

Experimental Investigation of a Condensing Gas Absorption Heat Pump for Residential Buildings

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ABSTRACT

Innovative technologies are always matter of study. In this work, an experimental performance analysis of an ammonia-water condensing gas absorption heat pump was conducted. This heat pump has a nominal heating capacity of 18,9 kW and it is designed for domestic hot water and space heating. The tests were carried out in a climatic chamber in order to emulate different outdoor ambient conditions regarding temperature and humidity. Different tests conditions were imposed based on a combination of the EN 12309 requirements and typical weather data for Belgium. Measures of gas consumption, electrical consumption, mass flow rates and temperatures were collected to compute performance indicators. The experimental results showed a COP that varies between 0.86 and 1.45 for different delivery water temperatures and ambient conditions. Also, it was observed that the humidity could have a positive or negative impact on the COP.

1. INTRODUCTION

The use of energy is a relevant topic nowadays. More and more studies and information are available on the origin, use and impact of our energy consumption on the environment and the building sector has been pointed as one of the key areas in the matter. In 2017, the household sector represented 30% of the final use of energy consumption in the European Union (EU) with a total of 288 million tons of oil equivalent (Mtoe), only being surpassed by the industrial sector (European Environment Agency, 2020). Furthermore, 80% of the energy used in the residential sector was destined for domestic hot water (DHW) and space heating.

Hence, more efficient ways to heat water both for DHW and space heating are sought, encompassing new and better technologies that offer an acceptable payback value and that are compatible with residential applications. It is here and in the frame of this work where heat pumps represent an interesting focus of study since it is a market that is undoubtedly growing. Between 2005 and 2018, the use of heat pumps in the heating sector in the EU represented an increase in energy consumption of 8.3 Mtoe, surpassed only by the use of solid biomass (23.7 Mtoe) (European Environment Agency, 2020). Related to this and according to 2016 data, the energy consumption in dwellings in Belgium destined just to space heating and water heating represented 1.52 Mtoe (European Environment Agency, 2016). Additionally, if we observe the final energy consumption by area in 2010, the household's presence is strongly reflected in the consumption of natural gas, representing 45.6% of the total energy consumption; this is followed by electricity consumption (29.7%), solid fuel consumption (22.4%) and petroleum products (9.4%) (European Environment Agency, 2013).

This is where the study of gas absorption heap pumps (GAHP) appears as an attractive alternative to traditional appliances. Keinath and Garimella (2017) showed that a GAHP offers substantial cost and energy savings compared to conventional commercially available systems for water heating thanks to the competitive price of gas and to the involved technologies. (Famiglietti, et al., 2021) compared the environmental profiles of a condensing boiler and a gas driven absorption heat pump as competing technologies to provide space heating and domestic hot water in Europe, showing that the use phase contributes for more than 97% of the total environmental impact over the entire

life cycle of the appliance. On the other hand, (Fumagalli, et al., 2017) highlight the importance of the correct integration, sizing, and control of the system so as not to negatively affect the COP.

Several studies have been conducted to compare different gas driven heat pumps considering their components, the complexity of the systems and the related parameters such as delivery water temperature and gas mass flow rate input, arriving to complex and detailed models (Babak Dehghan, et al., 2020) (Wu, et al., 2020), some of them experimentally calibrated (Aprile, et al., 2016) (Aprile, et al., 2017). However, the role of outdoor conditions and the reaction of the system to them rest still a matter of study. A lot of research has been done in this area for electric heat pumps as is pointed by (Tassou & Marquand, 1987) (Kaygusuz, 1994) (Vocale, et al., 2014) (Chen & Guo, 2009), unlike the area of the GAHP that remains to be a subject of study.

In this work, an experimental analysis of a gas driven absorption heat pump was conducted. Different outdoor ambient conditions regarding temperature and humidity were emulated to characterize its behavior depending on weather conditions and to estimate performance indicators.

