A R C H I T E C T U R E C I V I L E N G I N E E R I N G E N V I R O N M E N T
The Silesian University of Technology No. 4/2019

doi: 10.21307/ACEE-2019-056

RESEARCH OF SINGLE ROOM DECENTRALIZED HEAT RECOVERY UNIT

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Received: 28.07.2019; Revised: 30.09.2019; Accepted: 2.10.2019

Abstract
Mechanical ventilation systems with heat recovery units are one of the key elements of low energy residential buildings and it has gained increasing interest during the last years. This research presents experimental investigation of the work of the compact recuperator, as a part of the decentralized, single pipe ventilation system, in real conditions. An experimental study was carried out to obtain data for tested unit efficiency calculation, for supply and exhaust mode. The scope of measurement encompassed the air temperature and airflow rate. Tested recuperator fulfilled its role of ventilation with heat recovery, however, some deficiencies were indicated. As a result of measurements it has been noticed that it is possible to improve the device in the future. This possibility of improvement of this device was suggested in the conclusions.

Keywords: **Decentralized ventilation; Heat recovery; Measurements; Single room unit; Recuperator.**

1. INTRODUCTION

New devices emerging on the market such as small ventilation equipment contribute to the need for research on their compliance with manufacturers' data. They are new, poorly known and still not very popular. However, they have great potential to ventilate single rooms. Moreover, they are compact and inexpensive. One of them is single room decentralized heat recovery unit, commonly called compact recuperator, which can be installed in almost every building in the external partition, away from sources of pollution. It is characterized by a very simple construction. Its task, in addition to providing fresh air, is heat recovery using a small accumulated heat exchanger. Together with the accumulated heat exchanger, according to the manufacturers, it is able to obtain up to 90% of heat recovery from exhaust air. It is an energy-efficient way of ventilation of single and residential rooms in the temperature range of -20 to 50°C.

The compact recuperator is a part of a decentralized, single pipe ventilation system. Typical decentralized ventilation (DV) is a type of traditional mechanical ventilation that is implemented in a similar method to fan-assisted natural ventilation. However, fresh outdoor air is supplied and distributed into the room by passing through a compact decentralized air-handling unit rather than being utilized directly [1, 2]. Compared to centralized ventilation, DV has many advantages. DV minimizes duct space, can simplify individual zoning control in spaces and is less influenced by outdoor environmental condition [2]. Furthermore, the use of decentralized installation results in reduced operating costs. However, there are also some disadvantages: possibility of location in buildings and compliance with standards [3] in terms of air inlets and outlets location. This is the reason that sometimes they technically cannot be used [4].

Nowadays, the world attaches great significance to the saving of energy. Energy-efficient construction focuses mainly on the continuous reduction of thermal losses [5]. The airtight construction required by the current state of the art of the building shell increases the need for fresh ventilation air. According to standards [3], heat recovery is obligatory in installations with an air volume above 500 m^3/h. Most often this is accomplished by recirculating air or just by the use of equipment for the recovery of heat and moisture. In small detached houses and in residential buildings, heat recovery devices such as recuperators are still seen as expensive and luxurious. However, in the future, in the light of new requirements, they will become a standard. Therefore, a compact recuperator is a noteworthy alternative, in particular for modernized and emerging residential and commercial buildings. The need for using such devices was also discussed by Amanowicz & Szczechowiak [6]. Considering the need to introduce solutions using heat recovery, also with a small supply air stream, in order to obtain a low primary energy ratio, the use of such solutions should soon be common in Poland.

