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Heat recovery efficiency of local decentralized ventilation devices

J. Zemitis*, R. Bogdanovics
Riga Technical University, Riga, Latvia,
** E-mail: jurgis.zemitis@rtu.lv*

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Abstract. Decentralized, room-based ventilation systems have become increasingly popular in the Baltic countries. Such systems are easy to install and, according to technical information, ensure high heat energy recovery potential for new and renovated buildings. The specified heat recovery efficiency is used for building energy simulations and to calculate the necessary heating energy that is needed to warm up the supply air. However, this value is stated at non-existent pressure difference between indoor of the building and the outside. In real-case situations, there is always some pressure difference due to wind and stack effect. In this study, a ventilation device is tested in a laboratory environment at different simulated outside air temperatures and pressure differences. The simulations are conducted in a climatic chamber where the air temperature and pressure differences can be set. The temperature is adjusted using a cooling device but the pressure difference with an exterior fan device. Different combinations of simulated outside air temperatures and pressure differences were tested. The results suggest that the heat recovery efficiency is highly dependent on the pressure difference and it rapidly decreases with the rise in pressure difference. If the pressure difference is in the range of 10–20 Pa, the heat recovery efficiency will be only between 20 and 50 %, while the stated value in the technical data sheet is 85 %. Even at a pressure difference of 0 Pa, the average heat recovery efficiency is 73 %, and only for the first few seconds of the supply cycle, the efficiency reaches 85 %. This can influence the calculated building energy efficiency class, as well as lead to undersized heating system elements.

1. Introduction

Nowadays almost all newly built buildings are equipped with some sort of mechanical ventilation system, whether it would be hybrid type, full supply/exhaust system or local, personalized ventilation system [1–3]. Such systems can also be applied to historical buildings [4]. This is done to increase indoor air quality (IAQ) and, at the same time, to maximally reduce the heat losses that occur through the natural ventilation system. For a moderate climate, as in Europe, combined infiltration and ventilation is responsible for approximately 50 % of the total heat losses in well-insulated buildings [5]. Similar results are presented for buildings located in Russia, stating that for multi-apartment buildings, heat losses due to ventilation, including infiltration, exceed transmission losses and account for about 60 % of total heat losses [6]. To reduce this, mechanical ventilation with heat recovery is used. Such systems can also help to significantly save primary energy, at the same time ensuring good indoor air quality while the airtightness of new and renovated buildings is increasing [7–9]. Several different heat recovery types can be applied to ventilation systems [10], but especially in cases of nearly zero-energy buildings (nZEB), it is vitally important to take into consideration that the exhaust temperature after heat exchanger for fully mechanical supply/exhaust ventilation systems must be limited to 0° to +5 °C, depending on the heat recovery type, as for such buildings more often the aim is to have as high heat recovery efficiency as possible [9]. A study analysing indoor relative humidity in apartments with a room-based ventilation system with a rotary heat exchanger indicated that it is suitable for single-room ventilation of dry rooms, such as living rooms and bedrooms, while excessive moisture from kitchens and bathrooms provided a mould risk [11]. Other publications indicate that a specific type of decentralized ventilation system with two heat exchangers, which are hydronically connected in parallel, is an appropriate solution for use in hot and humid climates because more cooling and dehumidification capacity is available [12]. Other studies have focused on developing modern ventilation solutions that are integrated into the façade and work on convective heat flow without the use of a mechanical ventilator [13].

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The IAQ and adequate ventilation are linked to human wellbeing, health and even performance [14]. Studies on how to evaluate the performance and effectiveness of different ventilation system types have been performed for more than 20 years [15]. In some cases local recirculation diffusers, which in comparison to central recirculation ventilation systems eliminate such disadvantages as – increased power consumption for moving the ventilation air; increased size of the air ducts; recirculation of hazardous substances to the supply air with air transfer, can be applied to reduce the consumed heating energy [16]. These savings can be simulated and various scenarios analysed to find the best solution [17, 18]. However, it can often be hard to find the necessary funding and space to install full-size centralized ventilation systems with large ducts and AHU. Therefore, to save space and to reduce the amount of ductwork in buildings, quite often a room-based decentralized ventilation system is installed. It is especially desirable in case of renovations, as they require almost no indoor works and, according to existing studies, can provide good IAQ and thermal comfort [19]. There are various designs and types of such systems. These can be either as a small AHU with a rotary heat exchanger, which, according to studies, can have thermal recovery efficiency up to 85 % [20] or like a single fan that changes directions and leads air through the regenerative heat exchanger. At the same time, some research has indicated that the actual performance of ventilation systems on-site is often lower than expected [21]. Also, noise can be a negative factor, when operating room-based ventilation devices, as the ventilator and the user are in the same room. Additionally, in most cases, there are no sound dampening devices installed. Although, according to the manufacturer data, the sound pressure level of such devices is relatively low – from around 30 dB to 43 dB at 1m distance, in real case scenarios this can cause a nuisance. This can be linked by changes in the sound profile when the ventilator stops after each cycle and starts again in a reversed direction.

