefined boundary conditions. In Europe, residential heat pumps are tested according to EN 14511-3:2022 e EN 14825:2022 [2,3]. The first standard defines the test method, while the second standard defines the method for the part load test and the calculation of the seasonal efficiency. Following these standards, the laboratory sets the compressor frequency (and sometimes the expansion valve and the fan speed) in order to obtain the part load, which is linearly dependent on the design load and the outdoor temperature. When the heat pump reaches the minimum modulation and the load is lower, deviations outside the permissible deviations for load and supply temperature are accepted to ensure continuous operation and degradation coefficients are considered for the suppressed on/off operation.

Operative and dynamic boundary conditions impact the performance showing a wide variation of performance factors [4]. Therefore, as an alternative approach to steady-state, dynamic or load-based testing methodologies have been introduced for the laboratory assessment of the performance, including the heat pump control in the test boundary. These methodologies rely on reproducing predefined load curves or replicating representative building loads calculated in real time with a numerical model.

Menegon et al. [5,6] developed a dynamic test method that reproduces in a laboratory environment a short test sequence representative of the seasonal boundary conditions. The methodology initially was based on simulations that give pre-defined test wave forms, and later it was evolved to a load emulation in the “Prescribed Load - Performance Extrapolation” PLPE method [7].

Palkowski et al. [8,9] proposed a load-based test method (compensation method) for measuring the performance of residential heat pumps and air conditioners. The method was influenced by the inertia of the laboratories, affecting the reproducibility of the method [10]. In the second version, the emulation of the building based on a two-mass model enables the representativity of the building load and the reproducibility of the method [11]. The results of the validation of the method is presented in [12]. The validation of the method was performed with the test guideline of 2023 [13] while the last version of the method is dated July 2025 [14].

Cremaschi and Perez Paez [15] proposed a method for the test of heating, ventilation, and air conditioning that directly controls the test room, generating different heat gains simulating the different building loads.

Dhillon et al. [16] developed a load-based testing methodology for air conditioners, which was validated comparing the performance on a residential house and in laboratory. The method is proposed as draft for the testing standard CSA SPE07 (Canadian Standards Association).

Giannetti et al. [17] and Dondini et al. [18] proposed a load-based testing methodology with a room emulator for the performance measurement of residential air conditioners. Following this method, Giannetti et al. [19] proposed a feed-forward compensation for emulator-type testing facilities in order to improve the reproducibility of the load-based test. As result of the feed-forward compensation, the method enhances the reproducibility to limit performance deviations to within 2 %.

The paper presents the impact of the controller on the performance of four heat pumps by comparing the results obtained using the standard EN14825:2022+EN14511:2022 and the two mass model of the compensation method in the version of 2023 [13]. In the standard test the controller is bypassed by the laboratory while in the compensation method the controller operates in a real-like condition (of a system with defined thermal capacity and load). Therefore, with the standard method the measured performance can deviate from the actual operating performance. The controller has effects on the modulation of the heat pump components, the on/off cycling and the defrost control [20–22]. The paper investigates the effect of the controller on four reversible air-to-water heat pumps present in the market.

Section 2 presents the method for the evaluation of the impact of the controller, presenting the two test methods and the characteristics of the four heat pumps. Section 3 presents the results of the application of the two test methods and discuss the implications of considering the controller in the test boundary.

The tests were carried out at the Heat Pumps Lab of Eurac [23] with the financing of the European Regional Development Found “SustainHEAT” founded by the Autonomous province of Bolzano.

2. Method

The impact of the controller has been evaluated applying the test method of the EN 14511-3:2022 and EN 14825:2022 [2,3] and the two-mass model of the compensation method [13].

