MODELING AND EXPERIMENTAL VALIDATION IN PARTIALLY WET CONDITIONS OF AN AIR-TO-AIR HEAT RECOVERY EXCHANGER

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ABSTRACT

Nowadays, important efforts are deployed to reduce our current residential building consumption. The most common retrofit option concerns the air tightness and the thermal insulation improvement. However, this latter retrofit option could decrease the air indoor quality because of a reduction of air infiltration flow rate. Installation of an air-to-air heat recovery system allows for an efficient combination between consumption reduction due to the air tightness improvement and acceptable air indoor quality. The study presented in this paper has been realized in the frame of the 'Green +' project, which aims at developing decentralized heat recovery ventilation systems.

The present paper focuses on modeling and experimental validation of an air-to-air heat recovery exchanger in partially wet conditions (i.e. where condensation might occur in one of the two air streams). The knowledge of the thermal performance in dry and wet regimes is essential since it highly impacts on the heat recovered from the vitiated air flow rate (extracted from the building) to the fresh air flow rate (coming from the outdoor).

The first part of the paper briefly describes a solving procedure able to determine the regime (completely dry, completely wet and partially wet). A moving boundary model for the partially wet regime is applied in order to predict performance of such device.

Secondly, the experimental apparatus (and its control) designed to characterize thermal performance in different operating conditions (dry and wet regime) is presented.

Thirdly, experimental data are presented and analyzed, which includes a comparison with simulation results.

INTRODUCTION

Quantification of the condensate flow rate is also important in the design step of such systems (especially the decentralized ones) in order to size the condensate evacuation. Decentralized heat recovery ventilation units are wall or window-frame mounted heat exchangers coupled with fans (dedicated to extract vitiated air and to pulse fresh air into the room). Such devices are mostly used as retrofit ventilation options since they avoid any air extracting and air pulsing ducts through the house.

Accurate and robust model, able to predict the behavior in dry and wet regimes, is then needed for the sizing step and the performance characterization of such device.

Modeling and experimental investigations described in this paper concern a quasi counter-flow heat exchanger made in synthetic material.

DESCRIPTION OF THE DEVELOPED MODEL

As mentioned above, quantification of the heat performance and the condensate is important in the design step of a heat recovery exchanger and especially the decentralized ones.

Regimes of the heat recovery device

In heat recovery devices, three regimes can be observed:

- Totally dry regime: no condensation of water occurs in the heat exchanger;
- Totally wet regime: since air entering the heat recovery device is saturated, condensation appears directly after the inlet;
- Partially wet regime: condensation occurs in the heat recovery device but not directly (there is a dry and a wet part of the heat exchanger).

A schematic representation of the partially wet regime is given in Figure 1. We can observe two parts of the heat recovery device: a dry and a wet one. The interface between these two parts corresponds to a wall surface temperature equal to the supply dew point temperature of the (warm and humid) vitiated air-flow rate.

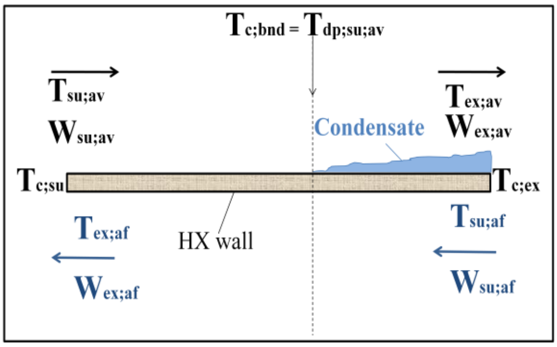

Figure 1 : Schematic representation of a partially wet heat recovery device

Developed model

Rose et al. (2008) have already presented an air-to-air heat-exchanger model. This model is based on a discretization of the heat exchanger. This latter is split into a finite number of segments, wherein the heat exchanger is considered to occur as 1D steady state. Conservation of mass and energy are expressed for each of the defined control volumes.

The present paper proposes a less time-consuming model to implement.

Actually, the presented simulation model of the heat recovery exchanger is a mix between three models initially dedicated to describe the behaviour of cooling coils:

- The one presented by Lebrun et al. (1990);
- The one presented by Brandemuehl et al. (1993);
- The one presented by Morisot et al. (2002).

