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## **New Methodology of Characterization of Seasonal Performance Factor of An Air-To-Water Heat Pump**

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### **ABSTRACT**

The threat of the global warming is alarming. The increase in oil prices remains a persistent worry and today's energy challenges are truly critical. In this context, heat pumps in the residential sector appear to be an interesting solution to reduce energy consumption and emissions of greenhouse gases.

The French government has developed, under the leadership of the "Grenelle of the environment", two successive tax incentives that have increased the number of heat pumps (HP) sold in France for heating and / or production of domestic hot water.

European standards used to characterize the air-to-water heat pump performances are based on one test at maximum load operation for an outdoor temperature of 7°C and an outlet temperature of hot water of 35°C. These operating conditions are very different from real life conditions. They can even be quite unrealistic as they exclude start-up and shutdown, defrosting of the outdoor unit, the part load impact, and the weather and water temperature lifts.

In this paper, an experimental study is performed on three air-to-water heat pumps. Tests are realized for outdoor temperature varying between -10°C and 14°C and for three water temperature levels. These tests allowed calculating the average heating seasonal performance factor of the HPs. The aim of the experimental study is to select a limited number of testing conditions and to establish a methodology to assess, more realistically the HP performance.

### **1. INTRODUCTION**

The European commitment to limit greenhouse gas emissions and to improve energy efficiency, together with the increase of fossil fuel prices, creates new opportunities for the development of heat pump (HP) technologies.

European and international standards most commonly used by manufacturers of HPs are: (ISO 13256), (ISO 5151), and (EN 14511). These standards describe the test protocols by which the performance of HPs is established. These performances are measured by tests conducted in steady state, non-cycled, with a maximum operating load of the HP for an outdoor temperature of 7 ° C and an outlet water temperature of 35 ° C. This type of operating conditions is far from real operations. It can even be quite unrealistic as it excludes part-load operation, on-off cycling and defrosting cycles at low outdoor temperatures. In fact, energy performances have to be expressed in terms of Seasonal Performance Factor (SPF), which is the useful heat delivered during the heating season divided by the electrical energy provided to the compressor and auxiliaries (pump, fans, control, standby...).

Currently, there is no European standard for determining the SPF of heat pumps for realistic operating conditions, taking into account partial loads and outdoor temperatures variations. To demonstrate the impact of these factors on the system performance, an experimental study has been conducted on three "air-to-water" heat pumps. This study aims at establishing a methodology to assess more realistically the energy performance of heat pumps (HP).

## 2. EXPERIMENTAL APPARATUS AND TEST METHODOLOGY

### 2.1 Description of the experimental apparatus and measurements

The design and the implementation of the test bench and the air treatment unit had to comply with several requirements. Measurement systems and control conditions of the temperature and of the relative humidity meet the basic requirements of the European standards (EN 14511) such as the size of the climatic room, the recording time steps, and the accuracy of the temperature and the relative humidity sensors.

Experiments were carried out on a specifically designed test bench comprising two climatic rooms. A schematic diagram of the experimental test bench is shown in Figure 1. The air treatment unit is installed in the left climatic room. The room temperature is controlled with an air-to-air refrigerating system, where the evaporator is installed inside the unit. The humidity is controlled by a humidifier. The temperature and the relative humidity of the air treatment unit are measured in several points. Temperatures are measured at the inlet and outlet of the condenser and of the evaporator on the refrigerant side. The water temperatures at the outlet and inlet of the condenser are also measured as well the water mass flow rate. The HP compressor, the evaporator fan, and the circulating pump power consumption are measured independently. The water loop is installed in the right climatic room. A 200-L tank stores the hot water produced by the HP, and creates a thermal inertia. The heat loss is simulated by the chiller evaporator installed on the water loop and corrected by the heater. The water temperature is measured on the different points indicated on the water loop.