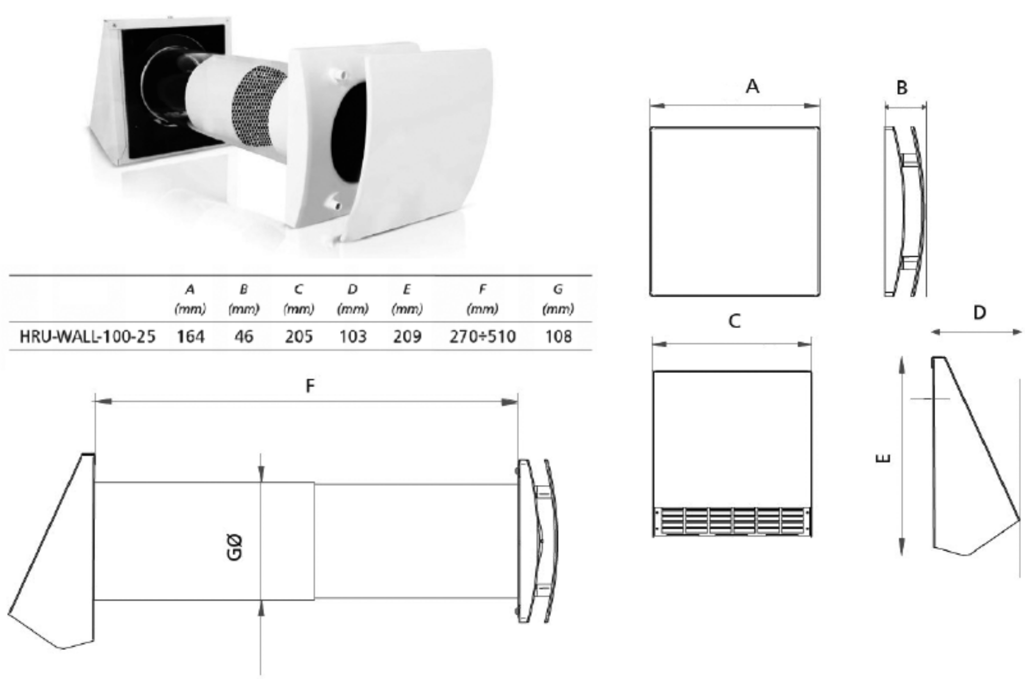
The one-pipe ventilation system is relatively unknown and poorly supported by research. Most of the information comes from catalogs and brochures from manufacturers. In the literature, experimental and theoretical study of the heat exchanger with a periodic change in the flow direction for room ventilation [7], and stochastic analysis of compact plate-fin recuperators for microturbines [8] can be found. Research relating directly single room ventilation units with recuperative or regenerative heat recovery was conducted by Manz et al. [9]. More research results can be found directly on DV in the field of its energy analysis, for example [2, 10].

The aim of this study was experimental investigation of the work of the compact recuperator in real conditions. The device's efficiency was determined based on measurements results and compared with manufacture's data.

2. OVERVIEW OF THE TESTED RECUPERATOR

The compact recuperator offered by Alnor was tested (Fig. 1). A unit named HRU-WALL-100-25 with the amount of air flow of 25 m^3/h was purchased in December 2018 for the price of PLN 1160. It is the smallest of the two types of units offered by the manufacturer. The exact dimensions of this device were given in Fig. 1.

As stated by the manufacturer, HRU-WALL is a single room decentralized heat recovery unit for concealed installation. The unit comes with a ceramic heat exchanger which boasts a maximum heat recovery ratio of 90% (the nominal heat recovery ratio is 74.3% as per standards [12]). The energy-efficient EC fan changes the running direction every 70 sec-



	A (mm)	B (mm)	C (mm)	D (mm)	E (mm)	F (mm)	G (mm)
HRU-WALL-100-25	164	46	205	103	209	270÷510	108

Figure 1.
HRU-WALL-100-25 compact recuperator and its dimensions in millimeters [11]

onds to alternate between air supply and air exhaust. Low energy consumption and extremely low operating noise make this heat recovery unit a recommended solution for non-stop operation. The heat recovery unit has three speed levels to choose from, depending on the size of the area and indoor demands. This device operates in the outdoor temperature rangeing from -20 to 50°C. It weighs 2.4 kg and belongs to the “A” energy class [9].