Although decentralized room-based ventilation systems with alternating airflow and storage type heat exchanger can provide good opportunities for heat energy savings and low energy consumption, their actual performance can be strongly influenced by external parameters like wind and stack effect, therefore noticeably lowering the actual heat recovery efficiency and overall building energy class [22]. The rated heat recovery energy efficiency is the only value stated in the datasheets of such units, but it is measured under specific conditions according to the standard EN 13141–8:2014 [23], which describes the measuring procedure for the alternating units. The standard does not require to perform measurements at various pressure differences between indoors and outside but only states that indoor temperature must be +20 °C and outside temp. +7 °C. This rated heat recovery energy efficiency is often the only information available for the individuals or software performing energy calculations it can be misleading as usually it is taken as a constant value. Therefore it can influence the heat loss calculation results that take into consideration the infiltration rate and supply air temperature [24].

In some cases, the negative or positive pressure is required by the local regulations to prevent pollution coming from the outside or to prevent exfiltration of moist indoor air into the structures [25]. Even if it is not designed specifically, the pressure difference between indoors and outdoors on average is approximately 7 Pa for naturally ventilated buildings, but this can vary widely, depending on the specific building form, location, orientation of openings and temperature difference [26]. Existing measurements in apartments with room-based air handling units with regenerative ceramic heat exchangers have shown insufficient results because they fail to guarantee a continuous air change in apartments and do not comply with energy efficiency requirements [25]. Such shortcomings are noticed also for heat recovery ventilators, as the measurements indicate that the heat exchange efficiency is not constant and changes depending on the actual operating conditions [27]. Another study suggests that external short circuits, internal air leakages, and heat flows through the casing can also reduce the performance of single-room ventilation units [28].

In previous studies, the changes in ventilation air volume that gets supplied/exhaust from building through decentralized ventilation devices at various pressure differences have been analysed. The results suggest that at a differential pressure of 10 Pa, the deviations in fresh air amount can vary between 30 to 100 % from the nominal flow of 30 m^3/h flow. This means that the supply rates were higher and the extract rates were lower than the nominal flow rates, and on cold winter days, this could lead to drought and also to decreased heat recovery efficiency [29]. In the same study, it was mentioned that the specific heat recovery efficiency is lower than stated in the datasheets, but as the measurements were field-based, the exact values may be inaccurate and case dependent. Therefore, in this study, a specific ventilation device is tested in a laboratory environment at different simulated outside air temperatures and pressure differences to find out how heat recovery efficiency changes.

2. Methods

The study was conducted in a closed and controlled environment of a climatic chamber, simulating the variable outside air conditions. The climatic chamber has dimensions of 3 by 4 meters and a ceiling height of 2.3 m, thus the total volume of the chamber is 27.6 m^3. The climatic chamber was tightly sealed, therefore making it almost perfectly airtight. In one of the chamber walls, the local decentralized ventilation system device was installed (see Figure 1 and 2). It supplied air to an open room, in such a way simulating indoor conditions. The conditions in this room were not specifically maintained but were stable by themselves as the air, supplied from the climatic chamber, did not have an influence on the average temperature, according to the measurements. The technical parameters of the test ventilation device were as follows: max airflow 25 m^3/h, electrical power consumption 13 W, specified heat recovery efficiency up to 85 %. The device operated in two 70 seconds long, variating cycles, switching between supply and exhaust.