2. DESCRIPTION OF THE SYSTEM

2.1. Gas absorption heat pump

The tested gas absorption heat pump is designed for space heating and domestic hot water (DHW) for residential applications, with a nominal heating capacity of 18.9 kW. The system is based on the Water-Ammonia absorption cycle using outdoor ambient air as renewable energy source (low-temperature heat source) and natural gas combustion as high-temperature heat source; the delivered hot water is the medium-temperature heat sink.

Figure 1 shows a schematic diagram of the system. Here the Burner (BRN) driven by natural gas is used to heat the absorbent-refrigerant solution in the Generator (GEN) causing the separation of the two components by desorption. The desorbed ammonia vapor leaves the GEN and passes through the Rectifier (REC) to remove the last parts of residual water. Then it continues to the Condenser (COND), transferring the heat of the refrigerant to the water destined to the Heating Circuit (HC) e.g., radiators, floor heating or others, previously Pre-Heated (PH) by the combustion gases and impulsed by the Water Circulation Pump (WP). The refrigerant leaving the COND is throttled to reduce its pressure by means of a restrictor valve and cooled down inside the Pipe in Pipe heat exchanger (PiPHx); then, by means of a second restrictor valve, is brought to the ideal pressure and temperature conditions to enter the Evaporator (EVAP). The liquid refrigerant is directed to the EVAP where it is evaporated taking heat from the surrounding air. Then, the low-pressure vapor ammonia is overheated in the PiPHx before arriving to the Solution Cooled Absorber (SCA) where it meets the poor refrigerant solution coming from the GEN. The pressure of the incoming solution is reduced by a restrictor valve.

To complete the absorption process and since it is an exothermic reaction, the solution is sent to the Water Cooled Absorber (WCA) where a considerable amount of thermal energy is transferred to the thermal fluid of the heating circuit. Once this process is completed, the solution is pumped back to the GEN using a Solution Pump (SP).

