EXPERIMENTAL PERFORMANCE CHARACTERIZATION OF A NEW SINGLE ROOM VENTILATION DEVICE WITH HEAT RECOVERY

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ABSTRACT

Nowadays, important efforts are made to reduce the residential building energy consumption. In this context, a growing interest for heat recovery ventilation has been observed during the last decades. The present paper focuses on a new single room ventilation with heat recovery. Double flow ventilation is achieved through the integration of the unit into windows ledges. The developed device is particularly suitable compared to traditional centralized heat recovery ventilation units for retrofitted houses due to the absence of air extracting and air pulsing ducts through the house.

The first part of the paper consists in describing the characteristics and properties of the developed device (volume, components, flow configuration, advantages and drawbacks).

In the second part of the paper, an experimental approach is presented to characterize the unit. The criteria of performance are based on:

- Thermal effectiveness of the unit (testing of a recovery heat exchanger),
- Hydraulic aspects (flows delivered by the unit vs energy supplied to the unit),
- Acoustic aspects.

The overall performance of the unit can be established based on the experimental results described here above. Cartography of performance (ratio between the recovered heat and the supply electrical power) can be drawn, depending on the flow rates delivered by the unit and the indoor/outdoor temperature difference.

The last part of the paper compares the new system with natural, simple exhaust ventilation and traditional centralized systems in terms of primary energy, consumer price and carbon dioxide emissions. Results show that the presented device seems more competitive than natural and simple exhaust ventilation for the Belgian climate. The single room ventilation investigated in this paper also shows better performance than most of the centralized ventilation systems tested on site.

KEYWORDS

Ventilation, heat recovery, laboratory measurement, air-to-air heat exchanger

1 INTRODUCTION

According to Pérez-Lombard (2008), in 2004, energy consumption of buildings represented 37% of the total final energy consumption of the EU, corresponding to a larger share than industry (28%) and transports (32%) sectors respectively.

The residential sector accounts for the major part (70%) of this building energy consumption. As referred in the Trias Energetica concept (2012), the first step to make a building climate-friendly is to reduce the energy demand by implementing energy-saving measures. To this end, the first retrofit options to be considered for existing residential buildings are the improvement of the thermal insulation and air tightness. Improving the building envelope tends to increase the relative part of the energy consumption due to ventilation. According to Roulet et al. (2001), more than 50% of the total energy losses can be due to ventilation losses, in building with a high thermal insulation. In this context, a large amount of heat recovery technologies have been developed in the last decades (Mardiana-Idayu and Riffat (2012)).

As referred by Fehrm et al. (2002), heat recovery ventilation dedicated to residential building started in the late seventies in Sweden. Heat recovery ventilation has now acquired a status of efficient ventilation strategy, especially for buildings with low or zero energy consumption (Handel (2011)). The supplementary study on Ecodesign Lot 10 (2012) estimates a potential market of 937500 mechanical heat recovery units to be met in 2025 in the EU 27, with an explosion of sales in the medium climate market. As reported by Wouters et al. (2008), this trend was already observed in Belgium (in the frame of the Walloon project *"Construire avec l'énergie"*) with an increasing of the share of the balanced mechanical ventilation systems.
Recently, a large amount of papers about heat recovery ventilation has been released in the scientific literature but these papers focus more precisely on the heat recovery exchanger. Adamski (2008a) carried out experimental studies and developed correlations on a longitudinal flow spiral recuperator. Fernandez-Seara et al. (2010) experimentally studied an off-the-shelf air-to-air heat recovery device for balanced ventilation. Kragh et al. (2008) also experimentally investigated a new counter-flow heat exchanger but focused more precisely on the frosting issue. A thermoeconomic investigation was carried out by Söylemez (2000) in order to optimize heat recovery exchanger size. Adamski (2008b) (2010) also estimated the financial effect due to the use of heat recovery ventilation instead of a simple ventilation system.
The present paper focuses on the performance characterization of a balanced single room ventilation unit with heat recovery. To the best knowledge of the authors, only the papers of Manz et al. (2000) and Schwenzfeier et al. (2009) presents experimental investigation of such units. The present investigated device is rather different in terms of components/flows configuration, dimensions and flow inlet/outlets geometry. Volume of the whole investigated unit is 0.041 [m^3] (1.05 X 0.148 X 0.265 [m^3]).
Finally, it should also be noticed that recent studies (Laverge (2011), Maripuu (2011)) investigated the potential of demand controlled ventilation (DCV), which could be particularly suitable with balanced single room ventilation.