The two methods considered the same part load condition defined according to the EN 14825:2022 as a function of the outdoor air temperature. The tests were set up in order to obtain the same heating capacity, and we evaluated the difference in COP and power input. As the standard foresees 10% of tolerance for the validity of the test in part load conditions, the laboratory also applied this rule to the compensation method. In this case, the power input has been recalculated from the measured COP.

$$Power\ Input = \frac{P_d \cdot PLR}{COP_{measured}} \quad (1)$$

where:

- P_d is the design capacity, expressed in kW;
- PLR is the part load ratio, expressed in kW/kW;
- COP is the measured coefficient of performance, expressed in kW/kW;

The difference has been evaluated in absolute and relative terms. Eq. (2) presents the deviation where X represents the COP or the power input. The subscript represents the method.

$$\Delta_X = \frac{X_{compensation} - X_{EN14511}}{X_{EN14511}} \quad (2)$$

2.1. Standard test method

The test method is described in the standard EN 14511-3:2022 and EN 14825:2022 [2,3]. The compressor frequency is set in order to meet the part load. For the calculation of exchanged powers, the direct enthalpy method was used. The heating capacity was calculated with Eq. (3):

$$\dot{Q}_H = \dot{V}_w \cdot \rho \cdot (c_{p,out} \cdot \vartheta_{out} - c_{p,in} \cdot \vartheta_{in}) \quad (3)$$

where:

- $\dot{Q}_H$ is the heating capacity, expressed in kW;
- $\dot{V}_w$ is the indoor heat exchanger flow rate, expressed in m³/s;
- ρ is the density, expressed in kg/m³;
- c_p is the specific heat at constant pressure, expressed in kJ/(kg K);
- ϑ is the temperature, expressed in °C.

The thermodynamic properties of water are defined as a function of temperature according to the tables in the "Perry's Chemical Engineering Handbook, 7th Edition, 1997".

The capacity and power input corrections, and the COP (Coefficient of Performance) are calculated according to EN 14511-3:2022.

2.2. Load emulation with the two-mass model

The load is emulated with a two-mass model. Figure 1 shows the schematic of the emulation. The heat pump is installed in the climatic chamber, and a laboratory circuit reproduces the return temperature that comes from the calculation of the emulation model, which is performed by the laboratory control program.

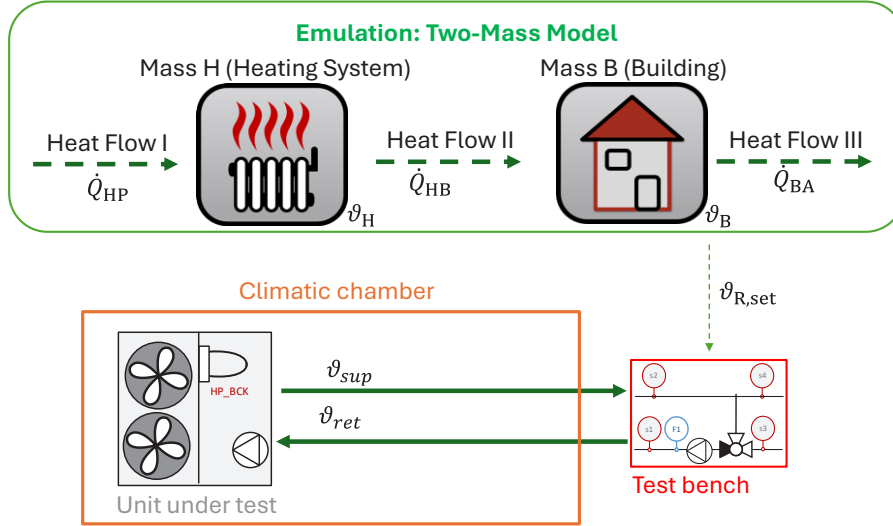


Figure 1 Schematic representation of the two-mass model.

The mass H represents the heat emission system, while the mass B represents the building. The mass H exchanges heat between the heat pump and the building, while the mass B exchanges heat from the mass H and the external environment. The return temperature to the heat pump is calculated from the heat exchange balance between the heat pump and the two masses according to the Eq. (5) to Eq. (9).

$$\dot{Q}_{HP}(t) = \dot{m}_w \cdot c_p \cdot (\vartheta_s(t) - \vartheta_H(t)) \quad (5)$$

$$\dot{Q}_{H-B}(t) = UA_{H-B} \cdot \left(\frac{\vartheta_s(t) + \vartheta_H(t)}{2} - \vartheta_B(t) \right) \quad (6)$$

$$\dot{Q}_{B-A}(t) = UA_{B-A} \cdot (\vartheta_B(t) - \vartheta_A(t)) \quad (7)$$

$$\vartheta_H(t + \Delta t) = \vartheta_H(t) + \left(\dot{Q}_{HP}(t) - \dot{Q}_{HB}(t) \right) \cdot \frac{\Delta t}{C_H} \quad (8)$$

$$\vartheta_B(t + \Delta t) = \vartheta_B(t) + \left(\dot{Q}_{HB}(t) - \dot{Q}_{BA}(t) \right) \cdot \frac{\Delta t}{C_B} \quad (9)$$

where:

- $\dot{Q}$ is the heating capacity, expressed in kW;
- $\dot{m}_w$ is the indoor heat exchanger flow rate, expressed in kg/s;
- c_p is the specific heat at constant pressure, expressed in kJ/(kg K);
- ϑ is the temperature, expressed in °C;
- UA is the heat transfer coefficient, expressed in kW/K;
- t is the time, expressed in s;
- Δt is the acquisition step, expressed in s;
- C is the thermal capacity, expressed in kJ/K;
- The subscripts HP, H, B, and A represent the heat pump, the mass H, the mass B, and the external environment, respectively.

2.3. Case Studies

Table 1 presents the four commercial units considered for the evaluation of the impact of the controller. The authors decided not to report the manufacturers and models of the four units as the aim of the paper is not to compare the performance of specific units but to evaluate the effect on different units. The four units have been selected to cover the different application temperatures requested by market (low temperature 35°C / medium temperature 55°C and fixed/variable flow).

The units considered for this study are four reversible air-to-water monobloc units, two of which have a hydronic indoor unit. The construction year ranges from 2022 to 2024. The application temperatures considered are low-temperature (nominal supply temperature of 35°C) and medium-temperature (nominal supply temperature of 55°C).

Table 1. Units tested.

	Unit 1		Unit 2		Unit 3		Unit 4	
Type	Reversible air-to-water	monobloc	Reversible air-to-water hydronic unit	monobloc with	Reversible air-to-water	monobloc	Reversible air-to-water hydronic unit	monobloc with
Construction year	2022		2024		N.D.A.		2024	
Capacity (A7/W35)	9.9 kW		7.9 kW		Not tested		4.3 kW	
COP (A7/W35)	4.6 kW/kW		4.4 kW/kW		Not tested		4.3 kW/kW	
Capacity (A7/W55)	10.1 kW		6.7 kW		About 10 kW		Not tested	
COP (A7/W55)	3.2 kW/kW		2.7 kW/kW		N.D.A.		Not tested	
Refrigerant	R32		R32		N.D.A.		R290	
Refrigerant charge	1.6 kg		1.15 kg		N.D.A.		0.95 kg	
Test configuration EN14825:2022	Table 8 and 10, fixed flow, variable outlet		Table 8 and 10, variable flow, variable outlet		Table 10, fixed flow, variable outlet		Table 8, fixed flow, variable outlet	
	EN 14825:2022 - Table 8 - Part load conditions for air-to-water(brine) units in low temperature application for the reference heating seasons							
	EN 14825:2022 - Table 10 - Part load conditions for air-to-water(brine) units in medium temperature application for the reference heating seasons							
Note: Unit 3 is covered by an NDA agreement which foresees to publish the relative differences between two methods. For this reason, all the parameters of unit 3 cannot be presented.								

2.4. Laboratory

The tests were carried out at the Heat Pumps Lab of Eurac [23]. The unit has been connected to the laboratory as indicated in Figure 2. The return and supply temperature sensors, the differential pressure sensor are installed close to the unit according to the indication of EN 14511-3:2022. The laboratory has a conditioning storage (2000 L) that is decoupled from the tested unit with a heat exchanger. A 100 L buffer has been added in the climatic chamber to smooth the supply temperature during the defrost cycles. The buffer is located in the climatic chamber only for practical motivation of installation.

The control of the return temperature is done with a three-way valve, while the control of the flow rate can be done with a two-way valve or a pump (depending on the presence of the internal circulator). In the standard test the three-way valve controls the return temperature with a constant set point while in the compensation method the set point is calculated with the emulation model that takes as input the measured flow and the supply temperature.

The laboratory guarantees measurement uncertainties compliant to EN14511-3:2022 requirements.