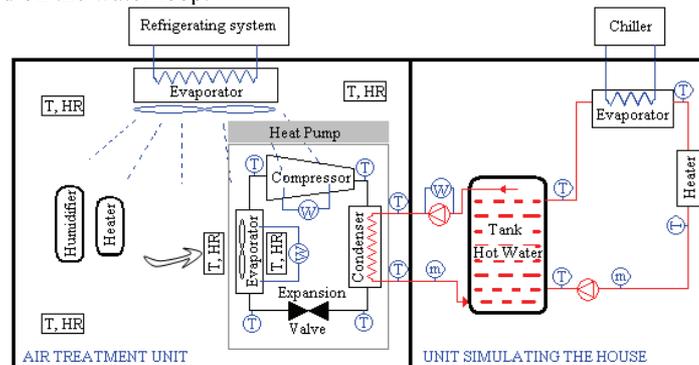


Figure 1: Test bench layout.

### 2.2 Test methodology

The protocol established by the Centre for Energy and Processes (CEP) aims at simulating actual operating conditions with heat loads varying according to the outdoor temperature. The objective of the study is to test three air-to-water heat pumps, and to evaluate the seasonal performance factor of each one. Tests are conducted for outdoor temperature varying from  $-7$  up to  $14^{\circ}\text{C}$ . The test methodology takes into account:

- the effective demand for heating, depending on the outdoor temperature;
- meteorological conditions (temperature and humidity);
- “Water temperature law” (WTL) curves, which means a linear relationship between the water temperature at the condenser outlet and the outdoor temperature, the lower the outdoor temperature, the higher the condenser outlet temperature. Three WTLs are considered: floor heating (LWTL), fan coil units (MWTL), and radiator (HWTL), L, M, and H standing for low, medium, and high;
- The selected heat transmission coefficient “U value”, is representative of the Building installed base in France;
- The indoor air comfort temperature.

The limitations of the study are:

- A single water law is used for every type of heat distribution system;
- a single heat transmission coefficient “U value” is considered;
- in heating mode, the HP is designed to supply the maximum heating capacity at  $-7^{\circ}\text{C}$  outdoor temperature.

2.2.1 Meteorological condition: meteorological conditions of the most representative cities in France, i.e. Nancy and Trappes were selected to calculate the HP seasonal performance factor (SPF). The heating season is considered

when the outdoor air temperature is lower than 15°C. The occurrences of temperature and humidity are calculated based on meteorological data collected for 10 years starting from 1995 until 2005.

2. 2. 2 Heating demand: selected heat pumps have distinct: geometry, heating capacity, and performances. All HPs, are considered to be designed with maximum capacity when the outdoor air temperature is -7°C.

In heating mode, the heat demand varies between the maximum heat capacity of the heat pump at -7°C and nil at 15°C outdoor temperature. The line connecting these two points represents the heating demand. The simulated house, where the heat pump is installed, has a constant heat transmission coefficient ( $U$ : in  $W/m^2 \cdot K$ ) and a corresponding surface ( $A$ : in  $m^2$ ). The heat demand of the house heated by the heat pump is expressed as follows:

$$H_L(T_i) [kW] = U \cdot A \cdot (T_i - T_a) \quad (1)$$

When the outdoor temperature is -7°C, the heat demand is equal to the heating capacity of the heat pump. The indoor air temperature is set to be equal to 20°C, so Equation (1) can be expressed as follows:

$$H_L(-7) = Q_{max}(-7) = U \cdot A \cdot (-7 - 20) \Rightarrow H_L(T_i) = -\frac{Q_{max}(-7)}{27} \cdot (T_i - T_a) \quad (2)$$

2. 2. 3 Heat distribution systems: water distribution systems (hydronic systems) are in France. Conventional radiator systems require high distribution temperatures, varying from 50 to 60°C; fan coil convectors are designed for a maximum operating temperature of 35 to 45°C, while 35°C is the maximum temperature for floor heating systems. In heating mode, the three water laws are tested, corresponding to each heating distribution system, for outdoor air temperature between -7°C and 15°C.