All of these three models present advantages and disadvantages. The main advantage of models developed by Lebrun et al. (1990) and Morisot et al. (2002) is the simplicity. In reality, these models consider simultaneously fully dry and fully wet regimes and applies Braun hypothesis by considering that the regime to be considered (totally wet or totally dry) is the one leading to the maximal cooling capacity. Braun (1988) showed that this approximation generally leads to an error less than 5% on the prediction of the total energy rate. Lebrun's and Morisot's models show good behaviour and performance predictions, except in the case of cooling coil operation with a sensible heat ratio (SHR) around 1. This is related to the Braun's hypothesis that, for SHR close to one, assumes the coil to be completely dry. Morisot's model presents an interesting solving procedure able to identify quickly the regime.

Since the Brandemuehl's model uses a moving boundary, good performance prediction is achieved even for SHR close to one. This conclusion has already been pointing out by the authors (Gendebien et al., 2010).

The developed model describes the partially wet regime in the same way than Brandemuehl et al. (1993). It divides the heat transfer area of the cooling coil in two parts: a totally dry portion and a totally wet portion. These portions are separated by a moving boundary, whose position is determined by means of the surface temperature. In reality, this method requires two interlinked iterations: one concerning the dry part of the heat recovery device and one concerning the fresh air temperature at the interface. In the cooling coil model, water enthalpies are replaced by "fictitious fluid" enthalpies, defined as the enthalpy of saturated air at the temperature of the water.

The difference between the model presented in this paper and the one proposed by Brandemuehl et al. (1993) concerns the description of the wet regime. Here, the wet regime is described in the same manner as proposed by Lebrun et al. (1990): a fictitious perfect gas, whose enthalpy is fully defined by the actual wet bulb temperature, replaces the air.

The determination of the regime (totally dry, totally wet and partially wet) is realized by means of a solving procedure developed by Morisot et al. (2002) and presented hereafter.

In the rest of the paper, the supply fresh air temperature is always considered as lower than the vitiated air temperature (we assume that conditions correspond to winter conditions).

The implemented solving algorithm includes the following steps:

1. The first step is to compare $T_{dp;su;av}$ and $T_{su;af}$.

   Obviously, if $T_{su;af}$ is higher than $T_{dp;su;av}$, the coil is supposed completely dry.

   If $T_{su;af}$ is lower than $T_{dp;su;av}$, the coil is considered completely wet in a first time.

2. The second step is to compare $T_{dp;su;av}$ to supply cooling coil contact temperature $T_{c;su}$.

   If the supply cooling coil contact temperature is lower than $T_{dp;su;av}$, then the cooling coil is completely wet.

3. In the opposite case, the third step is to compare $T_{dp;su;av}$ and the exhaust cooling coil contact temperature $T_{c;ex}$.

   If the exhaust cooling coil contact temperature $T_{c;ex}$ is higher than $T_{dp;su;av}$, then the cooling coil is completely dry.

In the opposite case, the cooling coil is considered partially wet and the variable boundary model dedicated to partially wet regime is applied.

The “partially wet” model only requires three parameters: convective heat transfer coefficients for vitiated and fresh air and conductive heat exchanger wall resistance (generally neglected). These latter convective heat transfer coefficients are determined by means of correlations (Gendebien et al.,2011).

The whole model procedure is summarized in Figure 2.