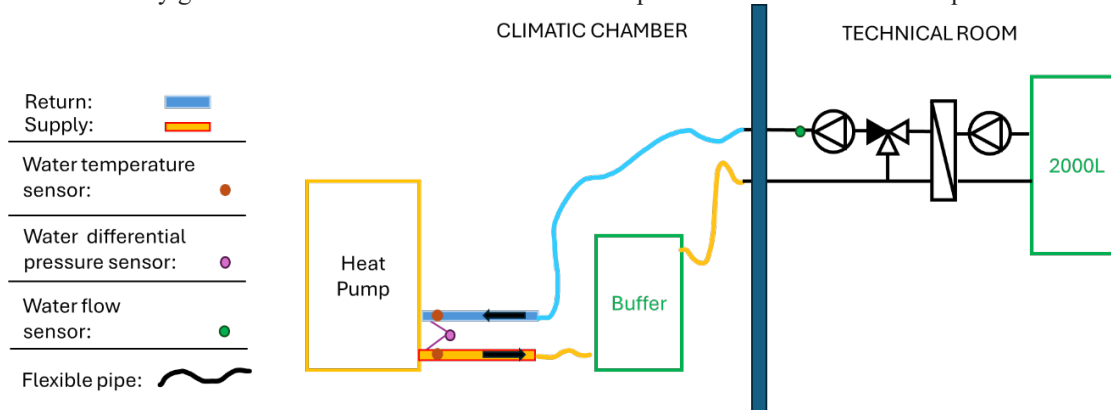


Figure 2 Connection of the heat pump to the test bench.

3. Results and Discussion

In this section, the presentation of results is done for each unit in the same way: the Tables present the comparison of the COP and power input, and the Figures plots the return and supply temperature, the heating capacity and the power input. The air temperatures (wet bulb and dry bulb) are not reported to simplify the presentation of results. As the air temperature is fully controlled by the laboratory there are no differences between the two methods. The air temperature has been controlled according to the requirements of EN 14511-3:2022.

Unit 1 has been tested according to the low and medium temperature applications with fixed flow and variable outlet. Table 2 presents the COP and power input measured applying the EN14825:2022 and the compensation method. Points B and C at low temperature present a high difference between the standard test and the compensation method. During those two tests, the lamination valves were instable. In contrast, the point B at medium temperature presents a difference of only 3% on the COP. In the test points B, the Unit 1 operated without performing defrost cycles.

Point D at medium temperature presents a difference of 9.2% in the COP and 10.2% in the power input. This difference can be explained by looking at the boundary conditions which are reported in Figure 3.

Table 2. Unit 1. Comparison of COP and power input.

Point	Part load	COP			Power input		
		EN14825	Compensation	Difference	EN14825	Compensation	Difference
B _{LT} (2/30)	5.05	4.18	2.98	-28.7%	1.21	1.70	40.3%
C _{LT} (7/27)	3.25	5.83	4.39	-24.7%	0.56	0.74	32.9%
B _{MT} (2/42)	6.90	3.27	3.17	-3.1%	2.11	2.18	3.2%
C _{MT} (7/36)	2.70	4.63	3.94	-15%	0.58	0.69	17.6%
D _{MT} (12/30)	1.20	6.38	5.79	-9.2%	0.19	0.21	10.2%

Regarding point D at 12/30 °C, the standard considers the minimum modulation of the unit and applies a degradation coefficient for the supposed on/off cycles depending on the capacity ratio (defined in the standard as the ratio between the part load and the declared capacity). In the standard test, the unit is operated at minimum frequency and maintained at a constant temperature. In the compensation method the unit reaches the set point and switches off. The model emulates the inertia of the heat emission system, and the return temperature is emulated and therefore controlled also when the unit is off. The controller of the unit activates the heat pump according to its control strategy. Also, in this case, the unit reaches higher peaks of supply temperature and capacity to provide on average the required heating capacity.