2. 2. 4 Test procedures: the test procedure is realized according to the European standards (EN 14511). The input parameters such as: outdoor air temperature and humidity, and the heating distribution system, are selected. Then, the heat loss of the house is calculated. Once all input parameters are determined, the HP is turned on. The test procedure includes the following steps:

- Preliminary phase, the outdoor temperature and humidity are set in steady state regime;
- Equilibrium phase, where the HP operates a complete on-off cycle;
- Recording phase for three hours, where the HP is in steady state cycling operation (except at -7°C where operation is continuous).

The heating capacity and HP performances are calculated using the following equations:

- The heating capacity is calculated on the condenser water side, as written in Equation (3):

$$Q(T_i) = \dot{m}_w \times C_p \times (T_{out,c} - T_{in,c}) \quad (3)$$

- The input power absorbed by the system is the sum of input powers of the compressor, the evaporator fan, and the circulating pump installed on the water loop of the condenser.
- The performance of the system is represented by the Coefficient Of Performance as follows:

$$COP(T_i) = \frac{Q(T_i)}{E(T_i)} \quad (4)$$

### 3. RESULTS AND DISCUSSIONS

#### 3.1 Comparison with the European Standards recommendations

The selected heat pumps are first tested under the conditions required by the European standards (EN 14511) i.e.: heat pump operating at full load, at 7°C outdoor temperature and 35°C condenser outlet water temperature. The measured performances at full load are first compared to the announced performances by the manufacturer. The heating demand at 7°C outdoor temperature is fixed in order to obtain a cycling mode operation of the heat pump. At these operating conditions, the mean outlet water temperature is set to be 35°C as well. Table 1 summarizes the performance obtained. The heat pumps will be presented as HP A, B, and C.

According to the European standards (EN 14511), tolerances on the performance displayed should be greater than or equal to 0.85 times the declared COP. These limits are met for HPs A and B, but not for HP C, where the difference is about 31%.

Table 1: Comparison of heat pump performances.

HP	COP announced by manufacturer (7/35°C – Full load)	COP measured (7/35°C – Full load)	Difference (%)	COP measured (7/35°C – cycling mode)	Difference COP(Full load) and COP (cycling mode)
A	3.57	3.21	10 %	2.56	20%
B	3.5	3.25	7 %	2.57	21%
C	3.9	2.7	31%	2.14	21%

Tests performed in cycling mode conditions at 7°C outdoor temperature are compared with steady state, full load tests, and present a difference of about 20% between the COP measured at full load and the COP measured in cycling mode.

The heat pump is designed to provide a full load at -7°C outdoor temperature. This means, that the HP is oversized at higher outdoor temperature. The over sizing induces an on-off control of the fixed speed electrical-driven compressor of the heat pump. This operating condition is closer to the real operating conditions. Figure 2 presents the variation of the inlet and outlet water temperatures in the condenser.

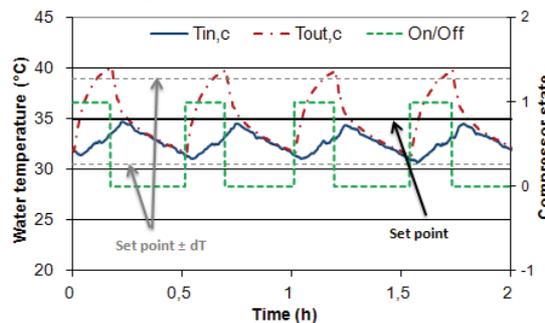


Figure 2: Cycling regime of the HP.

In these operating conditions, the water outlet temperature control of the heat pump is ensured by a succession of on / off cycles. The compressor state changes between on (1) and off (0), when the temperature reaches the low or high set point values, represented by the value “set point  $\pm$  dT” on Figure 2.

The main causes of the performance reduction in the cycling operating mode are related essentially to the following phenomena:

- The auxiliary equipment, such as the circulating water pump and the standby power consumption of the machine are absorbing electrical power during the compressor off periods;
- Losses associated to the establishment of the condensing and evaporating pressures at each startup and during defrosting occurrence.
- The HP control implies higher condensing temperatures than required when the outdoor air temperature is higher than 10°C.