2 PRESENTATION OF THE DEVICE

2.1 Centralized ventilation vs single room ventilation with heat recovery

As already specified, the principle of heat recovery ventilation is well-known, but most of already commercialized units are centralized (the supplementary study on Ecodesign Lot 10 (2012)), which involves air extracting and air pulsing ducts through the house. Usually, vitiated air is extracted from wet rooms such as bathroom, kitchen and fresh air is pulsed into dry rooms such as living room, bedroom (Dimitroulopoulou (2012)). This system is known in Europe as system D with heat recovery (NBN D50-001).

Some advantages of single room ventilation are listed by Manz et al. (2000):

- *Local ventilation units do not need any ducting within the dwelling and are therefore very suitable for retrofitting use.*
- *Independent ventilation per room is possible with optimal adjustment to local needs.*
- *Local room ventilation allows quick removal of pollutants from a source-room, before they mix up with the air in other rooms as might happen with central dwelling ventilation.*
- *A direct sound transmission from room to room through the ventilation system cannot occur.*

Others advantages can be added to this list:

- Avoiding ducts means shortening the hydraulic circuits, and hence the pressure drops related to the passage of air flow rates through them. From this fact, the specific fan power (SFP) can be reduced.

- Given their placement in habitable rooms and the accessibility of each component, the maintenance of the system (particularly, the filters replacement) is easier and cheaper than in centralized heat recovery ventilation systems.
- As referred by Wouters and Van den Bossche (2005), possible problems of installed centralized ventilation systems are leaking air ducts. According to Andersson (2013), *"many studies have identified defective ventilation and insufficient air flow as a mean reason for occurrence of sick building… Duct systems accounts for a large fraction of the energy use in a building. This is further increased with a leaky duct system."* These potential issues are avoided in single room units.
- Dust accumulation in ducting can lead to a performance degradation of the installation due to a rising of the pressure drop (Anon (2000)). Moreover, the indoor air quality can decrease due to a contamination of air flow rate by particles, micro-organisms or volatile organic compound (Barbat and Feldmann (2010)). Once again, these problems are avoided in single room ventilation units.

But these advantages imply a considerable challenge: developing a competitive heat recovery ventilation system despite of a small available volume by taking care of the aesthetic aspects. As for every heat recovery ventilation system, the developed device faces with a trade-off between a high thermal effectiveness and a related rise of pressure drops inducing a degradation of the global performance of the unit due to a higher energy use for the fan. Greater attention is paid to hydraulic performance than in centralized systems since they are directly related to the noise generated by the fans. Indeed, in the design step of this kind of device, it is important to keep in mind that the heat recovery device will be installed in life rooms and has to be as silent as possible. In Belgium, according to the NBN S01-400-1, requirements for each type of local are summarized in Table 1:

Table 1: Requirement in terms of acoustic comfort according to the Belgian norm NBN S01-400-1 for mechanical ventilation

Local	Normal acoustic comfort level	Superior acoustic comfort level
Bathroom, toilets	≤ 35 dB	≤ 30 dB
Kitchen	≤ 35 dB	≤ 30 dB
Life room	≤ 30 dB	≤ 27 dB
Bedroom	≤ 27 dB	≤ 25 dB

The World Health Organization recommends two values in the report *"Guideline values for community noise in specific environments"(1999)*: respectively, 35 dB for life rooms and30 dB inside bedrooms.

2.2 Investigated device characteristics

The investigated device has been recently developed in the frame of the Green + project. Several aspects of the device have been the object of several papers during the development steps:

- Aparecida et al. (2011) presents the main design steps of the unit,
- Masy et al. (2011) focuses on the interaction with the building air tightness and indoor hygro-thermal climate,
- Ajaji and André (2012)) focuses on the ventilation efficiency.