Figure 1. The experimental setup – air intake from the simulated cold outside conditions through integrated wind protection frame (on the left); air supply side in the warm room (in the middle); ventilator, which generates the simulated overpressure in the climatic chamber (on the right).

Figure 2. The cooling device, which ensured simulated outdoor air temperature in the climatic chamber.

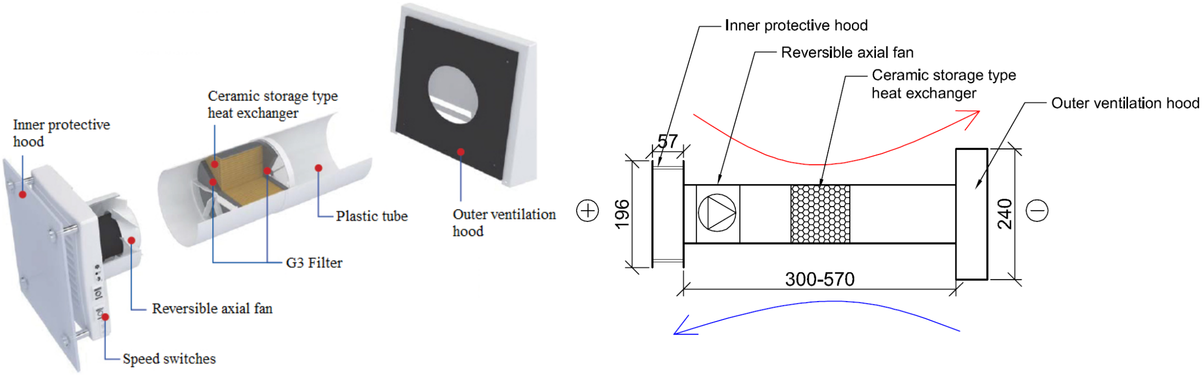


Figure 3. The ventilation device used in experiments – the construction of the device (on the left) and the section of equipment (on the right).

The inside of the chamber represents outside air conditions, as during the experiment the temperature of the chamber was controlled, and measurements were performed at a temperature range from +7 °C down to –5 °C. The relative humidity during the experiment in the climatic chamber varied between 30 to 60 %, while in the external premise it was constantly around 40 %. However, this should not affect the energy efficiency of the heat recovery unit as it is not an enthalpy type heat recovery unit and it only recovers sensible heat from exhaust air. Additionally, the inside air pressure of the chamber was controlled through the help of a Retrotech ventilator, which constantly adjusts the rotation speed to keep the specified pressure difference between the inside of the chamber and external premise. The ambient and climatic chamber temperatures were logged through the Extech SD800 logger with an accuracy of 0.8 °C. The temperature just in the front and back of the ventilation device was measured and logged with 1 second interval with Testo 435 device equipped with thermal velocity probe with temperature measuring range –20 to +70 °C ±0.3 °C and velocity measuring range of 0 to 20 m/s ±(0.03 m/s + 4 % of mv) and waterproof immersion probe, which has temperature measuring range –60 °C to +400 °C and accuracy class 2. The measurements of temperature were performed in front of the ventilation device, located in the climatic chamber, herewith regarded as inlet temperature (Θ_{12}), after the device as supply temperature (Θ_{22}), the temperature in the climatic chamber is regarded as outside temperature (Θ_{21}) and temperature in the open room as indoor temperature (Θ_{11}). See the full experimental setup in Figure 4.