### 3.2 Cycling mode tests used to calculate the seasonal performance coefficient

Cycling mode tests are performed for outdoor air temperature varying between -7°C and 14°C. The tests are repeated for the three water laws LWTL, MWTL, and HWTL, corresponding respectively to the floor, air coil and radiator heating distribution system. A total of 66 tests are performed on each heat pump. The performances of HPs are determined at every outdoor temperature, for the three WTLs. Figure3 shows the evolution of the COP values as a function of the outdoor temperature.

It may be noted that the choice of the WTL affects directly the energy performance of the heat pump. A difference of 30% is calculated between the performance obtained for the LWTL and the HWTL.

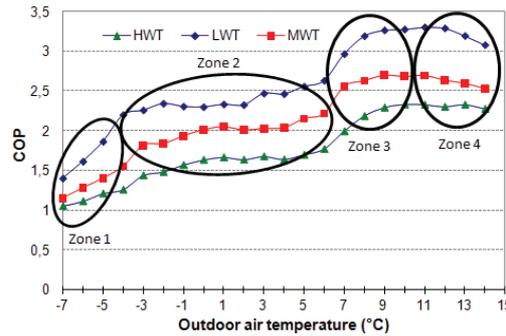


Figure 3: Evolution of the COP value of HP A, as a function of the outdoor air temperature and the WTL.

Four zones are identified on the figure:

- An area of severe degradation of the COP (zone 1), where the HP is not able to provide all the heating demand, and requires complementary heat provided by an electrical resistance.
- An area of degradation (zone 2), for temperature varying between -2°C and 6°C, due to defrosting cycles.
- Improved performance with the increase of outdoor air temperature between 6 and 11°C (zone 3).
- A new area of performance degradation due to on/off cycling (zone 4).

Zone 2 is the zone marked with defrosting cycles. When the air temperature is between -2°C and +5°C, the evolution of the measured COP is poorly variable, indicating an additional consumption of the system with outdoor temperature increase. The frost formed on the evaporator is slightly more important for outdoor temperature varying between 2 and 6°C. The supplementary electrical power consumption due to defrosting cycles occurrence were calculated for HPs A and B, in order to study the impact of defrosting cycles on the total energy performance of the heat pump (see Table 2).

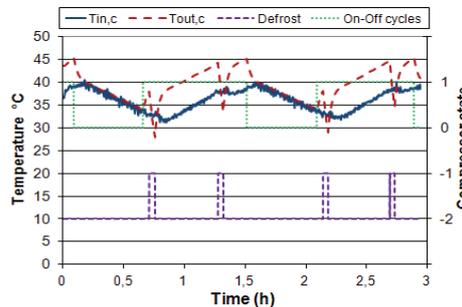


Figure 4: Impact of defrosting periods on the HP performances (HP A,  $T_i = 3^\circ\text{C}$ , MWTL).

Figure 4 shows the water inlet and outlet condenser temperatures of HP A, tested at 3°C outdoor temperature. Four defrosting cycles have occurred during the test duration of approximately three hours. The impact of defrosting on the HP performance is calculated and presented Table 2. The power consumption due to the defrosting cycle reduces the COP of the system by 4 to 12%.

Table 2: Impact of the defrosting cycle on the heat pump performance.

HP	T air outdoor (°C)	Water law	COP without defrost	COP with defrost	Performance degradation (%)
A	3	HWTL	1.76	1.63	7.4%
A	3	MWTL	2.26	2.11	6.6%
A	3	LWTL	2.76	2.42	12.3%
B	-1	HWTL	2.05	1.97	3.9%
B	1	HWTL	2.18	2.09	4.1%
B	3	HWTL	2.25	2.10	6.7%

Improved HP performance off-defrost zone (zone 3): from 6 to 9°C outdoor air temperature, the improvement of the heat pump performance is due to the absence of defrosting cycles, and the increase of the outdoor air temperature.

In zone 4 (see Figure 3), a COP improvement was expected because of the increase in the outdoor temperature. However, the performance of the HP decreases and the actual COP is lower. This result is related to the increased number of on-off cycles. This phenomenon is particularly visible on the evolution of the COP when the low and mid water temperature laws are tested.