The present paper aims to compare the overall performance (hydraulic, thermal and acoustic) of the final unit with the natural, simple exhaust ventilation and traditional centralized systems. The investigated device consists of a parallelepiped box containing two fans and two filters (for both fresh and indoor air flow rates), an electronic fan control, a set of sensors (depending of the model) and a heat recovery exchanger. Flow configurations inside the unit are represented in Figure 1.

The specificity of the units is the easiness of integration in the windows ledge, which makes them especially convenient in the frame of a house retrofitting (windows removal). Most single room ventilation with heat recovery units are installed on a wall with air inlet and air outlet through the building façade.

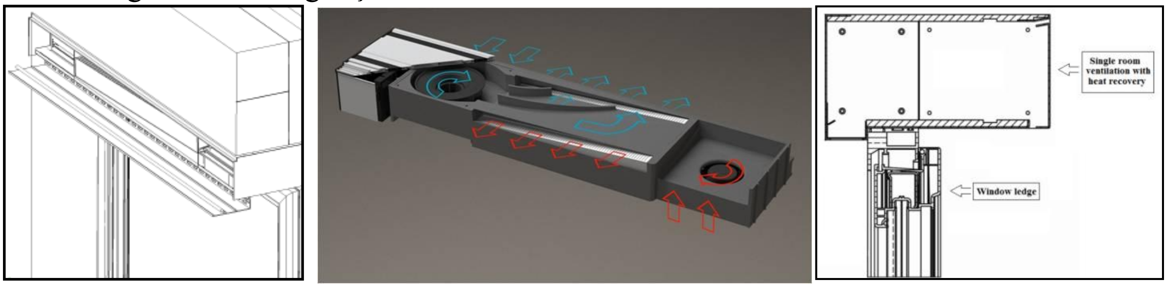

Figure 1: Investigated single room ventilation unit and flow configurations inside the device

The heat exchanger is the key component of the unit. The heat exchanger under investigation is a U-flow configuration heat exchanger. Nasif et al. (2010) has already investigated an enthalpy heat exchanger that presents a quite similar flow configuration (Z-flow configuration). Such exchangers (also called quasi-counter flow heat exchanger) present a counter flow configurations over the major part of their heat transfer area. The investigated heat recovery exchanger is made in polystyrene. The main disadvantage of polystyrene heat exchangers concerns their low thermal conductivity. However, this disadvantage can be counter-balanced by the high enlargement factor (ratio of the developed length to the protracted length) that can be reached with polystyrene heat exchangers compared to traditional plate heat exchangers made of metal (rarely superior to 1.5 according to Ayub et al. (2003)). The enlargement factor is close to 4 in the central part of the heat exchanger.

Filters dedicated to the indoor and outdoor air flow rates are placed upstream the fans and hence upstream the heat exchanger in order to protect the unit and its component against dust accumulation. Moreover, the system is designed in such a way that both filters are accessible from the inside of the house. The range of classification of available filters for the unit is comprised between G3 to F7 types, according to EN 779. The investigated single room ventilation was tested with G4 filters.

3 PERFORMANCE OF THE DEVICE

3.1 Components performance of the unit (design step)

In the design step of the device, several components and several combination of their integration have been investigated a large amount of time. From this fact, it was important to develop test benches that could be easily used for several geometries and configurations.

A test bench dedicated to the thermal/hydraulic performance of the heat exchanger has been constructed. Another test bench was developed to investigate the hydraulic performance of the device through the determination of the fan performance and the relation between the flow delivered and the electrical power supplied to the device. These test benches, their characteristics and some of intermediate experimental performance results are given by Gendebien (2012).

