Experimental investigation on a decentralized air handling terminal: procedure of aeraulic and thermal performance determination of the entire unit in several operating conditions

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Abstract. A new local ventilation device is actually developed in such a way to procure ventilation “on demand” in each room, with a maximal effectiveness. It consists in a wall or window frame mounted plane-parallel box, containing two (injection and extraction) fans, an electronic control, and a heat recovery exchanger. The present paper describes the experimental investigations carried out on some single components and on the entire unit in order to characterize the aeraulic and thermal performance of the device.

1. Introduction
Nowadays, important efforts are deployed to reduce the actual residential building energy consumption. The most common retrofit option concerns the thermal insulation and the air tightness improvement. However, this latter retrofit option could decrease the air indoor quality because of a reduction of air infiltration flow rate. Installation of heat recovery ventilation allows for an efficient combination between consumption reduction due to the air tightness improvement and acceptable air indoor quality. The principle of such system is very simple: vitiated air, extracted from the building is used to pre-heat (in winter) or pre-cool (in summer) the fresh air flow rate coming from the outdoor environment. Most of commercialized units are centralized, which involves a high number of air extracting and air pulsing ducts through the house. From this fact, centralized units are not well adapted to refurbishment because of the absence of room dedicated to pulsing and extracting ducts in existing buildings. The present paper describes the experimental apparatus developed to characterize the aeraulic and thermal performance of a decentralized air handling terminal (DAHT). Experimental investigations have been carried out on some single component (such as heat recovery exchanger or the fan) of the device but also on the entire unit in order to determine the flow rate it delivers. The first part of the paper presents and describes characteristics of the new local ventilation device. The second part of the paper presents the experimental apparatus developed to characterize the thermal and the aeraulic performance of the unit. Some experimental results are given. Finally, the discussion section presents the under-development test bench to determine the thermal performance of the entire unit in several operating conditions, such as freezing ones.

2. Presentation of the investigated device
2.1. Context and advantages
The principle of heat recovery ventilation is well-known and rather widespread, but most of already commercialized units are centralized, which involves a high number of air extracting and air pulsing ducts through the house.

![Comparison between a centralized and a decentralized air handling unit](image1.png)

**Figure 1**: Comparison between a centralized and a decentralized air handling unit

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. This system is commonly known as Controlled Mechanical Ventilation (CMV) or system D with heat recovery. As observed in Figure 1, the main advantage of the decentralized unit is the suppression of the ducts, and the easiness of placement of the units, especially after a refurbishment. But this advantage involves a considerable challenge: developing a very compact heat recovery device with a high thermal efficiency despite of a small volume. Attention has also to be paid to aeraulic performance, which has to be maximized in order to minimize the fan consumption (lower than 25 [W] expected, for the highest delivered flow rate) and hence, the generated noise. 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 rooms and has to be as silent as possible. According to the NBN S01-400-1 [1], the “normal” acoustic comfort for life and bed rooms are respectively 30 [dB] and 27 [dB]. For a considered “superior” acoustic comfort, 27 [dB] and 25 [dB] are required.

2.2. Description of the entire unit
Double flow ventilation can be handled through Decentralized Air Handling Terminals (DAHT) integrated in walls or window ledges and provided with heat recovery exchangers. Figure 2 shows a perspective and section of the studied system [2]. Flow configurations inside the unit are also represented in Figure 2.

![Investigated Decentralized Air Handling Terminal (DAHT) and flow configurations inside the unit](image2.png)

**Figure 2**: Investigated Decentralized Air Handling Terminal (DAHT) and flow configurations inside the unit

The proposed design differentiates from other ones, mainly by:
- the high effectiveness counter-flow heat exchanger,
- the adaptability of heat exchanger and fans to different window sizes and flow rates.