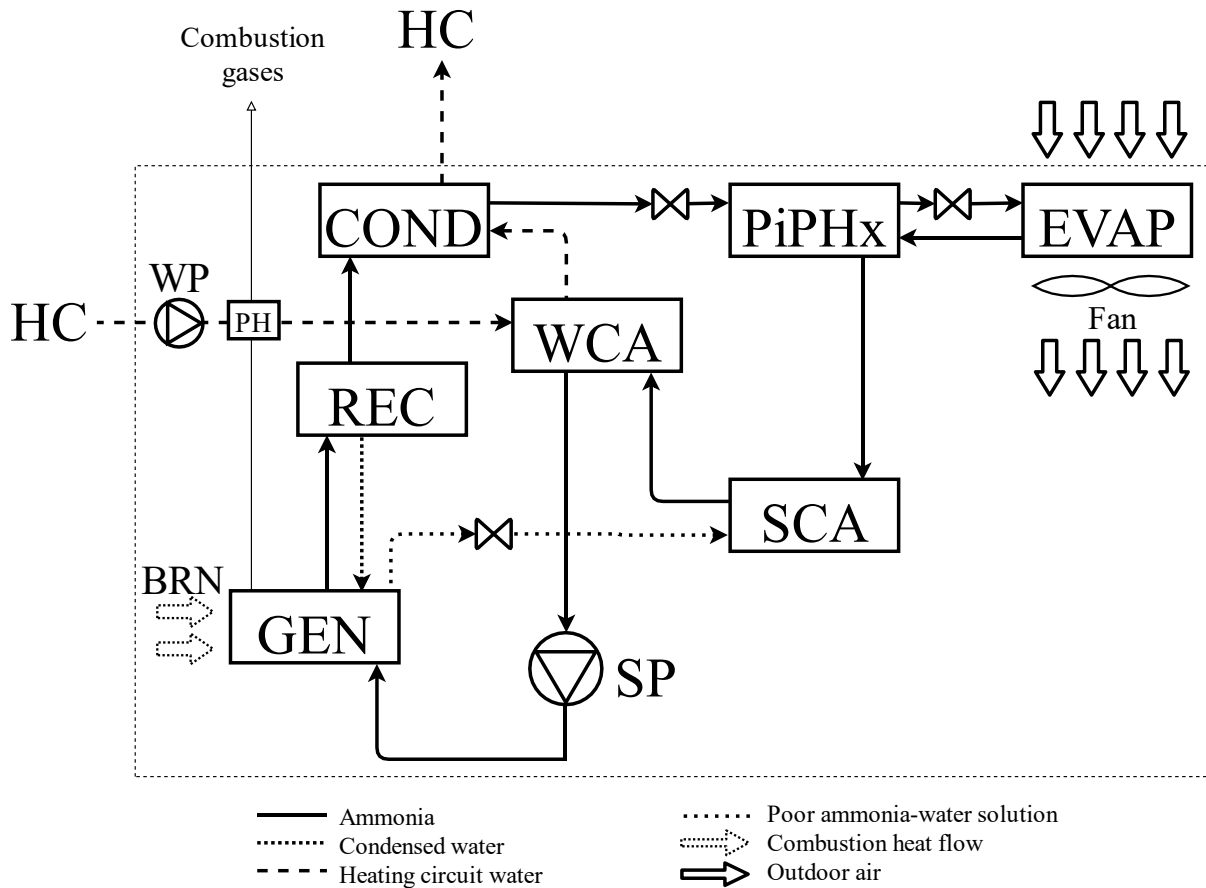


Figure 1: Schematic of the gas absorption heat pump components

2.2. Test bench

Since the system is designed to be installed outdoor, tests are performed in a climatic chamber to control temperature and humidity conditions. The distribution of the test bench components inside the climatic chamber is shown in Figure 2.

The test bench was designed in order to be supplied by electricity, natural gas, and chilled water. The electricity consumption is measured as well as natural gas and water mass flow rates, while the latest is controlled as well. The products derived from the operation of the system such as combustion gases, condensate and hot water are removed from the test bench.

The room temperature is controlled by means of an outdoor air-to-water heat pump unit located inside the chamber. On the other hand, the humidity of the room is reduced by water condensation in the evaporators of both units. Once the desired test temperature and humidity conditions are reached, the stationary conditions are maintained by means of an electrical heater and a humidifier. These latter are connected to an acquisition system and controlled by a PI controller which receives the signal of temperature and humidity sensors placed inside the room, more precisely at the entrance of the evaporator. To emulate the heat demand of a home, a heat exchanger is installed in the room adjacent to the climate chamber. The load is regulated by regulating the chilled-water flow rate through the exchanger.