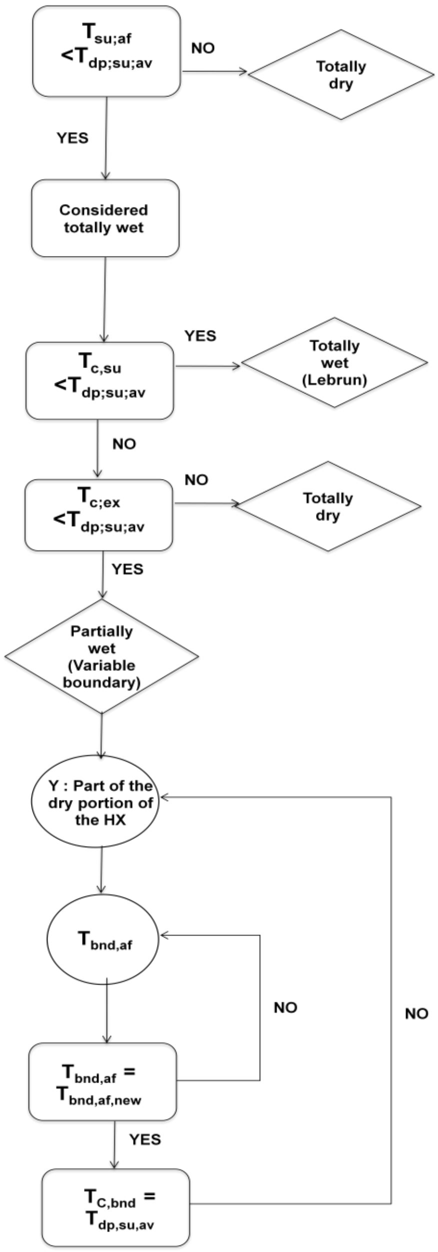

Figure 2 : Solving procedure of the developed model

EXPERIMENTAL APPARATUS

The present section presents geometric characteristics of the studied heat exchanger and offers a description of the designed test rig. In the context of the development of a heat recovery exchanger, carrying out an experimental study on an already commercialized heat exchanger can be interesting at different levels:

- To realize a performance benchmarking
- To validate by means of experimental results some of the modeling hypotheses considered during the design of the future heat exchanger.
- To point out some some defaults and/or dysfunctions of the test bench, in prevision of tests realized on the future ‘Green +’ heat exchanger.

Investigated heat exchanger

The investigated air-to-air heat recovery is made of several corrugated plate in polystyrene. The central and main region of the heat exchanger is in counter-flow and is composed of parallel triangular ducts, as shown in *Figure **3***. The inlet and outlet regions’ arrangements of the heat exchanger are cross-flows and are composed of channel with rectangular cross-sections.

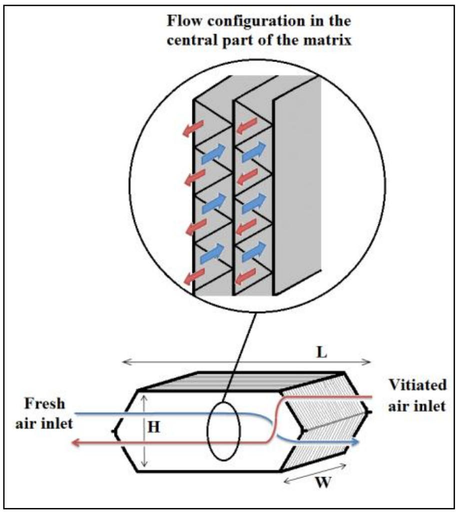

Figure 3 : Schematic representation of the studied exchanger

According to the manufacturer, the air flows (for both vitiated air stream and fresh air stream) operating range is comprised between 30 and 90 [m^3/h]. The dimensions of the heat exchanger (*Figure **3***) are 213*318*139 [mm] (W*L*H).

Test bench description

A schematic representation of the test bench is shown in *Figure 4*. Fresh air can be cooled down by means of the direct-expansion evaporator of an air-cooled chiller.

In order to avoid freezing of the evaporator, the latter is supplied with fresh air delivered by an air compressor coupled to an industrial dryer.

It is possible to control the fresh air temperature at the inlet of the heat exchanger by post-heating the fresh air flow rate with the use of variable electrical resistances. Ducts containing fresh air flow are insulated by mineral rock of 25 [mm] thickness.

Vitiated air (ambient air) can be cooled down and/or dried by by-passing a part of the flow rate exhausting from the evaporator in a mixing box situated at the inlet of the vitiated air fan. Once again, it is possible the pressure drop associated with the passage of the air flow rate through the exchanger is really important in such devices because it highly influences fan consumption and thus, the total performance of the heat recovery device.

Determination of hydraulic performance was carried out in a previous paper (Gendebien (2011)).