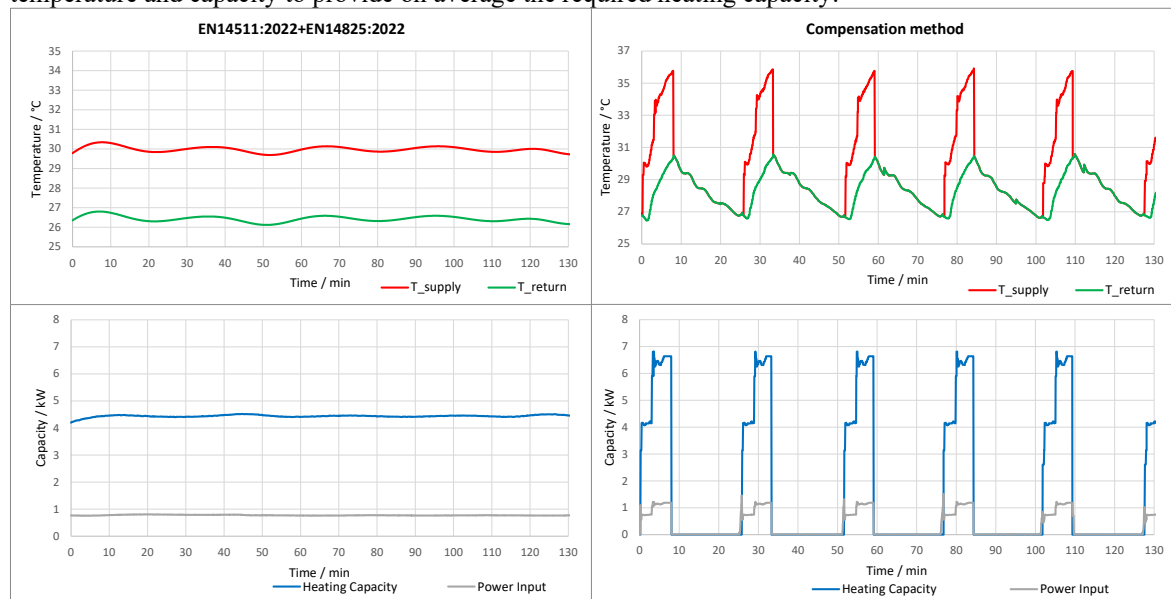


Figure 3 Unit 1, point D medium temperature. Comparison of supply and return temperatures, capacity and power input.

Unit 2 has been tested according to the low and medium temperature applications with variable flow and variable outlet. Table 3 presents the COP and power input measured applying the EN14825:2022 and the compensation method measured for the low temperature application. The table presents two averaging intervals of the compensation method considering the different adjustments of the heating curve of point B and one averaging interval obtained after stability reached during point C. The two averaging intervals of point B has been calculated as the duration of defrost cycles are different. Point B presents a difference between 7.9% and 10.9% in the COP and between 8.5% and 12.3% in the power input while point C presents a difference of almost 0% in the COP and power input. These differences can be analyzed in more detail looking at the boundary conditions which are reported in Figure 4 for point B and in Figure 5 for point C.

Table 3. Unit 2. Comparison of COP and power input.

Point	Part load	COP			Power input		
		EN14825	Compensation	Difference	EN14825	Compensation	Difference
B _{LT} ' (A2/W30)	4.31	4.28	3.94	-7.9%	1.01	1.09	8.5%
B _{LT} '' (A2/W30)	4.31	4.28	3.81	-10.9%	1.01	1.13	12.3%
C _{LT} (A7/W27)	2.77	6.4	6.42	0.3%	0.43	0.43	-0.3%

The standard test of point B presented longer duration of the heating period while in the compensation method the heating period where shorter (meaning more frequent defrost cycle). Also in this test, the unit tested with the compensation method reached the highest temperature and capacity peak. In the first cycle of the compensation test, the supply temperature and power input (and consequently the heating capacity) oscillated with small variations. This is due to the combination of the variation of modulation of the compressor and the electronic expansion valve.