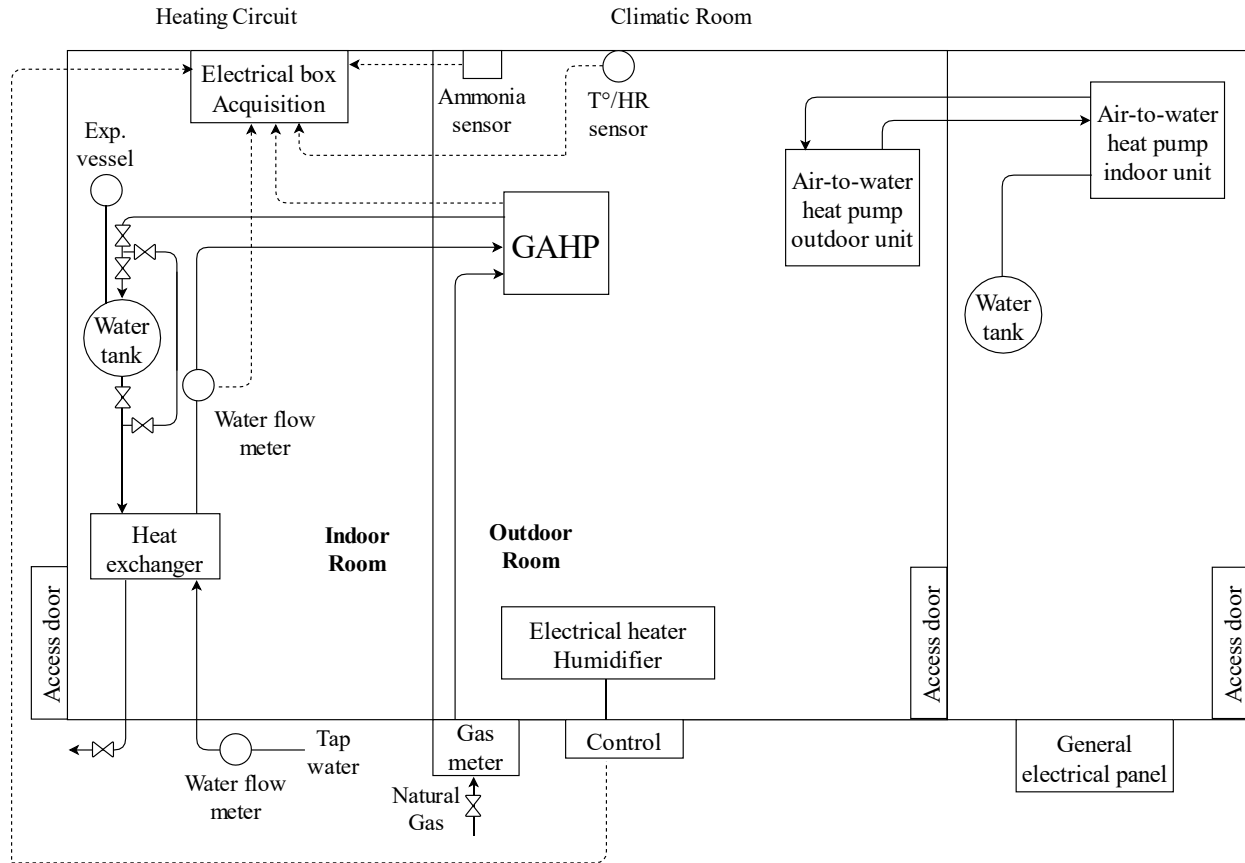


Figure 2: Schematic of the gas absorption heat pump test bench

Table 1 summarizes the measuring devices and their characteristics. Inside the heat pump only surface thermocouples were installed on the different pipes between the components. These thermocouples were placed on an electro-insulating and thermo-conductive sheet, fixed with plastic clamps at each measurement point to ensure thermal contact and subsequently insulated to measure fluid temperature. On the rest of the test bench, in-pipe thermocouples were used. In the cases where the temperature of a large cross-section had to be measured, e.g., fan exhaust and evaporator supply air temperatures, a grill of thermocouples was used; more precisely, 4 equidistant thermocouples were installed at the fan exhaust and 9 at the evaporator supply. Measurements of gas consumption, water consumption, electrical consumption, ambient temperature, and humidity are also collected.

The unit works with an ammonia-water solution contained on the internal circuit and that is harmful for health. Thus, and for caution, a release pipe was installed to extract the ammonia out of the room if necessary and an ammonia sensor was installed close to the unit to detect leaks.

Table 1: Measuring devices

Sensor	Type	Accuracy	Number of measure points
Thermocouples	T	± 0.3 K	45
Humidity	Capacitive - wetttable	± 2 %	2
Water meter	Volumetric	± 2 % Q_n ; ± 5 % Q_{min}	1
	Magnetic	± 0.5 % from 0.3 to 11.89 m/s	1
Gas meter	Diaphragm	± 0.5 %	1
Power meter	Multifunctional	± 0.5 %	1
Ammonia sensor	Electrochemical	± 5 ppm	1

2.3. System modifiable settings

The gas absorption heat pump has a display board which gives access to different menus that allow to configure certain operating parameters of the system. These parameters are supplied by the manufacturer and some of these are modified in order to facilitate the characterization of the system.