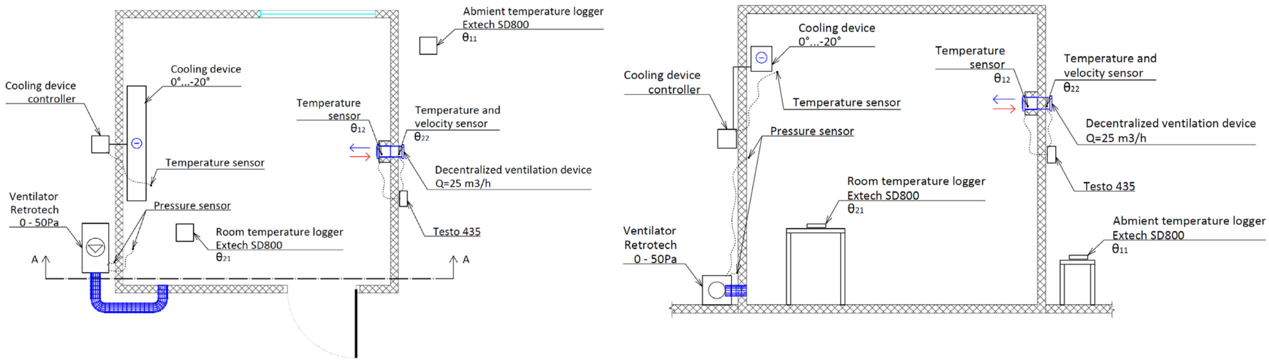


Figure 4. The plan of the experimental setup (on the left) and the section view A-A (on the right).

During the experiment, several sets of measurements were performed. For each set the temperature in the climatic chamber was regulated to be at +5, 0 or –5 °C while the air pressure difference between climatic chamber and external premise was changed to be 0, 5, 10, 15, 20, 30, 40 or 50 Pa, respectively, therefore representing a wide range of external simulated air velocities. Each of the measurements was run for at least 7 min, thus ensuring 3 full cycles of supply/exhaust. For a single measurement with set parameters of +2 °C in the climatic chamber and 0 Pa pressure difference, the experiment was run for 2 hours.

The heat recovery efficiency was calculated in accordance with the equation presented in the standard EN 13141-8:2014 [23]. It was calculated for each measuring time point (every second) and afterward, the average value for the supply working regime time (around 70 seconds) was determined according to the following equation:

$$\varphi = \frac{1}{t_{cycle}} \left[\int_t \left(\frac{\theta_{22} - \theta_{21}}{\theta_{11} - \theta_{21}} \right) dt \right] \cdot 100\ [\%], \tag{1}$$

where t_{cycle} is time of an operating cycle (s);

Θ_{22} is supply temperature (°C);

Θ_{21} is outside temperature (°C);

Θ_{11} is indoor temperature (°C).

In addition, the electric running power of specific ventilation device at various pressure differences ranging from 0 Pa to 113 Pa was measured with Energy Check 3000, which has a measuring range from 0 to 3000 W and accuracy of ±1 %.

3. *Results and Discussions*

In Figure 5, the results of the middle part of a long-term experiment are presented. The graph represents the data obtained from the 33rd minute to 42nd minute, as during this period it can be considered that the system is stabilized, and the results represent the actual system working regime. The outside and inside temperatures were stable, around +2 °C and +19.4 °C, respectively. At the same time, the supply temperature decreased from max 16.6 °C to 11.7 °C in each supply cycle. This represents the stage when the ventilation device is working in the supply regime, after which the ventilator changes the spinning direction and exhaust stage starts. In this stage, the temperature on the room side of the device increases and the heat recovery unit regenerates, as the warm room air gets sucked out. The graph also shows that the supply temperature during the supply working regime decreased linearly, at the same time the temperature on the cool side of the ventilation device decreased rapidly and almost instantly reached the simulated outside air temperature. On the other hand, during the exhaust phase, the temperature on the room side of the ventilation device increased logarithmically while the temperature after the device raised linearly.

During the experiment, it was found out that the actual length of the cycle was not exactly 70 seconds but a little bit shorter, around 68.5 seconds. Although this is not of high importance for end-users, it must be considered when calculating heat recovery energy efficiency, as for each second it is necessary to manually state if the device is in supply or exhaust operating mode, or the results would be compromised. The calculated average temperature efficiency for supply was 72 %, which is lower than the one stated in the technical datasheet. On the other hand, approximately for the first 6 seconds of each supply cycle, the efficiency actually reached exactly 85 %, which is the value mentioned in the technical datasheet. Therefore, it can be concluded that, although it is technically correct to state such value, it does not represent the actual situation and can cause under-sizing of heating systems and inaccuracy in calculated building energy efficiency certificate.