The relative humidities (RH) at the inlet and at the outlet of the vitiated exhaust air stream are measured by means of a humidity sensor with an accuracy of +/- 2 percent points. These sensors have been calibrated by means of LiCl and NaCl, which permits

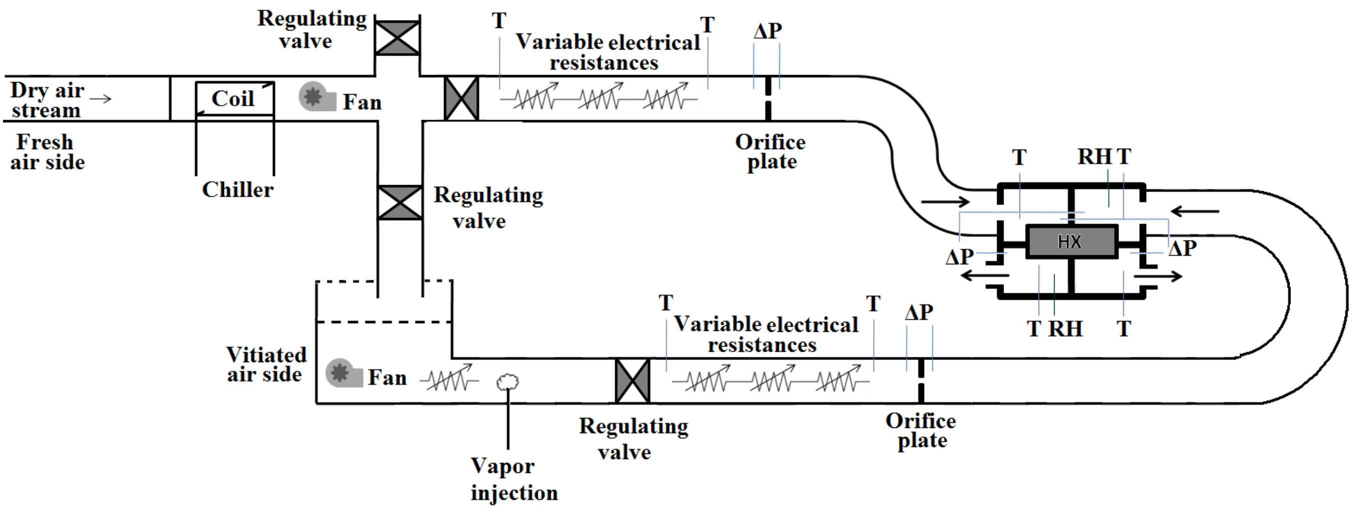

Figure 4 : Experimental apparatus

to control with precision the vitiated air temperature by means of variable electrical resistances.

Humidity is controlled by the use of electrical steam generators supplied with variable electrical power.

The mass flow rate of both fluids (fresh and vitiated air) are adjusted by means of a set of regulating valves and are measured by means of orifice plates, as recommended in the ISO 5167.
Differential pressure sensors dedicated to both air flow rates measurement have an accuracy of +/- 2.5 Pa. Air temperatures are measured with type T thermocouples with accuracy of +/- 0.3 K. In the rest of the paper, the mean supply temperature correspond to the average of two type T thermocouples measurements and the mean exhaust temperature corresponds to the average of eight type T thermocouples measurements.

The differential pressure between the inlet and the outlet of the heat exchanger is measured by means of two distinct differential pressure sensors: one dedicated to the lowest air flow rates with an accuracy of +/- 1 Pa with a full-scale value of 100 Pa and another one with an accuracy of +/- 2.5 Pa with a full-scale value of 500 Pa. In reality, quantification of to create an atmosphere at respectively 11.3% and 70% of relative humidity.

The range of the several used sensors and their accuracy are summarized in *Table 1*.

Table 1 : Accuracy of the measurement devices

Measurements	Accuracy
Type T Thermocouples	+/- 0.3K
Differential pressure sensors (Full-scale: 500 Pa)	+/-2.5 Pa
Differential pressure sensors (Full-scale: 100 Pa)	+/- 1 Pa
Relative humidity sensor (0 to 100%) Simultaneous measure of relative humidity and temperature	+/- 2 % +/- 0.4K

The heat exchanger is located in a box insulated by 30 [mm] thick polystyrene in order to reduce heat losses to the atmosphere.

TESTING CONDITIONS

The present part of the paper presents the testing conditions.

Mass flow rate

According to the manufacturer, the range of the volumetric flow rate is comprised between 30 to 90 [m3/h]. Condition concerning tests was to well-balance both mass air flow rates. This condition was fulfilled by using regulating valves of the test bench. An example of measured mass air-flow rate for both fluids is given hereafter in Figure 5. These latter air mass-flow rates correspond to the conditions in terms of relative humidities and temperature presented in Figure 7.