It is sought to maintain a constant water temperature difference between the delivery and the inlet of the appliance. To do so, the modulation of the circulation pump is activated and the water delta T° is set to 10 K.

On the other hand, to not restrict the system's behavior it is necessary to set the permissible water temperature range for the delivery and return of the system. In other words, a wide enough water temperature range allows to not operate, for example, at partial load. Therefore, the delivery water temperature range is set between 30°C and 75°C, while the return range goes between 20°C and 30°C.

The delivery water temperature can be controlled by a variable water setpoint which depends on the outdoor temperature (weather compensated control) or by means of a fixed setpoint. The latter method is used, coinciding with the maximum delivery water temperature set (75°C) with the option of been modified if necessary.

Finally, it is possible to establish a power control of the system based on a "On/Off" behavior or allow the modulation of the burner. This second option is chosen, giving rise to a modulation of the gas flow on the burner side, while in the absorption cycle it is reflected in variations in the fan voltage and in the water circulation pump.

The described configuration is shown in Table 2.

Table 2: Gas absorption heat pump adjustable parameters

Description	Setting
Modulation of circulation pump	Active
Heating circuit water ΔT° setpoint	10 K
Power modulation	Active
Delivery water temperature range	From 30 K to 75 K
Delivery water temperature setpoint	From 45K to 75 K
Return water temperature range	From 20 K to 30 K

3. TESTING CONDITIONS

In order to achieve the characterization of the system performance, the test campaign is based on the EN 12309 (European Standards, 2014) and on a weather data analysis for Belgium (WeatherUnderground, 2019).

The requirements of the European standard for test conditions at full load refer to the type of appliance (e.g., air-to-water, water-to-water), its application (e.g., low/medium/high temperature), the outdoor heat exchanger conditions referring to dry-wet bulb temperatures and the classification of the climate (e.g., medium, warm, or cold).

To consider the weather conditions to which the appliances are subjected in the field in terms of temperature and humidity, two local weather stations close to the systems were selected and a weather data analysis was made for the cold season from October 2018 to March 2019.

Based on these two aspects, the performed test matrix is shown in Table 3. The outdoor air-dry bulb temperature meets the requirements of the European standard and all types of application are tested, therefore, four water delivery temperatures. This matrix is performed for a relative humidity of 75% since it is the most frequent value obtained from the weather data analysis; additional tests are performed at relative humidities of 60% and 85% to evaluate their impact on the system performance for temperatures ranging from 2°C to 7°C, where frost formation is more likely to occur. Every test is performed at full load on steady state for a period of 20 minutes. The test conditions are monitored throughout the test with a smartphone connected to the appliance besides the test bench data acquisition system.

Table 3: Gas absorption heat pump test matrix

		Water delivery Temperature [°C]			
		35	45	55	65
Outdoor dry bulb Temp.[°C]	12	75%			
	7	60%, 75%, 85%			
	2	60%, 75%, 85%			
	-7	75%			
	-10	75%			

4. RESULTS

The European standard defines a valid data collection period based on the coefficient of change defined in Equation (1). This coefficient is the difference between the outlet and the inlet temperatures of the heat transfer medium at the indoor room heat exchanger and should be calculated every 5 minutes starting at the end of the previous period ($\tau = 0$). If this coefficient remains within 2.5% during the data collection period, then the test can be considered as steady state.

$$\Delta T\% = \frac{\Delta T_{(\tau=0)} - \Delta T_{(\tau)}}{\Delta T_{(\tau=0)}} \quad (1)$$

In addition, permissible deviation values from the set values are set. This corresponds to $\pm 0.3\text{K}$ for room temperature, $\pm 2\%$ for room mean humidity and $\pm 1\text{K}$ from the setpoint for the depart water temperature.