3.2 Final overall performance of the unit

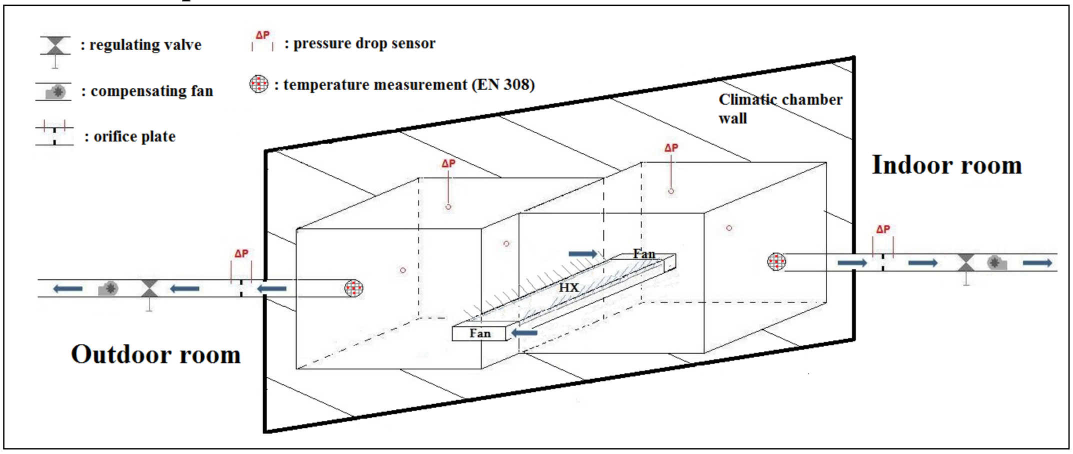

Figure 2: Schematic representation of the experimental apparatus dedicated to the thermal performance of the entire unit

In order to take into account the conduction effects in the unit and an eventual degradation of thermal performance due to a mis-distribution of the flow rate through the heat exchanger, the best way to determine the overall performance of the final device is to test it into a climatic chamber, as schematically shown in Figure 2.

The idea is to place the unit in a wall separating an outdoor and an indoor room of a climatic chamber. Flow rate delivered by each side of the unit are measured by the pressure compensated box method (Lebrun and Hannay, 1972). The mean outlet temperature of each side of the device is determined by means of five thermocouples T (placed as mentioned by NBN 308) situated at the exhaust of the pressure-compensated box. COP of the system is directly deduced by measuring the supply electrical power delivered to the unit.
Since the exhaust and the inlet of the unit consists of slits, it is important to mention that ensuring the air tightness between the unit and the experimental apparatus takes a large amount of time. That is the reason why this experimental apparatus is not suitable for the design step of the device but only for the overall performance of the final version of the device.

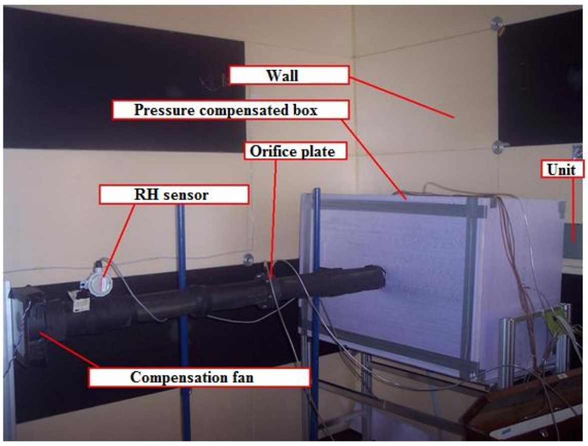

Figure 3: Climatic chamber test (outdoor side)

The overall performance of a centralized heat recovery ventilation is highly dependent on the hydraulic circuit (length and bending of the pulsing and extracting ducts) and so on the house and ducts configuration. In contrary, the overall performance of a single room heat recovery ventilation is not influenced by the rest of the installation.

Previous studies (Gendebien et al., 2013) have highlighted the fact that the annual amount of latent heat rate compared to sensible recovered heat can be neglected in moderate climates such as Belgian climate. From this fact, it is has been decided that the following results do not take into account the potential latent heat transfer rate in the establishment of the recovered heat transfer rate. The overall performance of the unit can be defined by the ratio of the recovered heat transfer rate to the electrical power of the fans and is given by Equation 1:

$$COP = \frac{Q_{recovered}}{W_{fans}} \tag{1}$$

By neglecting the potential increase of heat recovered due to latent term, the recovered heat transfer rate is given by Equation 2 and depends on the heat exchanger effectiveness (varying with the mass flow rate), the delivered mass flow rate and on the indoor/outdoor difference temperature:

$$Q_{recovered} = M_{fresh} \,.\, cp \,.\, \varepsilon \,.\, (T_{ind} - T_{out}) \tag{2}$$

with M_{fresh} the fresh air mass flow rate in [kg/s], *cp* the air capacity in [J/kg-K], ε the heat exchanger effectiveness [-], T_{ind} the indoor temperature and T_{out} the outdoor temperature. Cartography of performance can be drawn, depending on the delivered flow rate by the unit and the indoor/outdoor temperature difference, as shown in *Figure 4*:

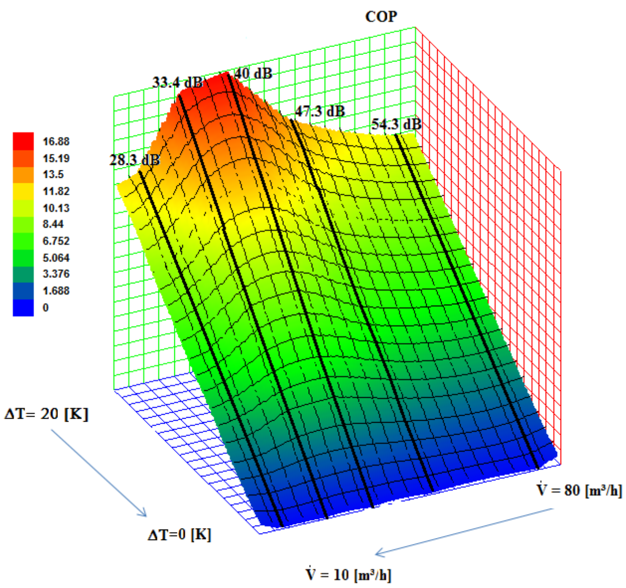

Figure 4 : COP [-] vs flow rate in [m^3/h] and difference indoor/outdoor temperature in [K] (performance cartography of the unit)

Sound pressure levels have been determined on the inner side of the unit. In order to have a complete cartography of performance of the device (thermal and hydraulic), the level of generated noise related to a specific flow rate is also indicated in Figure 4.

4 CO2 EMISSIONS, PRIMARY ENERGY AND ENERGY COSTS OF THE DEVICE

4.1 Competitiveness of the device

As shown in the previous section, the energy saved by the investigated device is highly dependent on the indoor/outdoor temperature difference. Many authors use a heating degree days (HDD) method to determine how much a heat recovery system is competitive in a given climate. For example, Adamski (2010) used it to estimate the financial effects of a ventilation system with a spiral recuperator in Poland. Kristler and Cussler (2002) combined the heating degree days and the absolute humidity days to define a cost effectiveness ratio (division of the actual energy cost savings of the investigated device by these energy costs) to optimize the performance of their membrane heat exchanger. More recently, Laverge and Janssens (2012) used the heating degree day method to evaluate the advantage of natural, simple exhaust mechanical ventilation and heat recovery ventilation over each others for European countries. In the frame of this study, the method is applied for mean average values for Europe and Belgium which can be considered as a typical moderate European climate.

The total annual heat recovered in [J/year] by the investigated device can be determined by integrating Equation 3 over one typical year:

$$Q_{recovered} = \quad V(t) \,.\, \rho \;\; t \;\; .\, cp(t) \,.\, \varepsilon \;\; V \;\; .\, \Delta T\,(t)\;\, dt \tag{3}$$

The total electrical energy delivered to the unit over one year in [J/year] can be determined by Equation 4:

$$E_{el} = \quad [\, W_{fan,fresh}(V) + W_{fan,out} \;\; V \;\;]\, dt \tag{4}$$

Equations 3 and 4 are quite difficult to evaluate since the flow rate delivered by the unit and hence the effectiveness and the electrical fan consumption vary with time and is dependent on many factors (type of ventilation control, type of room where is placed the unit, user's behavior,…). In the frame of this study, it has been decided to make some assumptions to solve them. Some assumptions are the same than the one used by Laverge and Janssens (2012):

- the ventilation system is considered to permanently run all along the year,
- the specific heat capacity cp [J/kg-K] and the air density ρ [$\mathrm{kg/m^3}$] are considered constant all year long and their products are equal to 1224 [J/m^3-K],
- integration of the indoor/outdoor temperature difference over a year can be realized through the use of the number of heating degree days HDD [K day]. According to Eurostat (2013), the heating degree for a given day is equal to the difference between 18°C and the mean outdoor temperature but only if this average daily outdoor temperature is inferior to 15°C. On the contrary, it is assumed equal to zero. The mean outdoor temperature is defined as the mathematical average of the minimum to the maximum temperature of that given day.
  Values used in the frame of this study for Europe and Belgium come from Eurostat (2013) and corresponds to the mean heating degree days over the period 1980-2004. They are respectively for Europe and Belgium equals to 3253 and 2872 [K day],
- effectiveness of the system is considered constant all year long.