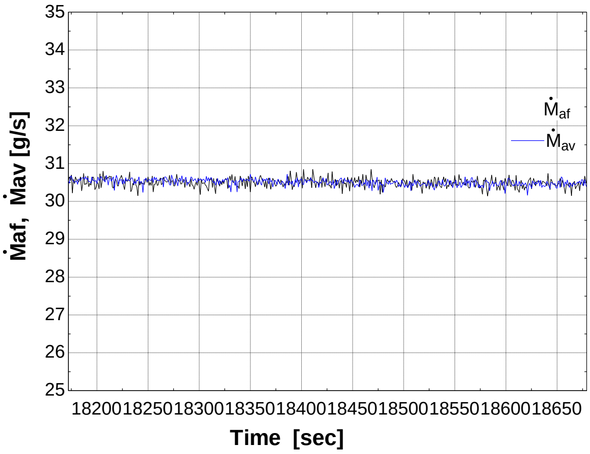

Figure 5 : Example of a well-balanced flow rates test

As we can see in Figure 6, air-flow rates are well-balanced for all of the performed tests:

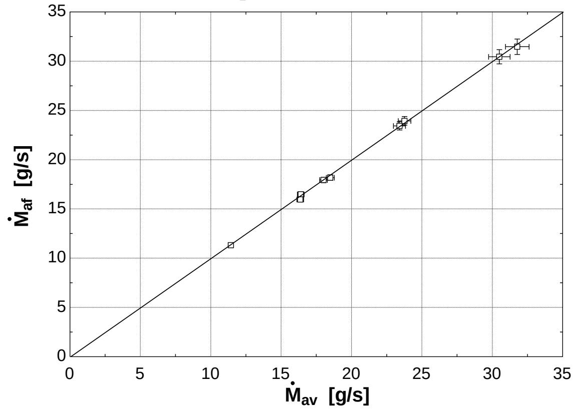

Figure 6 : Comparison of the mass air-flow rates for all tests

Vertical and horizontal bars indicate the uncertainty on measurements on the flow rates according to ISO 5167, respectively for the fresh air the vitiated air side.

Temperature and humidities

Temperatures at the inlet of the heat exchanger were respectively comprised between 0°C and 5 °C for fresh air-flow rate and between 20 °C and 25°C for the vitiated air-flow rate. Inlet relative humidity of the vitiated air flow rate ranged between 45% and 80%.

The value of the inlet fresh air temperature was chosen in order to avoid any risk of freeze and to allow the partially wet regime.

Inlet vitiated temperature and relative humidity representative of inside conditions of a domestic building were imposed.

The results presented in the following part of the paper correspond to the average value of stabilized regime of 500 [sec]. An example of stabilized test is given in Figure 7 and shows the mean temperatures at the inlet and at the outlet of the heat exchanger for both fluids. Relative humidities of the vitiated air at the inlet and the outlet of the heat exchanger are also given in Figure 7. It is interesting to stress the fact that the difference between the inlet and the outlet temperatures of the heat exchanger of both air flow rates are not the same. This fact can be explained by the appearance of condensation (and thus a part of latent heat transfer rate) on the vitiated side of the heat exchanger.

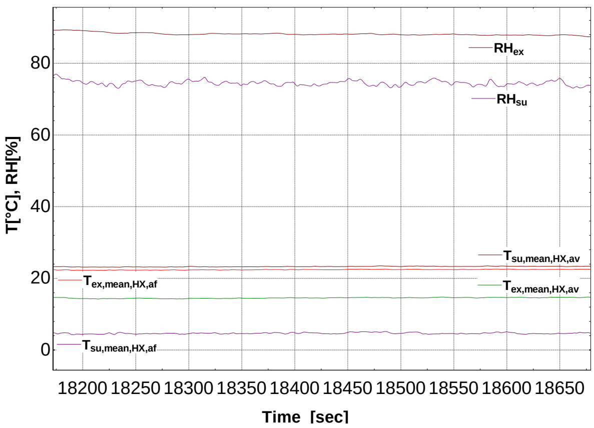

Figure 7 : Example of stabilized test

Balance of the heat transfer rate

Measured heat transfer rates on vitiated and fresh air sides are compared in Figure 8. It can be observed that energy balances on the fresh and vitiated air streams are comprised in a band of +/- 15%.