3. MEASUREMENTS

Measurements of air temperature and airflow rate were carried out for the tested recuperator. The first tests took place in winter, at a low outdoor temperature in order to measure the air temperature in the individual parts of the device during its operation in heat recovery mode. Fig. 2 shows the test stand. It consisted of the recuperator mounted in the plate with thermal insulation placed in the window frame together with the switch; five thermocouples connected to the thermometer, which recorded the measurement data; and the micromanometer for measuring the pressure difference between the room and the external environment.

Prepared test stand enabled change of the pressure in the room by installing two small fans in the door with adjustable performance.

Mentioned thermocouples were arranged as shown in Fig. 3: in the outdoor environment; in the air intake; between the heat exchanger and the fan; behind the fan (air inlet to the room); and in the room.

Temperature measurements in the indicated 5 measurement points were taken for three fan speeds in two different conditions: (1) in real conditions with natural air flow; (2) for balanced pressure indoors and outdoors. The measured temperatures were used to calculate the actual efficiency of the tested recuperator.

Measurement of air velocity distribution flowing through the device was also carried out in order to determine the air volume flow rate. The test was made using a thermoanemometer. The use of other measurement methods was in this case impossible. Prepared test stand (Fig. 4) consisted of tested recuperator together with a switch, a cardboard box with an air intake, a thermoanemometer installed in the channel, a laptop with a software for recording measurements and a micromanometer. Small fans were also attached to the box. Their speed was regulated using the power supply with adjustable voltage.

Open-box reproduced conditions without differential

Figure 2.
Air temperature test stand

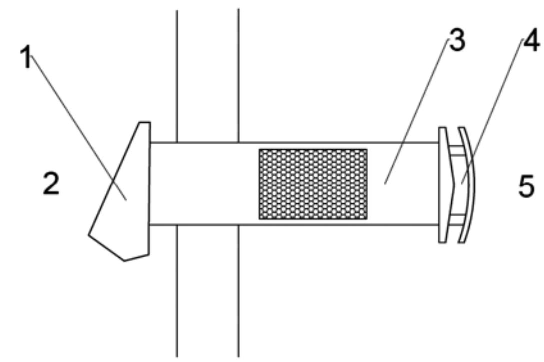


Figure 3.
Location of the thermocouples; 1) in the air intake, 2) in the outdoor environment, 3) between the heat exchanger and the fan, 4) behind the fan (air inlet to the room), 5) in the room