By using the enounced assumptions and by normalizing Equation 3 and 4, one can determine the total annual heat recovered per m^3/h $q_{recovered}$ in [Jh/m^3-year] and the annual electrical energy delivered to the unit per m^3/h for both fans e_{el} in [Jh/m^3-year]. For the completeness of the paper, the main equations proposed by Laverge et Janssens (2012) are recalled here:

$$q_{recovered} = 24 \,.\, 1224 \,.\, HDD \,.\, \varepsilon \tag{5}$$

$$e_{el} = 24 \,.\, 365 \,.\, SFP \tag{6}$$

with SFP, the specific fans power in [J/m^3]. In order to take into account some potential variation of the ventilation flow rate, the value used for the SFP of the unit and the effectiveness of the unit in Equations 5 and 6 is the mean average value related to five rotational speeds covering the flow rate range of the unit. The average effectiveness is equal to 0.748 and the total SFP for both fans is equal to 1376 [J/m^3].

The device can be evaluated by means of three performance parameters: CO_2 emissions, primary energy and energy costs of the device. Hence, the competitiveness of the heat recovery device is demonstrated if the dimensionless number Ω, defined in Equation 7, is superior to one for each of the investigated performance parameters:

$$\Omega = \frac{q_{recovered} \cdot f_{fuel}}{e_{el} \cdot f_{el}} = \frac{q_{recovered}}{e_{el} \cdot f} > 1 \quad (7)$$

with f_{fuel} and f_{el}, the traditional conversion factors for the space heating fuel and electricity. It is assumed that the equivalent of the recovered heat is generated with a100% efficient natural gas combustion. f is the conversion factor for 1J of electricity to 1J of gas fired heating for $C0_2$ emissions, primary energy and energy costs. Values used for f in the frame of the study for Europe and Belgium are listed in Table 2:

Table 2 : Used value for conversion factor for Europe and Belgium

Conversion factors	**Values**		**References**	
	UE	**BE**	**UE**	**BE**
CO2	1.72	1.16	Laverge and Janssens (2012)	Stabbat (2009)
Primary energy	2.74	2.5		Walloon EPB decree (2008)
Energy costs	2.8	2.9		Eurostat (2013)

Numerical values for Ω_{SRVHR}, determined from Equation 7, for CO2, primary energy and household consumer prices are resumed in Table 3 :

Table 3 : Ω_{SRVHR} values

	Ω_{SRVHR}	
	UE	**BE**
CO2	3.89	4.51
Primary energy	2.437	2.09
Energy costs	2.12	1.80

As shown in Table 4, Ω_{SRVHR} is higher than one as well for the UE as for Belgium. From this fact, the investigated device seems to be competitive from an environmental and economic point of view.

It is also possible to use the method to determine the minimal HDD from which the device is competitive given several values of conversion factor. For Belgium, the most restrictive conversion factor concerns the energy costs. By taking this latter, the minimal HDD from which the device is competitive is 1600 [K day]. That corresponds to HDD of a low energy building in Belgium (base temperature chosen for the determination of the HDD is 12.5 °C).