The investigated DAHT consists in a parallelepiped box containing two (injection and extraction) fans, two filters, an electronic control, and obviously, a heat recovery exchanger.
2.3. Heat recovery exchanger
The heat recovery exchanger is the most important component of the device. The developed heat recovery exchanger is made in synthetic material with a high enlargement factor, and hence, a high compactness (456[m²/m³]). This type of heat exchanger shows a quasi counter-flow configuration since, except the supply and exhaust zones, the central and major part of the device is in counter-flow configuration.

![Figure 3: Flow configurations in the heat exchanger](image)

2.4. Fan
One of the goals of the device is to well-balance the extracted and pulsing flow rates. The device is supposed to be symmetrical for both sides and hence, to have the same aeraulic performance curve. From these facts, identical extracting and pulsing centrifugal fans were selected.

3. Experimental investigations
The present section describes the experimental investigations carried out to determine the thermal and aeraulic performance of some components of the system.

3.1. Thermal performance: investigation on the heat recovery exchanger

3.1.1 Test bench. A schematic representation of the test bench dedicated to the heat recovery exchanger performance assessment and pictures of its practical achievement are given 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 part of the flow rate exhausting from the evaporator in a mixing box situated at the inlet of the vitiated air fan. Here also, it is possible 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.

![Figure 4: Experimental set-up to test heat recovery exchangers](image)
The mass flow rate of both fluids (fresh and vitiated air) can be adjusted by means of a set of regulating valves and are measured by means of orifice plates, as recommended in the ISO 5167 [3]. Differential pressure sensors dedicated to the 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 and exhaust temperatures correspond to the average of five measurements each by type T thermocouples, as mentioned in the NBN 308 [4].

The pressure difference 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.

The relative humidities (RH) at the inlet and at the outlet of the vitiated exhaust air stream are measured by means of humidity sensors with an accuracy of +/- 2 percent points. These sensors have been calibrated by means of LiCl and NaCl, which permits to create an atmosphere at respectively 11.3% and 70% of relative humidity.

The heat exchanger is located in a box insulated by 30 [mm] thick polystyrene in order to reduce heat losses to the atmosphere. In order to ensure a uniform air flow rate through the heat exchanger, dampers with filters are placed in the box upstream the heat exchanger.

All the measurements were automatically stored in a PC by means of several data acquisition cards.

Two tests have been carried out on the heat exchanger: one focusing on the aeraulic and the other on the thermal performance of the device.

### 3.1.2 Aeraulic performance: testing conditions and experimental results.

Concerning the aeraulic performance, the pressure drop was determined for several air flow rates (ranging approximately from 30 to 100 [m^3/h]). The conditions at the inlet of the heat exchanger for these tests were 22 [°C] and 40 [%] of relative humidity. Each presented result for the several air flow rates concerning the aeraulic performance of the heat exchanger corresponds to a stabilized time test of 200 [sec]. Two differential pressure sensors were used for the determination of the aeraulic performance: one with a full scale value of 100 [Pa] for the lowest air flow rates and one with a full scale value of 500 [Pa] for the highest air flow rates.

### 3.1.3 Thermal performance: testing conditions and experimental results.

Investigated flow rates are comprised between 0 and 100 [m^3/h]. Since the flow rates are rather small, the temperature difference between both inlets of the heat exchanger was chosen in a range comprised between 30 to 40 [K] (between 30 and 40 [°C] for the inlet vitiated air and between 0 and 10 [°C] for the inlet fresh air). Obviously, these conditions (especially the high virtual indoor temperature) do not correspond to the usual operating conditions of the device but allow minimizing the relative error on the heat transfer measurement.

The results presented in the following part of the paper correspond to the average value of stabilized time regime of 500 [sec]. The stabilized time regime is longer than for the determination of the aeraulic performance of the device. This choice has been made to avoid any potential thermal derive of the system.

The aim of these tests was to determine the performance of the heat exchanger in dry conditions. Part of the dry air coming from the dryer and exhausting from the chiller evaporator was deviated into the mixing box situated at the inlet of the vitiated air pulsing fan (see Figure 4).