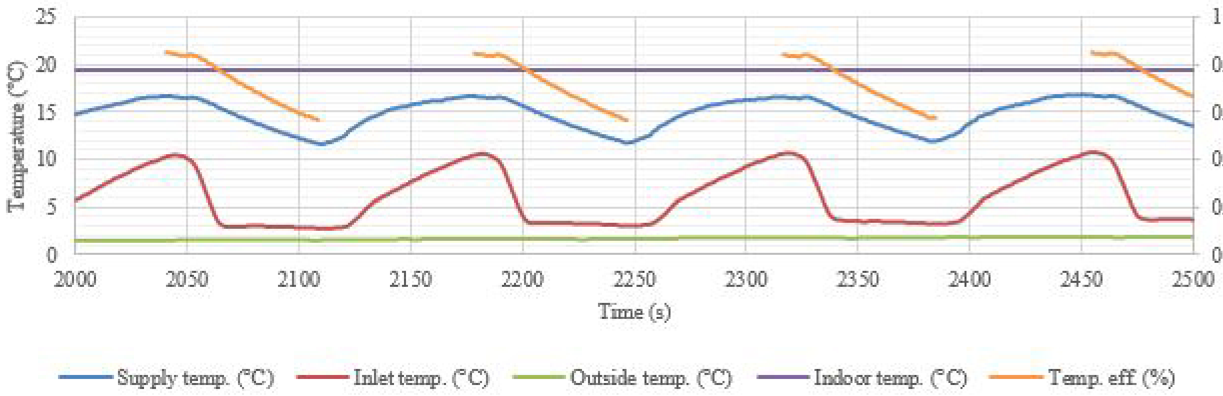


Figure 5. Variations in supply/exhaust temp. at +2 °C and $\Delta 0$ Pa.

Figure 6 to Figure 8 shows the results of short-term measurements at simulated outside air temperature of +0 °C and various pressure differences. As can be seen from the results, the pressure difference has a high impact on the supply temperature after the ventilation device. By increasing the pressure in the climatic chamber, which simulates the variable wind conditions of real case scenarios, the supply air temperature and, in retrospect, the heat recovery efficiency rapidly decreases. The graphs show that the maximal supply temperature after the ventilation device falls from +16 °C to +3 °C, depending on the air pressure difference, if the room temperature is +17.6 °C and outside air temp. +0 °C.

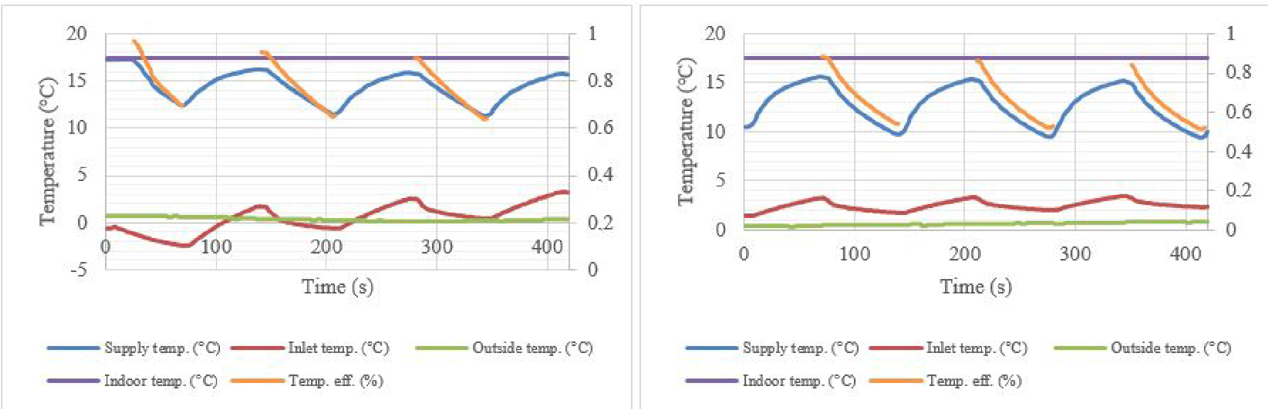


Figure 6. Short term measurements of supply/exhaust temp. variations at +0 °C outside air temp. and $\Delta 0$ Pa (left side) or $\Delta 5$ Pa (right side).