4.2 Comparison with other ventilation systems

In the present section, the device is compared with three other ventilation systems: natural, simple exhaust and “traditional” centralized heat recovery ventilation. Given results in Table 3, it is clear that the system is more competitive than natural ventilation since Ω_{SRVHR} is higher than one for each investigated case as well for Belgium as for Europe. The investigated system is even more competitive compared to the simple exhaust ventilation since the latter involves a supplementary electrical consumption related to exhaust fans compared to natural

ventilation. The comparison with traditional centralized heat recovery ventilation appears to be more complicated since the SFP of traditional centralized system is highly dependent on the used fan and on the hydraulic characteristics of ducts. According to the European standard EN 13779 [XX], Laverge and Janssens [XX] propose to take the boundary between SFP 3 and SFP 4 (1250 [J/m^3] per fan), as reference for heat recovery system.
A centralized ventilation system with heat recovery is assumed to be as competitive as the investigated device if Ω_{CHRV} is at least equal to the determined Ω_{SRVHR}. In other terms, the minimum effectiveness for centralized systems required to be as competitive as the investigated device is given by Equation 8:

$$\varepsilon_{min,CHRV} = \varepsilon_{SRVHR} \cdot \frac{SFP_{CHRV}}{SFP_{SRVHR}} \tag{8}$$

So, by assuming a total SFP of 2500 [J/m^3] (1250 [J/m^3] per fan) for a centralized heat recovery device, the required minimum effectiveness has to be equal to 1.35 [-], which is physically unrealistic. Recently, Caillou [XX] presented in situ measurements of SFP for centralized heat recovery ventilation systems. Results are given in Figure 5.

The required minimum effectiveness to be as competitive as the investigated single room ventilation is superior to unity (which is physically unrealistic) for more than half of the investigated systems (17 out of 28). By considering an average effectiveness equal to 0.9 for a centralized heat recovery exchanger, the investigated single room ventilation shows better performance for 75% of the investigated cases. To conclude, from an energetic point of view and compared to other systems on the market, performance of the investigated device sounds promising.

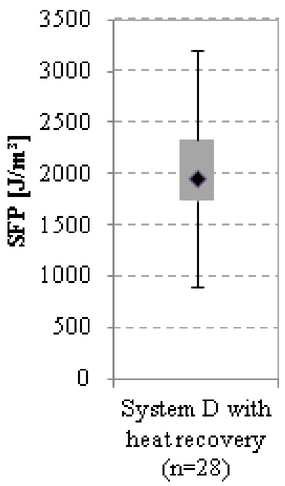

Figure 5: measurement of SFP in situ (Caillou [XX])

5 DISCUSSION

The presented COP of the device is determined in conservative conditions and takes into account the electrical conversion losses: transformation from AC (230~) to DC (24V). These losses are not negligible compared to the electrical power delivered to the fans, (especially for the low rotational speeds) and are entirely dependent on the current transformers used. In the determination of the SFP, these losses could be neglected if one assumes the presence of a DC domestic network, resulting from the use of photovoltaic panels for example.

By only taking a unique reference temperature, the method of the HDD is debatable since it doesn't take into account the thermal properties of the building, its air tightness characteristics, the solar and internal gains as well as the device operation/use. However, despite its simplicity, the method allows pointing out some trends (at a national/regional level) and permits to compare different types of heat recovery balanced ventilation in a fair way (see Equation 8). Moreover, the method also allows to determine a minimal HDD from which the device is competitive.

6 CONCLUSIONS

The present paper investigates a new single room ventilation unit with heat recovery particularly suitable in the frame of a house retrofitting. The main specificity of the investigated device is its possible integration into windows ledge. A single room ventilation unit with heat recovery presents a large range of advantages compared to centralized heat recovery ventilation but this implies a difficult trade-off between hydraulic (and hence fan noise generated by the unit) and thermal performances. An experimental procedure is presented in order to characterize the performance of the entire unit. This is realized by determining the thermal performance of the heat exchanger and the hydraulic interaction between the fans and the unit. It is proposed to graphically represent the measured overall performance of the device by means of a cartography taking into account the difference outdoor/indoor difference and the delivered flow rates. In order to have a comprehensive representation of the performance in one graphic, the generated noise level corresponding to specific delivered flow rates is also indicated. The competitiveness of the device is evaluated by means of a heating degree day method through three performance parameters: CO_2, primary energy, and energy costs. The method also permits to highlight the competitiveness of the investigated system from an energy point of view compared to other ventilation systems. As expected, the main negative aspect of the investigated device concerns the generated average noise levels which are higher for the highest delivered flow rates than the requirements provided in the standard NBN S01-400-1. However, the studied device responds to an actual growing need (high rate of retrofitting in EU). Some improvements concerning the acoustic performance of the device are currently under development.

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