With the previous conditions mentioned, the COP for each test was estimated as the ratio between the thermal power output to heat and electric power input. The thermal power output corresponds to that given to the heating circuit while the inputs are the thermal heat obtained from the natural gas combustion and the electric input to the appliance as defined in Equation (2). The thermal power transferred to the water is the one obtained from the water cooled absorber, the condenser, and the combustion gases as defined in Equation (3).

$$COP = \frac{\dot{Q}_{HC}}{\dot{Q}_{gas} + \dot{W}_{in}} \quad (2)$$

$$\dot{Q}_{HC} = \dot{Q}_{WCA} + \dot{Q}_{cond} + \dot{Q}_{gases} \quad (3)$$

Since the internal configuration of the system makes it difficult to install sensors between components that allows to measure the previously defined heat inputs individually, it is decided to estimate the heat input transferred to the water as defined in Equation (4). Similarly, the heat input is defined in Equation (5) as the result of the gas combustion considering a daily average high calorific value. The electric consumption of the appliance \dot{W}_{in} is constantly registered and considered on the results, with maximum variations of 2% between tests and close to 0.35 kW. This consumption includes components such as the fan, oil pump, water circulation pump and sensors.

$$\dot{Q}_{HC} = \dot{m}_{HC,w} * c_{p,w} * (T_{out} - T_{in}) \quad (4)$$

$$\dot{Q}_{gas} = \dot{V}_{gas} * PCS \quad (5)$$

The results obtained for the test matrix carried out at a relative humidity of 75% are shown in Table 4. The results are computed based on the average values of the measurements carried out during a 20-minute test, meeting the requirements of Equation (1).

Table 4: COP and Thermal Capacity values at a relative humidity of 75%

COP 75%		T° delivery			
		35	45	55	65
Outdoor T°	12	1.45	1.34	1.19	1.05
	7	1.38	1.29	1.13	1.04
	2	1.36	1.21	1.09	0.95
	-7	1.21	1.13	1.02	0.86
	-10	1.16	1.14	0.95	0.86

Th. Capacity [kW] 75%		T° delivery			
		35	45	55	65
Outdoor T°	12	21.11	19.31	16.98	14.82
	7	19.55	18.57	16.30	14.92
	2	20.10	18.01	15.64	13.70
	-7	18.33	16.89	15.13	12.65
	-10	17.36	16.82	14.14	12.66

It can be observed that at a relative humidity of 75%, the delivery water temperature has a higher impact on the COP than the outdoor room temperature. These variations are reflected in an increase in the COP between 0.3 and 0.41 for a delivery temperature between 35°C and 65°C; for outdoor room temperature between -10°C and 12°C, the COP increases between 0.19 and 0.29 for the presented test's conditions. The latter is reflected in a decrease in the thermal capacity of the system as the outdoor temperature decreases, having as a consequence a decrease in the water flow of the heating circuit.

The results obtained at different relative humidities are shown in Table 5.

Table 5: COP and Thermal Capacity values at 60% and 85% room humidity

COP 60%		T° delivery			
		35	45	55	65
Outdoor T°	7	1.34	1.34	1.15	0.98
	2	1.30	1.28	1.10	0.94

Th. Capacity [kW] 60%		T° delivery			
		35	45	55	65
Outdoor T°	7	19.68	19.40	16.73	14.20
	2	19.09	18.68	16.08	13.56

COP 85%		T° delivery			
		35	45	55	65
Outdoor T°	7	1.36	1.30	1.17	1.02
	2	1.36	1.21	1.09	0.96

Th. Capacity [kW] 85%		T° delivery			
		35	45	55	65
Outdoor T°	7	19.69	18.87	17.02	14.54
	2	20.01	17.53	15.92	13.79

For outdoor room temperatures ranging from 2°C to 7°C and relative humidity of 75%, the COP varies between 0.34 and 0.41 for a delivery water temperature between 35°C and 65°C; for the same outdoor room temperature range and delivery water temperatures, it has been found that the COP varies by 0.36 for room humidity of 60% and between 0.34 and 0.40 for 85% room humidity.