The influence of part load factor on the HP performance has been analyzed to understand the cause of degradation occurring at high outdoor air temperature. To take into account the part loading rate, Henderson et al (Henderson et al., 2000) have introduced the Part load factor (PLF). The part load factor (PLF) is calculated according to Equation (7) as the ratio of part-load COP on COP at full load tested at the same outdoor air temperature  $T_i$ . The loading factor is the ratio of the heating load at  $T_i$  and the maximum heating capacity at the same outdoor air temperature (Equation (8)).

$$PLF = \frac{COP(\text{Part load at } T_i)}{COP(\text{Full load at } T_i)} \quad (7)$$

$$\text{Loading rate (\%)} = \frac{Q(T_i)(\text{Part Load})}{Q_{\max}(T_i)(\text{Full load})} \quad (8)$$

Figure 5 shows the variation of the part load factor (PLF) as a function of the part load rate. Tests on all HPs A, B, and C reveal the same trends. Figure 5 shows that, at very low loading rate, below 30%, a strong influence of loading rate on the PLF factor is observed; there is a decrease in performance when the loading rate is reduced. The PLF follows a linear trend when the loading rate is between 30% and 100%. This degradation is mainly due to the increased number of on / off cycles of the HP. The National Bureau of standards has performed a large study on the effect of the transient period on the HP energy performance (Kelly and Bean, 1977) and confirms that the degradation of performances is the result of the passage from static to transient phase, after the HP shutdown.

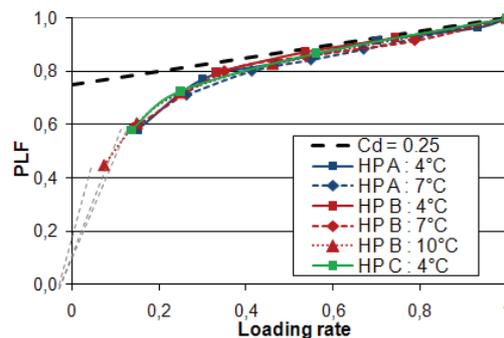


Figure 5: Evolution of the HP part load factor as a function of the loading rate.

The Part Load factor drops sharply from 70% to 0%, when the loading rate decreases from 0.3 to 0 implying a sharp fall of the COP. When the loading rate is greater than 0.6, the PLF is around 90%. Therefore, the COP decrease is low for loading rates varying between 100% and 30%, following a linear degradation curve ( $C_d = 0.25$ ), and becomes very sharp for loading rates below 30%. This area of low loading rate is called area of critical cycling, and corresponds to Zone 4 in Figure 3.

### 3.3 Seasonal performance factor

The seasonal performance factor (SPF) obtained for the two studied climatic zones in France, Trappes and Nancy, are calculated according to Equation (9). Table 3 shows the SPF calculated for the three HPs, the three water laws and the two regions of Trappes and Nancy.

$$SPF = \frac{\sum_{i=-7^{\circ}\text{C}}^{i=14^{\circ}\text{C}} n_i \times Q(T_i)}{\sum_{i=-7^{\circ}\text{C}}^{i=14^{\circ}\text{C}} n_i \times E(T_i)} \quad (9)$$

Table 3: Measured Seasonal Performance Factor.

Climatic region		Trappes			Nancy		
Water Law		LWTL	MWTL	HWTL	LWTL	MWTL	HWTL
SPF	HP A	2.70	2.26	1.85	2.60	2.16	1.77
	HP B	3.07	2.41	2.04	2.89	2.31	2.00
	HP C	2.74	2.35	1.80	2.63	2.23	1.71

#### 4. SEASONAL PERFORMANCE FACTOR

Figure 6 presents the variation of the heating capacity supplied by the HP, the input power consumed, and the HP COP expressed as a function of the outdoor air temperature. Figure 6 shows that:

- The evolution of heating power is linear with the outdoor air temperature;
- The evolution of the input power consumption is linear with the outdoor air temperature. However, the COP slope changes, corresponding to the aforementioned performance degradations.