These results are considered as satisfactory for this kind of device, due to the little heat transfer rate.

Fernandez-Seara (2010) has obtained the same range of error on tests realized on the same type of heat recovery exchanger.

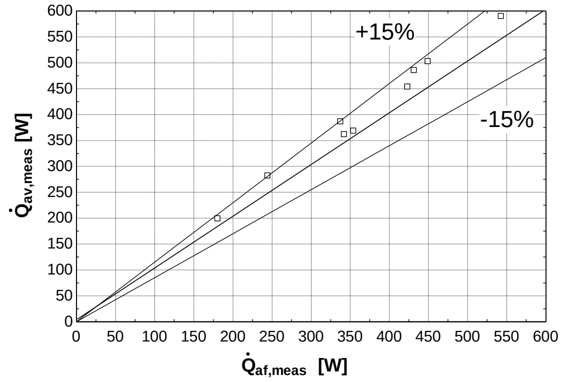

Figure 8 : Heat transfer rates determined from the experimental data through the energy balances on the vitiated and fresh air-flow rates.

VALIDATION OF THE MODEL

The aim of this part of the paper is to compare simulation results of model with experimental results. The correlations used for the estimation of the convective heat transfer coefficients were identified for the calibration of the model in dry conditions (Gendebien et al., 2011). It should be stressed that the same correlations were used for the description of the wet regime.

Validation compares simulation and experimental results in terms of :

- Total heat transfer rate;
- Sensible heat transfer rate;
- Latent heat transfer rate;
- Condensate flow rate (which is obviously linked to the latent heat transfer rate).

Comparison between model and experimental results presented hereafter are realised by comparing only measured sensible heat transfer rate. This is justified by the fact that sensible heat transfer measurement present less error than the latent heat transfer measurement (RH sensor present an important uncertainty measurement compared to the type T thermocouples). Thus, the considered measurements taken into account for the validation are:

$$\dot{Q}_{tot;meas} = \dot{Q}_{af;meas} \quad (1)$$

$$\dot{Q}_{tot;meas} = \dot{M}_{af;meas} * cp_{af} * \Delta t_{af;meas} \quad (2)$$

$$\dot{Q}_{sen;meas} = \dot{M}_{av;meas} * cp_{av} * \Delta t_{av;meas} \quad (3)$$

$$\dot{Q}_{lat;meas} = \dot{Q}_{tot;meas} - \dot{Q}_{sen;meas} \quad (4)$$

Total heat transfer rate

Figure 9 presents the comparison between model and experimental results in terms of total heat transfer rate.

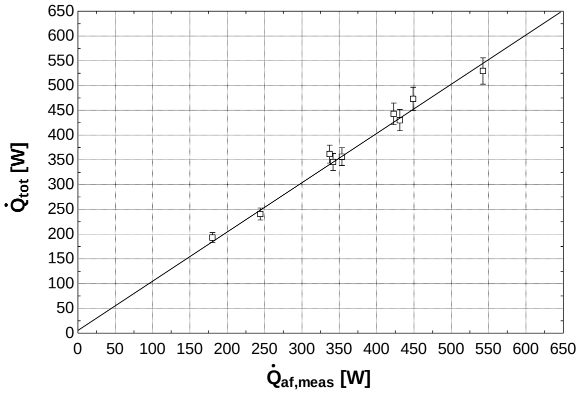

Figure 9 : Comparison between model and experimental results in term of total heat transfer rate

The model is able to predict the total heat transfer rate within 5% (vertical bars represent an error of 5%).

Sensible heat transfer rate

Figure 10 presents the comparison between model and experimental results in terms of total heat transfer rate.