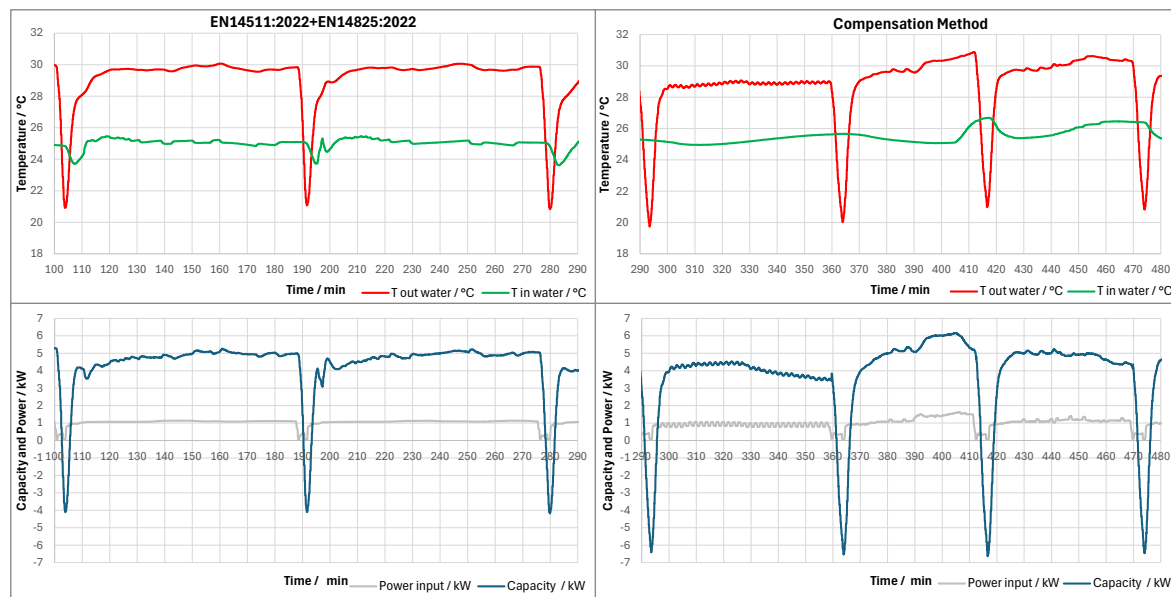


Figure 4 Unit 2, point B low temperature. Comparison of supply and return temperatures, capacity and power input.

The part load of point C is 2.77 kW, and the laboratory performed the standard test with a capacity of 2.92 kW and 2.7 kW in the compensation method. Both points are within the 10% of tolerance. The flow rate obtained from the standard test has also been applied within the compensation method. Figure 5 presents the raw measurements; in other terms, the power input presented in the figure has not been processed using Eq. (1).

In the compensation method, the transition to reach stability required a long period. At the start the unit controlled the compressor with a higher frequency than needed and the supply temperature increased over the set point. After one hour the compressor frequency decreased and the unit maintained the frequency to lower values with continuous oscillations. After 300 minutes the unit reached a similar stability to the standard test. In this case the two COP values are the same.