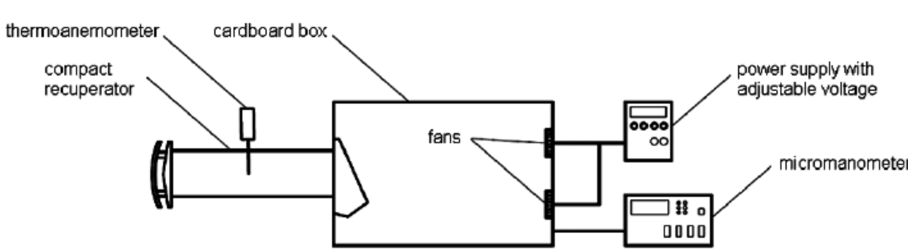


Figure 4.
Scheme of air velocity distribution test stand

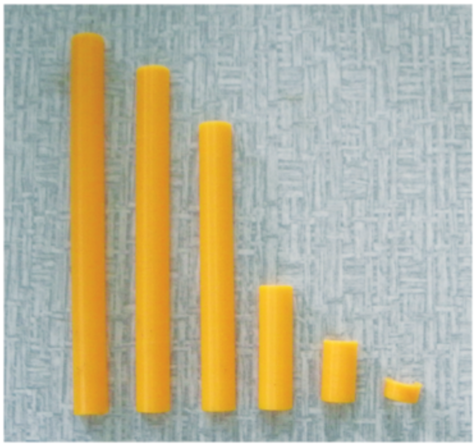


Figure 5.
Tube halves printed in a 3D printer

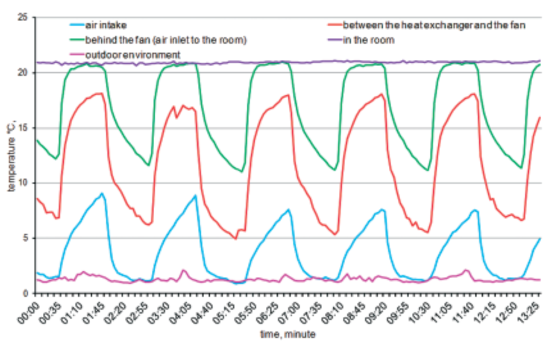


Figure 6.
Sample graph of air temperatures for the second fan speed at balanced pressures in the room and outside

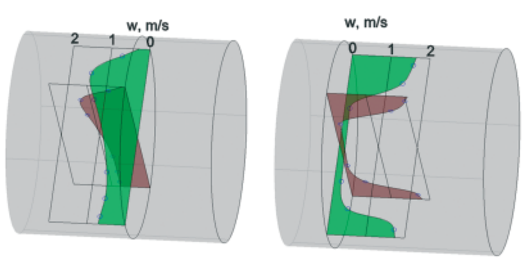


Figure 7.
Air velocity profile for supply mode (left) and exhaust mode (right) for third fan speed and 0 Pa of pressure difference

pressure on both sides of the device; closed and sealed – overpressure conditions on the outside. Mounted adjustable fans enabled the change of pressure inside, measured by the micromanometer.

The air velocity was measured in 6 points at two axes of the device channel, in accordance with the standard [13]. All fan speeds were considered, for supply and exhaust.

For correct and accurate placement of the thermoanemometer in the tube, the tube halves printed in a 3D printer were used (Fig. 5).

The measured air velocity values were averaged and the air flow volume rate for each fan speed for supply and exhaust was calculated. It was based on a product of averaged velocity and area of channel.

4. RESULTS AND DISCUSSION

The graph shown in Fig. 6 shows the distribution of measured air temperatures for the second fan speed when pressures in the room and outside were balanced. The indoor and outdoor temperatures were almost constant during all measurements. The remaining temperatures show 70 seconds of air supply and exhaust operation in heat recovery mode. It clearly shows five work cycles of the HRU-WALL-100–25 compact recuperator. There was also a noticeable difference between the indoor temperature and the temperature measured between the exchanger and the fan. Theoretically, the difference should be very small, because there were no elements that could decrease or increase the air temperature. This may be due to the fact that in addition to the heat exchanger, heat accumulation was also carried out on other elements of the device such as fan elements.

In order to obtain the efficiency of the device, the averaged air temperature in measured 5 points for 3 fan speeds was calculated, for the time step of 5 s. Table 1 and Table 2 show the calculated efficiency of the unit separately for supply and exhaust air. The exhaust efficiency means the amount of energy kept by the device in relation to the total energy-related to the temperature difference between the indoor and outdoor air, whereas supply efficiency, the amount of energy supplied to the air by the device in relation to the total energy necessary to warm the air to room temperature.

The efficiency was calculated from the ratio of heat supplied/received from the heat exchanger (Σ_{q1}) to the sum of that energy and energy needed for the air to reach room temperature (Σ_{q2}):

$$\eta = \frac{\Sigma_{q1}}{\Sigma_{q1} + \Sigma_{q2}}\ ,\% \qquad (1)$$

The values of air flow volume rate used for calculations (converted into mass flow rate) were measured values (Table 3÷5). It has been assumed that the average air flow volume rate is identical to the calcu-

Table 1.
Calculated supply efficiency for the tested working conditions

Working conditions	Supply efficiency
third fan speed at balanced pressures	64%
second fan speed at balanced pressures	68%
first fan speed at balanced pressures	79%
third fan speed at a natural pressure difference	56%
second fan speed at a natural pressure difference	47%