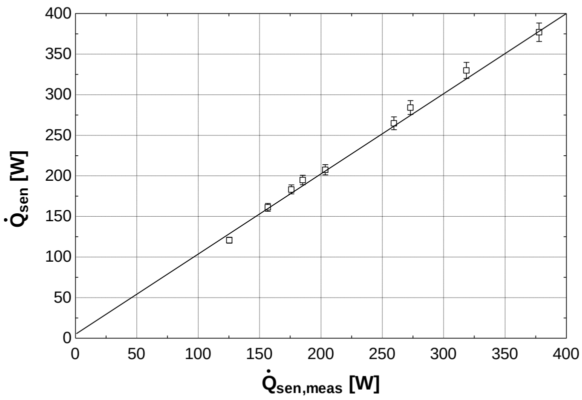

Figure 10 : Comparison between model and experimental results in term of sensible heat transfer rate

Sensible heat transfer rate can be predicted with a maximal error of 3%, as it can be observed in the previous figure.

Latent heat transfer rate

Latent heat transfer rate is predicted by the model with a mean error of 10% over all the measurements. Comparison between experimental and results model is given in Figure 11.

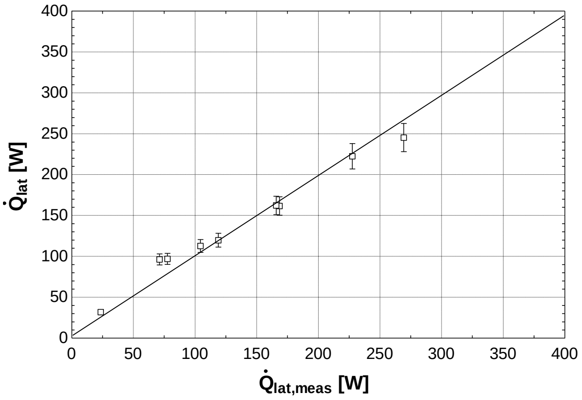

Figure 11 : Comparison between model and experimental results in term of latent heat transfer rate

Condensate flow rate

As already mentioned, condensation flow rate is obviously linked to the latent heat transfer rate. Comparison between experimental and model results is given in *Figure 12*.

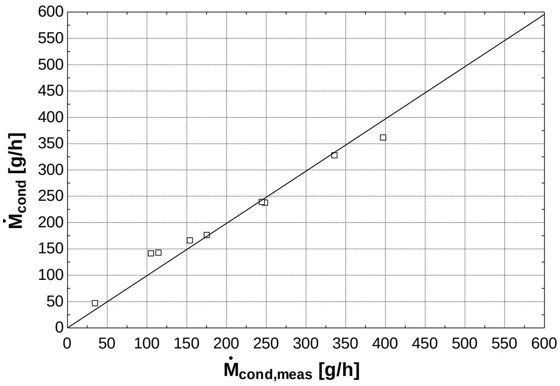

Figure 12 : Comparison between predicted and measured condensate flow rate

CONCLUSION

The present paper proposes a solving procedure and a model based on a mix of different cooling coil model in order to describe the behaviour of a ventilation heat recovery exchanger for residential buildings.

Thermal performance tests have been carried out and focused more precisely on the partially wet regime. The model is able to predict with a good accuracy the heat transfer rate (as well as latent and sensible parts). The proposed model requires only two parameters (two convective heat transfer resistances), that have been determined previously based on measurements in dry regime.

Determination of latent heat transfer rate is important in heat recovery device (and especially the decentralized one) since it allows to quantifying the condensate flow rate.

The model can be used in the design step of a heat recovery exchanger to determine the annual total amount of condensate flow rate and the heat transfer rate by integrating the presented model in an hourly-based simulation model of a domestic building.

The following step of the work will consist in the experimental study of frost formation that can occur in very low exterior temperature condition. Experimental investigations on new prototypes of heat recovery exchangers will also be carried out.

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NOMENCLATURE

Cp: specific heat [J/kg-K]	*Subscripts*
H: height of the exchanger [m]	*a:* air
L: length of the exchanger [m]	*bnd:* boundary
$\dot{M}$ *:* mass flow rate [g/s] or [g/h]	*c:* contact
$\dot{Q}$*:* heat transfer rate [W]	*cond :* condensate
RH : relative humidity [%]	*dp:* dewpoint
T : temperature [°C]	*ex :* exhaust
w : humidity ratio [kg/kg]	*f :* fresh
W: width of the exchanger [m]	*hx :* heat exchanger
Y: Dry part of the heat Exchanger [-]	*lat:* latent
	meas: measured
	new: new (result of an algorithm run)
	sen: sensible
	su : supply
	tot: total
	v : vitiated

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