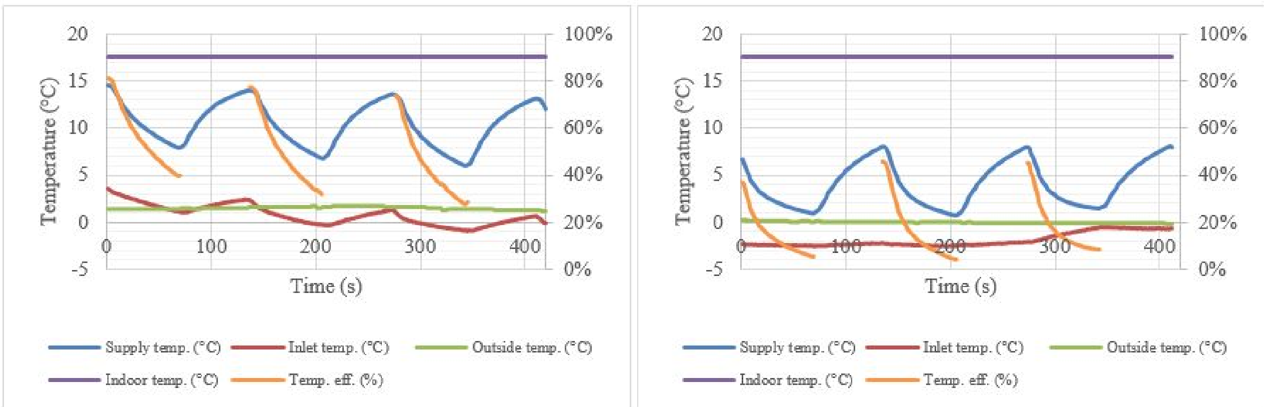


Figure 7. Short term measurements of supply/exhaust temp. variations at +0 °C outside air temp. and $\Delta 10$ Pa (left side) or $\Delta 20$ Pa (right side).

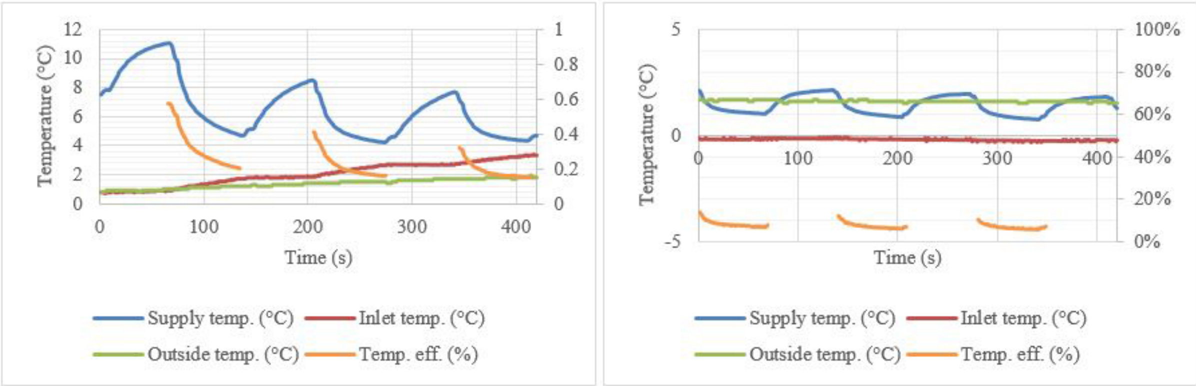
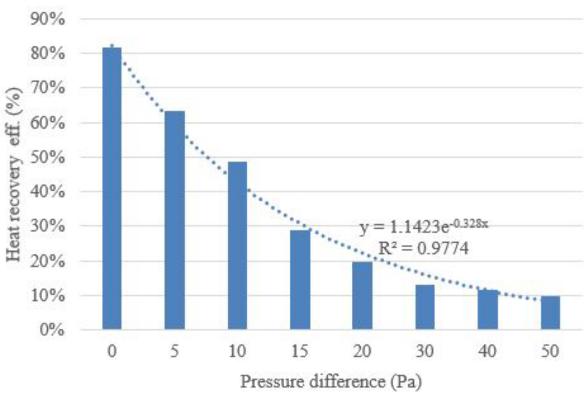


Figure 8. Short term measurements of supply/exhaust temp. variations at +0 °C outside air temp. and Δ30 Pa (left side) or Δ50 Pa (right side).