The operation in dry regime was checked by means of the measurements of the humidities at the inlet and at the outlet of the heat exchanger and was achieved during the whole set of tests. In order to be as close as possible to the nominal operating conditions, another condition concerning these tests was to well-balance both mass air flow rates. This condition was fulfilled by using regulating valves of the test bench.
Since experimental conditions have been chosen in order to avoid condensation in the exchanger, heat transfer rates $\dot{Q}$ in [W] were measured on both the vitiated and fresh air sides, respectively, by the two following equations:

$$
\dot{Q}_{av} = \dot{C}_{av} \ast (t_{su,av} - t_{ex,av}) \tag{1}
$$

$$
\dot{Q}_{af} = \dot{C}_{af} \ast (t_{ex,af} - t_{su,af}) \tag{2}
$$

The difference between these two values is always lower than 3.2 %, and the mean relative difference is equal to 1.8%, which was considered as satisfying considering the small investigated flow rates.

The first step to determine effectiveness from measurements consists in weight averaging the supply and exhaust temperature measurements provided by the several type T thermocouples for each flow rates. Two effectiveness’s can be determined from measurements: the one measured at the fresh air side and the one measured at the vitiated air side.

The presented effectiveness is the one permitting to avoid error measurements on the mass flow rates by comparing the two capacity flow rates, as explained hereafter:

$$
\text{if } (\dot{C}_{min} = \dot{C}_{av}) \text{ then }
\epsilon_{meas} = \frac{\dot{C}_{av} \ast (t_{su,av} - t_{ex,av})}{\dot{C}_{min} \ast (t_{su,av} - t_{su,af})} = \frac{(t_{su,av} - t_{ex,av})}{(t_{su,av} - t_{su,af})} = \epsilon_{meas,av}
$$

$$
\text{else}
\epsilon_{meas} = \frac{\dot{C}_{av} \ast (t_{ex,af} - t_{su,af})}{\dot{C}_{min} \ast (t_{su,av} - t_{su,af})} = \frac{(t_{ex,af} - t_{su,af})}{(t_{su,av} - t_{su,af})} = \epsilon_{meas,af}
$$

3.1.4 Experimental results. The experimental results in terms of aeraulic performance for each side of the matrix (c1 and c2) are given in Figure 5a. As observed, the discrepancies between aeraulic performances for each side of the matrix reveal a dissymmetry of the matrix.

If the air mass flow rate is converted into a volumetric flow rate in standard condition (25 [°C] and 101325 [Pa]), the evolution of measured effectiveness shown in Figure 5b is obtained. Horizontal error bars indicate the uncertainty on measurements of the flow rates and vertical error bars indicate the uncertainty on measurements on the effectiveness.
3.2. Aeraulic performance of the device
The present section focuses on the experimental investigation carried out to characterize the aeraulic performance of the entire system.

3.2.1 Delivered flow rate by the device. It is rather difficult to determine the delivered flow rates by means of velocity sensors since the velocity at the exhaust of the device is quite small and not homogeneous. Moreover, use of intrusive sensors can modify the delivered flow rate. A method initially dedicated to fan-coil unit and described by Hannay and Lebrun [5] was applied.

![Figure 6: Schematic representation of the experimental apparatus to determine the delivered flow rate by the device](image_url)

This method consists in connecting the exhaust slit of the device into a pressure-compensated box. As shown in Figure 6, at the exhaust of the compensating box, a set of orifice plates is used to determine the flow rate. In order to annihilate the decrease of the delivered flow rate due to the passage through the orifice plate (which induces an additional pressure loss), a balancing fan is used. A differential pressure sensor is placed between the atmosphere and the inside of the compensated box. Once the measured differential pressure is equal to zero, the device is supposed to operate in nominal conditions and the volumetric flow rate measured by means of the orifice plate is the one really delivered by the unit.

The delivered flow rate was measured for each side of the unit (with and without filter) and for several rotational speed of the fan.