Table 6 presents the results for different relative humidities for temperatures of 2°C and 7°C and compared to the results obtained for a relative humidity of 75% as baseline. The results shows that for the minimum and maximum outlet water temperatures tested the COP is negatively affected by the changes in room humidity except for the W65/A2/HR85 case; contrariwise, the COP is positively affected for the warm temperatures of 35°C and 45°C.

Table 6: Variations of COP values with respect to room humidity

COP		T° delivery											
		35			45			55			65		
		75%	60%	85%	75%	60%	85%	75%	60%	85%	75%	60%	85%
Outdoor T°	7	1.38	-0.04	-0.02	1.29	+0.05	+0.01	1.13	+0.02	+0.04	1.04	-0.06	-0.02
	2	1.36	-0.06	-	1.21	+0.07	-	1.09	+0.01	-	0.95	-0.01	+0.01

It can be observed that for the cases with the same humidity and outlet water temperature, the COP increases as the outdoor temperature does. This can be explained by the fact that there is a bigger amount of latent heat available on the outdoor air and therefore in the evaporator, resulting on a better COP.

On the other hand, for the cases with the same outdoor and delivery water temperature, a higher humidity has less impact on the COP than a lower humidity. This is due to the fact that in a short period of time, the higher the humidity the lower the density of the frost layer and therefore the accumulated mass. The way the frost layer is built up affects the overall heat transfer coefficient and increases also the air pressure drops on the evaporator. At lower humidity, the frost layer is more dense and has a greater insulating effect, affecting the heat rate transfert on the evaporator and reducing the COP.

5. CONCLUSIONS AND PERSPECTIVES

An experimental investigation of a gas absorption heat pump has been conducted were coefficients of performance depending on the delivery water temperature, air temperature and humidity have been calculated. A description of both the system and the test bench is made, including the sensors used and the internal system settings.

During the development of this work, it was possible to verify that the measurements of the installed sensors on the test bench were consistent with the ones registered by the system software. In addition, it was possible to observe part of the system's internal control and the variations of the system components during, for example, a defrost cycle. It is also possible to ensure that the system worked at full load during every test performed.

Interesting results were found for the tests performed. It was observed that the biggest effect on the COP is caused by the delivery water temperature, the latter being affected by the thermal power output of the water cooled absorber, condenser, flue gases, and circulation water pump behavior. On the other hand, the outdoor room temperature and humidity have less effect on the COP, being the evaporator the main affected component.

Another point to consider is the frost accumulation across time. The defrost cycles are designed to maintain the performance of the system. However, the time between these cycles involves the formation of a frost layer that could degrade the performance of the appliance in a non-neglected way before arriving to the defrost point, making the control of the system of vital importance to ensure good performance. Nevertheless, considering the defrost cycles in the performance indicators such as the seasonal performance factor will lead to a degradation of the latter.

It would be interesting to perform test at higher temperatures and humidities without arriving to frost formation since the moist effect on the evaporator heat transfer coefficient could improve the COP due to the amount of latent heat

available. Also, the study of the impact of the different adjustable settings on the performance of the appliance remains to be investigated as well as the evaluation of the environmental performance of the appliance.

Although it was not the objective of this work, observations were made regarding the defrost cycles between tests. The components and parameters involved in the control and realization of a defrost cycle have a high dependence between them making hard to predict the conditions that trigger it. It was constantly observed that for the same outdoor conditions defrosting cycles were carried out or not, remaining within the perspectives of this work to delve more into the conditions that favor the startup of a defrosting cycle.

NOMENCLATURE

\dot{Q}	Heat transfer rate	(W)
T	Temperature	(°C)

Subscript

abs	Absorber
cond	Condenser
gas	Gas
gases	Gases
gen	Generator
HC	Heating circuit
in	Input
out	Outlet
w	Water
wca	Water cooled absorber

Greek symbols

Δ	Difference
τ	Time

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