To obtain a more general overview of the results on how the pressure difference influences the heat recovery efficiency, they are combined in a single table/figure (see Table 1). As it shows, the average heat recovery efficiency decreases with the increase of pressure difference, independently of outside air temperature. The decrease follows an exponential trendline with high precision ($R2$ = 0.98). At $\Delta 50$ Pa the heat recovery efficiency is only 10 %. Although this is according to the Russian building code SP 50.13330.2012, which states that the minimum efficiency of recuperator should be at least 10 %, this is a very low value and causes high energy consumption for air heating.

Table 1. Average heat recovery efficiency at various outside air temperatures and pressure difference combinations

Pressure difference; ΔPa	Outside temperature; °C			Average heat recovery eff.; %
	+7	+2	–2	
0	86 %	78 %	81%	82 %
5	63 %	66 %	61%	63 %
10	45 %	50 %	51%	49 %
15	23 %	30 %	33%	29 %
20	26 %	16 %	17%	20 %
30	7 %	20 %	13%	13 %
40	14 %	9 %	N/A	12 %
50	12 %	8 %	N/A	10 %



The results on how the pressure difference influences the air velocity on the supply side of the ventilation device are shown in Figure 9. The results show that with increased pressure difference the air velocity after ventilation device exponentially increases. This means that an increased amount of air is passing through the ventilation device as the size of the device does not change. This corresponds to the data representing the fall of heat energy recovery efficiency as the device is not capable to heat up all the air passing through. The results show that for the specific ventilation device, an average air velocity almost doubles if the pressure difference rises from 0 to 10 Pa. However, this does not necessarily mean that the double amount of air volume is passing through. To make such a conclusion, more specific measurements should be performed, but it can still serve as a general indicator of change in magnitude.

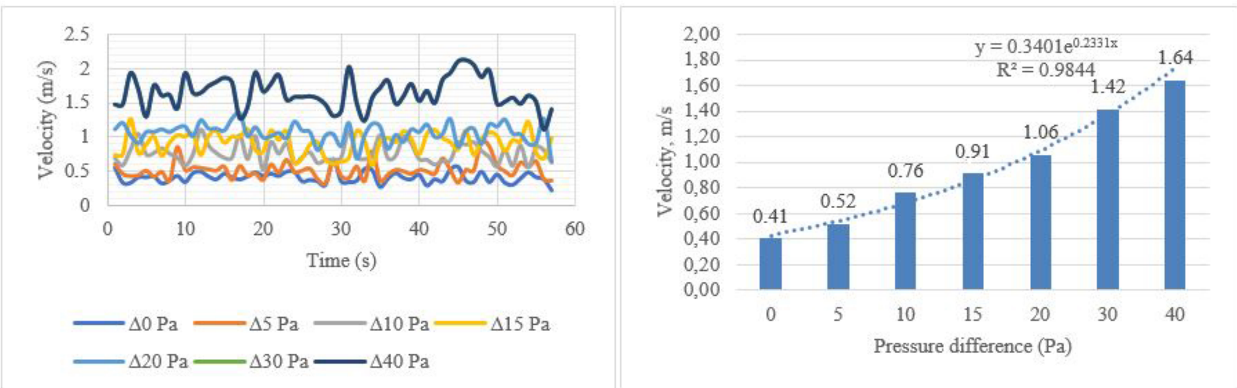


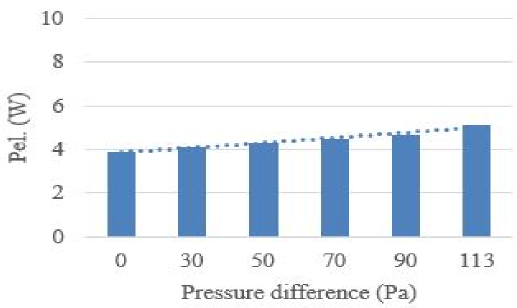
Figure 9. Velocity profiles (left side) and an average velocity depending on the pressure difference (right side).

The obtained results back up data from studies carried out previously from different studies. For example, the study [29] stated that at 10 Pa pressure difference the air volumes can vary from 50 % up to 100 % in comparison to no pressure difference. Similar data were obtained in our study, as can be seen from Figure 9, the average velocity at Δ10 Pa was 0.76 m/s compared to 0.41 m/s at Δ0 Pa. This would correspond to an increase in the volume flow of 85 %. Also, the obtained data are very similar to recent research [30]. The research shows that the heat recovery efficiency noticeably decreases with the rise of pressure difference and is only 25 % at Δ20 Pa, while at Δ5 Pa it can reach 70 %. These data are in close correspondence to the ones obtained in our study, see (see Table 1). Also, according to existing studies [31] such devices produce high noise levels and therefore are often operated at lower speed levels or switched off by occupants. This causes an increase in the difference between supply and exhaust airflow volumes and to the point that at Δ8 Pa the supply airflow is 2 times higher than the exhaust airflow. Therefore, the heat recovery efficiency is reduced.