Table 2.
Calculated exhaust efficiency for the tested working conditions

Working conditions	Exhaust efficiency
third fan speed at balanced pressures	73%
second fan speed at balanced pressures	83%
first fan speed at balanced pressures	85%
third fan speed at a natural pressure difference	79%
second fan speed at a natural pressure difference	93%

Table 3.
Calculated averaged air flow volume rate for all fan speeds based on measurements for 0 Pa

Air supply	0 Pa
first fan speed	9 m^3/h
second fan speed	16 m^3/h
third fan speed	28 m^3/h
Air exhaust	0 Pa
first fan speed	8 m^3/h
second fan speed	14 m^3/h
third fan speed	29 m^3/h

Table 4.
Calculated averaged air flow volume rate for all fan speeds based on measurements for 4 Pa

Air supply	4 Pa
first fan speed	14 m^3/h
second fan speed	20 m^3/h
third fan speed	32 m^3/h
Air exhaust	4 Pa
first fan speed	2 m^3/h
second fan speed	12 m^3/h
third fan speed	27 m^3/h

Table 5.
Calculated averaged air flow volume rate for all fan speeds based on measurements for 7 Pa

Air supply	7 Pa
first fan speed	19 m^3/h
second fan speed	22 m^3/h
third fan speed	36 m^3/h
Air exhaust	7 Pa
first fan speed	-8 m^3/h
second fan speed	6 m^3/h
third fan speed	25 m^3/h

lated average value from the measurements, and the ratio of the amount of supply air to exhaust air was adopted so that the amounts of energy delivered to the device and consumed were the same. The calculations were made in Microsoft Excel with the use of the Solver tool.

Air flow volume rate, according to manufacturer data, should be as follows: 25 m^3/h for the third fan speed, 18 m^3/h for the second fan speed and 10 m^3/h for the first fan speed. Thus, the test confirmed the accuracy of the data provided by the manufacturer. Results differed by a maximum of 5 m^3/h for atmospheric pressure. Moreover, the device was characterized by unevenness of the air flow in relation to the exhaust and higher efficiency of exhaust than supply mode. Therefore, the air velocity profiles for supply and exhaust mode were different, as shown in Fig. 7.

On the air velocity profile for exhaust mode, the air was blown towards the channel walls. At the walls, the air velocity value was the highest. A small amount of air flowed through the channel. The centrifugal force produced by the fan threw air at the sides of the channel. Air velocity distribution was more even for the air supply, however, slightly more air flowed over the upper part of the channel, probably due to the fact that the intake has an inlet at the bottom. Air velocity values were lower than in the case of exhaust mode.

5. CONCLUSIONS

1. The test allowed to calculate the actual efficiency of the device and showed some of its deficiencies.
2. Efficiency given by manufacturers did not differ significantly from those measured and calculated. However, it was higher for operation in the exhaust air mode. This is due to the fact that the exhaust air flows had lower values, and thus the air flowed with lower air velocity. As a result, exhaust air better contact with the surface of the heat exchanger. Moreover, air temperature for exhaust mode changes more than in supply mode, because the air volume is lower, however, the same amount of energy is exchanged.
3. Air flow measurements showed uneven flow efficiency for air supply in relation to exhaust. In real working conditions, the recuperator supplies more air than it removes.
4. In general, the equipment was rated very well. It fulfilled its role of ventilation with heat recovery. In the room in which it was temporarily installed, the improvement in air quality was observed.

5. This device is new and will certainly undergo many modifications to improve its work. One suggestion could be the use of flow guides to more accurately guide the airflow path so that it evenly covers the entire heat exchanger. The use of this solution would probably improve the heat recovery performance of the device.
6. The correctness of operation and the possibility of improving this device will be the subject of many subsequent research works due to the curiosity of engineers.

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