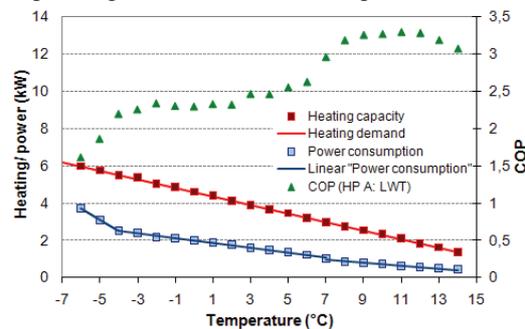


Figure 5: Evolution of the heating capacity, the electrical power consumption and the HP COP.

While observing the evolution of these parameters for the other WTLs and the HPs B and C, it can be concluded that the same trends are observed. Similarly, the transition temperatures from one operating mode to another are nearly identical. Hence, the method used to calculate the Seasonal Performance Factor of a HP requires four test points for each WTL, at partial load:

- The four selected test points corresponds to the outdoor air temperature equal to: -3, 3, 7, and 14°C;
- The heat capacity and the input power consumption, corresponding to intermediate temperatures are interpolated from the measurement results;
- The COP can then be calculated for every temperature varying between -3°C and 14°C. The Seasonal Performance Factor can be calculated as well from the interpolated heating capacity and the input power consumption, and the occurrence of the outdoor air temperature.

Table 4: Comparison between calculated and measured SPF's.

		LWTL		MWTL		HWTL	
		Trappes	Nancy	Trappes	Nancy	Trappes	Nancy
HP A	Measured SPF	2.70	2.60	2.26	2.16	1.85	1.77
	Calculated SPF	2.70	2.62	2,25	2.17	1.79	1.73
	Difference	0.14%	0.74%	-0,13%	0.89%	-2.81%	-2.24%
HP B	Measured SPF	3.07	2.90	2,41	2.31	2.04	2.00
	Calculated SPF	3.03	2.92	2,39	2.31	2.10	2.02
	Difference	-1.01%	0.94%	-1,15%	-0.14%	1.97%	1.11%
HP C	Measured SPF	2.74	2.63	2,35	2.25	1.82	1.74
	Calculated SPF	2.65	2.54	2,33	2.24	1.77	1.68
	Difference	<b>-3.29%</b>	<b>-3.10%</b>	-0,62%	-0.21%	-2.52%	<b>-3.46%</b>

The difference between the calculated and the measured Seasonal Performance Factor of the three HPs, the different WTLs, and the two climatic regions is summarized in Table 4. A maximal difference of 3.5% is calculated between the measured and calculated SPF's according to the interpolation method.

## 4. CONCLUSIONS

The current European standard EN 14511 is not satisfactory, and energy performances expressed according to this standard overestimate the actual energy performances of the equipment. A difference of 7 % to 30% is calculated between declared performances of the machine and measured ones.

Experiments have been performed on three air-to-water heat pumps and showed that the SPF is close to 3 only for the floor heat distribution system.

An empirical method for the SFP calculation of a heat pump, based on four tests, is proposed. The corresponding outdoor temperatures selected are -3, 3, 7, and 14°C. The method was applied on all the tested HPs. A maximum difference of 3.5% was calculated between measured and calculated SPF. This method has to be validated on other heat pumps.

## NOMENCLATURE

The nomenclature should be located at the end of the text using the following format:

H	Heat	(kW)	<b>Subscripts</b>	
U	heat transmission coefficient	(W/m <sup>2</sup> .K)	L	Loss
A	surface	(m <sup>2</sup> )	i	indice
T	temperature	(°C)	a	ambient
Q	heating capacity	(kW)	max	maximum
m	mass flowrate	(kg/s)	w	water
C <sub>p</sub>	specific heat	(kJ/kg.K)	c	condenser
E	Electrical power consumption	(kW)		
COP	Coefficient Of Performance	(kWh/kWh)		
n	occurrence	(-)		
SPF	Seasonal Performance Factor	(kWh/kWh)		

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