Table 2 shows how the electric power of a specific ventilation device changed depending on the pressure difference. As the results show, the initial power is lower than specified by the manufacturer – 3.9 W compared to 6 W. As it was expected, the electric power increased when additional pressure was introduced, but it reached only 5.1 W even at 113 Pa pressure difference between the climatic chamber and the open room. This means that the actual electric consumption for running such a device will be lower if calculated theoretically according to the technical datasheet.

Table 2. Electric power of specific ventilation device at various pressure differences.

Pressure difference; ΔPa	0	30	50	70	90	113
Electric power; W	3.9	4.1	4.3	4.5	4.7	5.1



4. Conclusions

In this study, the heat recovery efficiency of a decentralized ventilation device was measured and analysed for various pressure differences between the inlet and supply-side at a controlled environment of the climatic chamber. The results suggest that for a specific decentralized supply/exhaust ventilation device with an integrated heat recovery unit the average heat recovery efficiency for the supply cycle at 0 Pa pressure difference was 73 %. For the first few seconds of the supply cycle, the efficiency reached 85 %, which was the number specified in the datasheet, but afterward, it decreased and at the end of the cycle was only 57 %.

Data analysis suggests that the heat recovery efficiency was strictly related to the air pressure difference between the inlet and supply side of the ventilation device. The average temperature efficiency at Δ5 Pa was 63 % while at Δ10 Pa it fell to 49 % and at Δ15 Pa was only 29 %. The efficiency kept falling with a further increase in pressure and followed the equation of $y = 1.1415e - 0.328x$. This means that for actual buildings in real case scenarios, where the pressure difference is constantly higher than 0 Pa, the energy consumption for building heating will be higher than calculated if the specified heat recovery efficiency will be taken into account. The actual pressure difference will vary depending on specific building size, shape and location but, according to other studies, for the most part of the year is in the range of 10–20 Pa, thus meaning that the heat recovery efficiency will be between 20 and 50 %. This can influence the calculated building energy efficiency class, as well as lead to undersized heating system elements and, if the room temperature will drop due to the above, the occupants will try to seal off the ventilation devices, therefore, compromising the indoor air quality.

The measurements suggest that the air velocity also was affected by the increased pressure difference although the device was equipped with a special protection plate. The results suggest that the supply air velocity on the inside of the device increased more than two times (from 0.41 to 1.06 m/s) if the pressure difference increased from 0 Pa to 20 Pa. This can also lead to a decrease in thermal comfort for occupants if the supply air is too cold, causing a sensation of draught and therefore leading to blocking or shutting off the ventilation devices.

The study suggests that the electrical power of the device was very low (3.9 to 5.1 W) even if the pressure difference raised from 0 to 113 Pa. This was lower than specified in the technical datasheet.

For future experiments, it would be necessary to study the heat recovery efficiency for similar types of devices but from different manufacturers and with various sizes. It could be expected that larger size units could have better results at higher pressure differences, as they have a larger heat recovery unit. Also, ventilation devices with an enthalpy type heat recovery unit should be tested to see if they can achieve higher efficiency and how they would perform in real case scenarios. Additionally, the thermal comfort rating ensured by such ventilation types should be tested to see how the relatively cold air during the windy outside conditions relates to it. Another thing to test could be how the changes in cycle working lengths would affect both the heat recovery efficiency as well as IAQ. At the moment most of the manufacturers have chosen 70 seconds to be the length of each cycle but maybe by making it a bit longer, the efficiency could be increased as the results showed that the temperature on the outside of the device did not reach room temperature during the exhaust phase. This could indicate that the heat recovery unit has still not been fully regenerated.

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Contacts

Jurgis Zemitis, jurgis.zemitis@rtu.lv
Raimonds Bogdanovics, Raimonds.Bogdanovics@rtu.lv