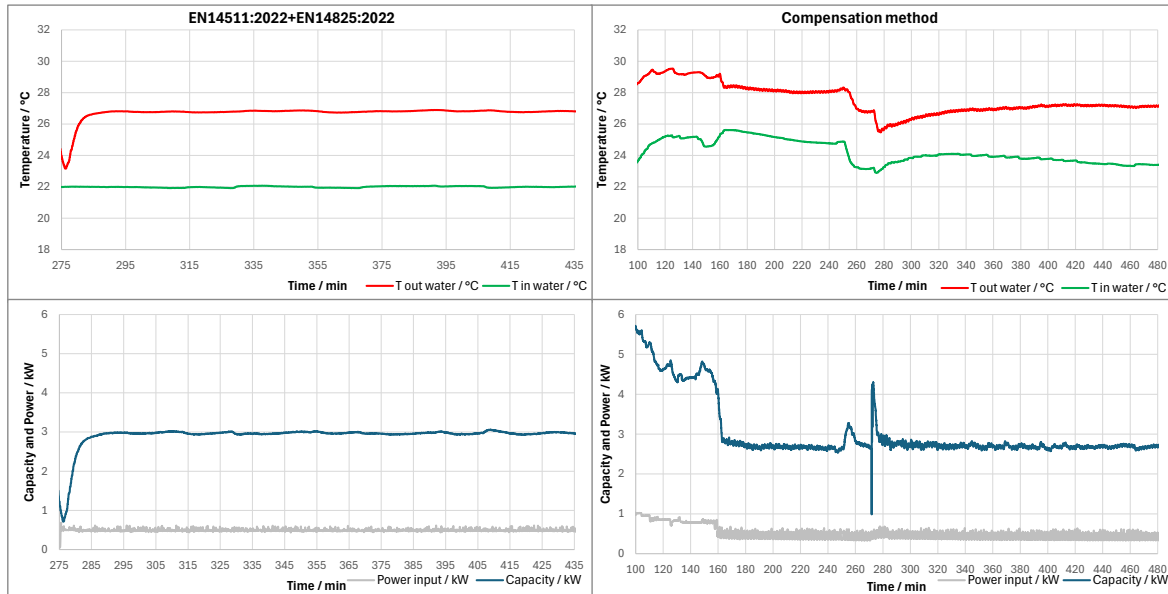


Figure 5 Unit 2, point C low temperature. Comparison of supply and return temperatures, capacity and power input.

Unit 3 confirm the difference on the defrost control. Point A (-7/52 °C) presents a difference of 10% in the COP and 11.2% in the power input. The standard test maintains constant compressor frequency and the temperatures except for the defrost period. The compensation method instead controls the return temperature according to the load emulated and the units follows the climatic curve. Even if the average supply temperature and the heating capacity are equal to the standard test, in the compensation method the unit reaches higher peaks of supply temperature and higher peak of capacity. This is reflected in the duration of the heating time where the unit performs a defrost cycle with higher frequency. The consequence is a higher power input and lower COP.

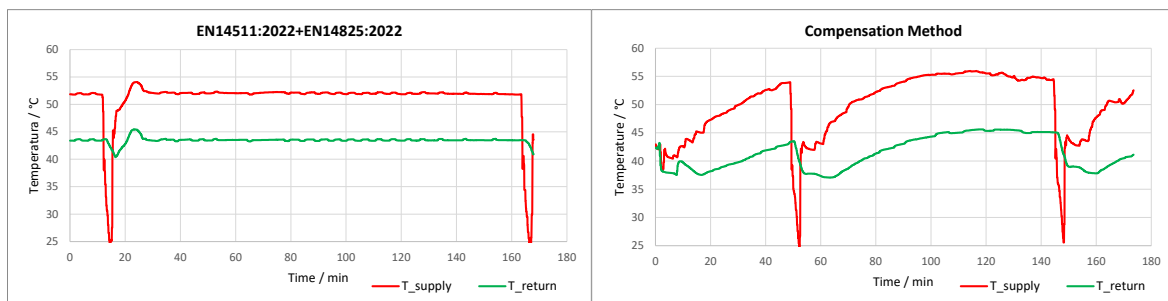


Figure 6 Unit 3, point A. Comparison of supply and return temperatures.

Figure 7 presents the range of differences between the standard test and the compensation method for the four tested units. Unit 1 presents the highest differences, reaching the highest deviation with a -28.7%. In this case, during the test with the compensation method in some test points, the unit presented an instable control of the electronic expansion valves that has degraded the COP in those points. Unit 2 presents the smallest differences with a range between 0.3% (within the measurement uncertainty) and -10.9%. Similar results have been obtained with Unit 4, which presents a range of deviations between the standard test and the compensation method between 0.3% and -14%. The deviations of COP for the Unit 3 are intermediate compared to the other units, ranging from -3.4% to -17.6%.