3.2.2 Determination of fan curve. The same experimental apparatus than the one used to determine the delivered flow rate has been used to determine fan curves (total pressure versus volume flow rate). Fan rotational speed can be modified by means of an electronic control integrated in the device. Since the fan manufacturer only indicates the fan curve for the nominal rotational speed, it has been decided to determine the curves for several other rotational speeds. The developed experimental apparatus is shown in Figure 7. For a given rotational speed, it is possible to modify the total differential pressure drop between the inlet and the outlet of the fan by means of the set of regulating valves and the rotational speed of the balancing fan. The mass flow rate delivered by the fan is measured by means of a set of orifice plates, according to ISO 5167 [2], at the exhaust of the pulsing box.

![Figure 7: Determination of fan curve for several rotational speed](image_url)

The total pressure drop between the inlet and the outlet of the fan is given by the following equation:

\[
\Delta P_{\text{tot}} = \Delta P_{\text{stat}} + \Delta P_{\text{dyn}} \ [\text{Pa}]
\]  

(5)
The differential static pressure drop is measured by means of differential pressure drop sensor and the
dynamical pressure drop is deduced, knowing the measured volumetric flow rate (orifice plates) and the
supply and exhaust flow areas of the fan.

3.2.3 Determination of total pressure drop for a given rotational speed. The method presented in section
3.2.1 allows for determining the volumetric flow rates for a given rotational speed of the fan. In order to
determine the aeraulic characteristic curve of the unit (flow rate – total pressure drop), the total pressure
drop at the outlet and the inlet of the fan was determined in the same manner than the one already
explained in section 3.2.2. Once again, total pressure drop was determined for a device with and without
filter.

3.2.4 Reconciliation of experimental results. The three previous tests can be used to represent the total
aeraulic performance of the device, as represented in Figure 8. The rotational speed of the fan was
measured by a tachymeter.

Figure 8: Aeraulic performance of the device

Figure 8 permits to quantify the influence of the filter on the delivered flow rate.

4. Further work
Further work will consist in testing the entire system in a climatic chamber. In reality, both (fresh and
vitiated) volumetric air flow rates are pulsed through slits of the same length of the heat recovery device.

Figure 9: Schematic representation of the experimental apparatus dedicated to the thermal performance
of the entire unit
From this fact, it is very difficult to determine a fresh and vitiated mean temperature at the exhaust of the exchanger. Moreover, 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 thermal performance of the entire device is to test it into a climatic chamber by merging all tests presenting here above, in this paper. 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 same method than the one exposed in section 3.2.3. The mean outlet temperature of each side of the device is determined by means of five thermocouples T (placed as mentioned by NBN 308 [3]) situated at the exhaust of the pressure-compensated box.

The main advantage of the present exposed method is that the test bench allows for testing in several operating conditions, which includes freezing ones. From this fact, electronic control dedicated to avoid freezing in the unit (by mis-balancing the fresh and vitiated flow rate, as an example) could also be tested. COP of the system could also be determined by measuring the supply electrical power delivered to the unit.

5. Conclusion
The present paper offers a description of the experimental investigations that can be carried out to determine the thermal and aeraulic performance of a decentralized air handling terminal (DAHT). Compared to a centralized one, even if its thermal effectiveness is lower (mainly due to the small dedicated volume), this system presents a set of advantages. From the fact that its installation avoids pulsing and extracting ducts through the house, it is appropriate for heat recovery ventilation of retrofitted houses. Avoiding ducts also means shortening the aeraulic circuit, and hence the pressure drop related to the passage of air flow rates through these. It induces an improvement in terms of environmental hygiene (by avoiding a potential dust accumulation in ducts). Accessibility of each component and hence the maintenance of the system (particularly, the filters replacement) is easier.

The first part of the paper focuses on the experimental investigation that can be carried out in dry regime on the heat recovery exchanger (one of the components of the DAHT). Experimental investigations dedicated to thermal and aeraulic performance of the components are presented.

The second part of the paper focuses on the aeraulic performance of the unit.

Finally, a method dedicated to determine the thermal performance of the entire DAHT in several operating conditions is presented.

6. References
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