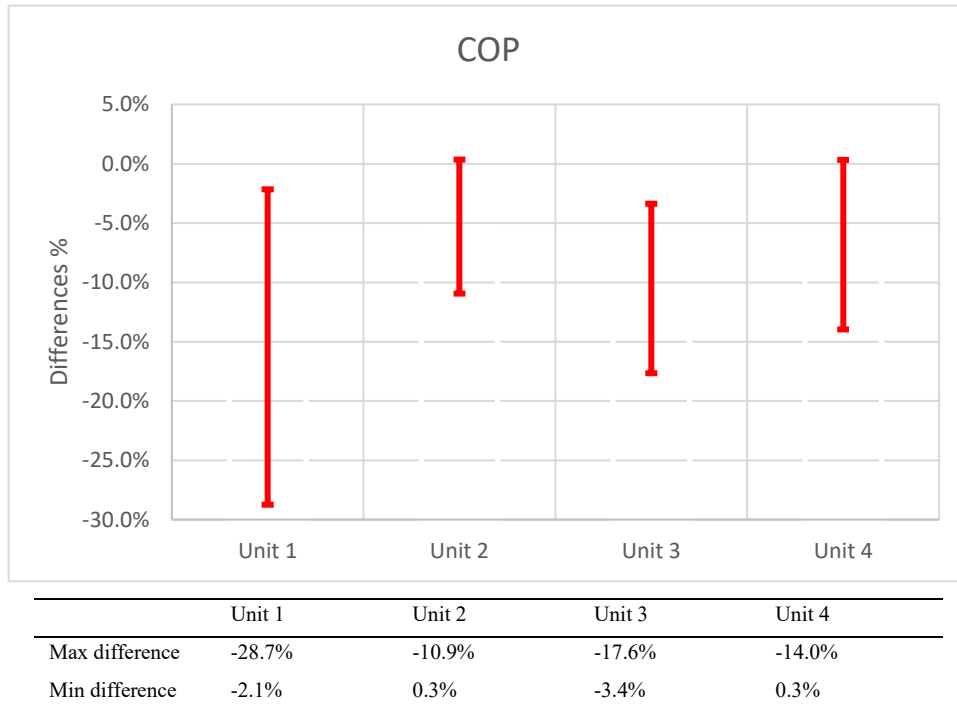


Figure 7 All units. Range of differences in the evaluation of COP.

4. Conclusions

The paper presented the impact of the controller on the performance measured in laboratory by comparing the results evaluated with standards EN14825:2022+EN14511:2022 and the two-mass model of the compensation method. The evaluation has been done by comparing COP and power input for the same part load points of four different air-to-water heat pumps. The part load has been defined considering the design capacity declared by the manufacturer for the average climate according to Table 8 (low-temperature application) and Table 10 (medium-temperature application) of EN 14825:2022.

With the standard test the unit controller is somehow bypassed since the laboratory controls or limits the compressor frequency while in the compensation method the laboratory emulates a load, and the unit controller modulates the capacity according to its strategies.

The comparison reveals that the influence of the controller is different between the different tested units and the test points. Those differences are present in the control of the defrost and the control of the capacity in part load conditions. The test performed with the compensation method presented more frequent defrost (heating time shorter than the standard test) with the presence of higher peaks of supply temperature. In the tests where there was required a part load lower than the minimum modulation of the unit, the compensation method measured the performance under on/off cycling. Also in this case, the compensation method reached highest peaks of supply temperature.

Unit 2 presented differences in COP and power input ranging from 0% to 10%. The difference of 0% has been observed in Point C after a long stabilization time. Instead, Unit 1 presented a difference up to 28.7% due to an instability of the control of the lamination valves. Considering this as an exceptional case, the variation of the differences can be considered between 0% and 15%.

From the point of view of the test bench, to implement the compensation method, the laboratory has implemented the emulation model (two-mass model) without the need of changes in the setup of the test bench or modifications of the structure of controller of the laboratory. The laboratory had only to tune the PID controller of the water loop since the unit was operating with on/off cycles.

The results of the paper demonstrate the importance of including the unit controller in the test boundary for the development of the heat pump.

Nomenclature

P_d	Design capacity, expressed in kW
PLR	Part load ratio, expressed in kW/kW
COP	Coefficient of performance, expressed in kW/kW
$\dot{Q}$	Heating capacity, expressed in kW
$\dot{V}_w$	Indoor heat exchanger flow rate, expressed in m ³ /s
$\dot{m}_w$	Indoor heat exchanger flow rate, expressed in kg/s
ρ	density, expressed in kg/m ³
c_p	specific heat at constant pressure, expressed in kJ/(kg K)
ϑ	temperature, expressed in °C
UA	heat transfer coefficient, expressed in kW/K
C	Thermal capacity, expressed in kJ/K
HP	Heat pump
H	Mass H (heating system)
B	Mass B (building)
A	External environment

Acknowledgements

The laboratory activity performed at Eurac was financed by ERDF 2021-2027 1034 SustainHEAT CUP D53C23003060009 of the program of the Autonomous province of Bolzano-Alto Adige.

The activity performed by BAM was financed by Deutsche Bundesstiftung Umwelt.

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