# ANALYSIS OF THE TEMPERATURE, HUMIDITY, AND TOTAL EFFICIENCY OF THE AIR HANDLING UNIT WITH A PERIODIC COUNTERFLOW HEAT EXCHANGER

by

***Marek JASZCZUR[a*], Marek BOROWSKI[b], Daniel SATOLA[c], Slawosz KLESZCZ[d], and Michal KARCH[b]***

[a]AGH University of Science and Technology, Faculty of Energy and Fuels, Cracow, Poland
[b]AGH University of Science and Technology, Faculty of Mining and Geoengineering, Cracow, Poland
[c]Norwegian University of Science and Technology, NTNU, Faculty of Architecture and Design, Trondheim, Norway
[d]Frapol Sp. z o.o., Cracow, Poland

Original scientific paper
https://doi.org/10.2298/TSCI19S4175J

*In this work, thermal, humidity and enthalpy recover efficiency of innovative energy recovery exchanger is presented. The system under analysis allows adjustment of the humidity recovery especially useful in the winter period and forefend energy use for an anti-froze system of energy exchanger. It is shown that the presented method can achieve the real value for humidity and thermal efficiency above 80% and 90%, respectively. Such high efficiency was possible to obtain because the proposed system does not require energy consuming anti-freeze systems. The presented system is able to work even in extremely adverse outdoor air conditions (–20 °C and humidity 100%).*

Key words: *counterflow heat exchanger, frost formation, heat transfer*

## Introduction

In 2010 the EU introduced the Energy Performance of Buildings Directive (EPBD) on the energy efficiency of buildings, which states that after 2019 all public buildings will have to fulfil the requirements for nearly zero-energy buildings (nZEB) [1]. The ventilation, heating, hot water and cooling systems of buildings are responsible for half of the final energy consumption in the construction sector. Follow the national requirements defining the nZEB, these systems need to be designed to meet the criteria for the maximum use of non-renewable primary energy for ventilation, heating, domestic hot water and cooling. With the growing share of energy consumption, heat recovery seems to be essential solutions able to save significant amounts of energy [2-4]. However, this required a new solution for ventilation able to maintain the desired indoor air quality [5, 6].

Ventilation, heating and air conditioning units are increasingly applicable in modern industry and for domestic purposes. Public buildings and private houses are often equipped with ventilation and air conditioning systems [7, 8]. The key part of such a system is a heat

* Corresponding author, e-mail: jaszczur@agh.edu.pl

exchanger which is responsible for efficient energy recovery. In most solutions, parallel-plate heat exchangers are used. This type of heat exchanger was analysed by Vera and Linan [9] to provide formulations for construction design. Authors developed a 2-D model to evaluate expressions able to describe parallel-plate heat exchangers. Detailed experimental and numerical analysis of the fluid-flow within the heat exchanger with elliptical tubes was performed in [10, 11]. For the parallel-plate channel flows DNS numerical simulation [12] provide very accurate results. Kragh *et al*. [13] analysed a new solution for counterflow heat exchange used in ventilation systems in low temperature areas. The performance of the heat exchanger was calculated based on the model and validated against experimental data. In another research, the influence of channel geometry on the performance of a counter flow heat exchanger was studied [14].

The most commonly used heat exchangers have a maximum efficiency of 90% [15]. However, this relatively high value is received under carefully selected laboratory conditions and may differ under real conditions, especially under low outside air temperatures [16]. With the growing share of heating loads and ventilation, the heat recovery seems to be the main solutions to reduce heat losses in well-isolated buildings [17]. In residential buildings, ventilation plays a substantial role in the total heat loss. Heat losses resulting from the ventilation can be in the range of 25-55%. For this reason, it is impossible to fulfil the regulations about an energy-efficient without well designed mechanical ventilation with the heat recovery. The catalogue sheets of heat exchangers usually delivery the temperature efficiency under optimum or standard conditions. However, the velocity of the air-flowing through the exchanger channels is an additional parameter, affecting the efficiency but this values it is not standardised. Because of that the companies usually based on their own appropriately adapt the air velocity to obtain as high as possible efficiency. During the winter months, the air temperature may fall below –5 °C and under such conditions, the danger of heat exchanger icing may occur and at a later stage of total freezing, appears. This can generate a significant decrease in system performance and is caused by water dropping out from the exhaust air and occurs at the contact of the heat exchanger surface with cold air drawn from the outside (air temperature decline below the dew point) [18]. Under the influence of negative temperatures, the condensate accumulated in the exchanger starts gradually freezing, causing a significant reduction of the heat recuperation as well as results in decreasing the effective air-flow surface thereby leading to a substantial increase in the flow resistance. In adverse situations, such an effect can lead to irreversible mechanical changes and system damage.

The most common methods used solution against the occurrence of frost on the heat exchanger under low temperate includes two options: temporary shutdown of the supply fan, bypassing of the heat exchanger, and the use of an electrical heater. While the first method of protection against the occurrence of frost is highly controversial (the deactivation of the ventilation) the second solutions for anti-freeze systems contribute to very high energy consumption, due to the need to heat the ventilation air to comfortable temperature. In such systems, the recovery efficiency of sensible, latent and total heat combined with possible recovery of moisture from the air without increasing the demand for energy is essential [19]. It should also be noticed that apart from the temperature, relative humidity is another main parameters affecting the level of human comfort. The recommended relative humidity level for indoor air is considered to be in the range of 30-60% [20]. When relative humidity of indoor air exceeding 60% this can lead to the condensation of moisture on the low-temperature surfaces. On the opposite extreme relative humidity below 30% create discomfort for humans (dry skin, dry eye syndrome, susceptibility to infection). In ventilation systems equipped with counterflow air-to-air exchangers, the heat exchange occurs only in the sensible form, which in winter (low moisture content) does not prevent the drying of the ventilated rooms.

The study aims to determine the temperature, humidity and total efficiency of the air handling unit with a new generation of high performance periodic counterflow air-to-air heat exchanger. The article presents the results of studies of the efficiency of recovery of sensible, latent and total heat for a periodic heat exchanger. The proposed in this paper heat exchanger significantly increases the energy efficiency of the ventilation system and can be implemented together with the recovery of moisture from the used air. The presented solution prevents the heat exchanger from frosting, and this is done without increasing primary energy demand. As a result, very high and constant seasonal efficiency of heat and humidity recovery, independent of outside air parameters, can be expected.

## Methodology

The periodic-flow heat exchanger unit consists of a standard counterflow exchanger and a set of opposing air dampers with a short opening/closing time, used to air-flow cyclically direction modification. With this solution, the heat exchanger is able the intake air to absorb the moisture from the cold side of the heat exchanger. The above also protects the exchanger against frosting. Each unit with this solution is equipped with four counter-rotating aluminium dampers controlled by actuators, figs. 1 and 2. Switching position allows changing

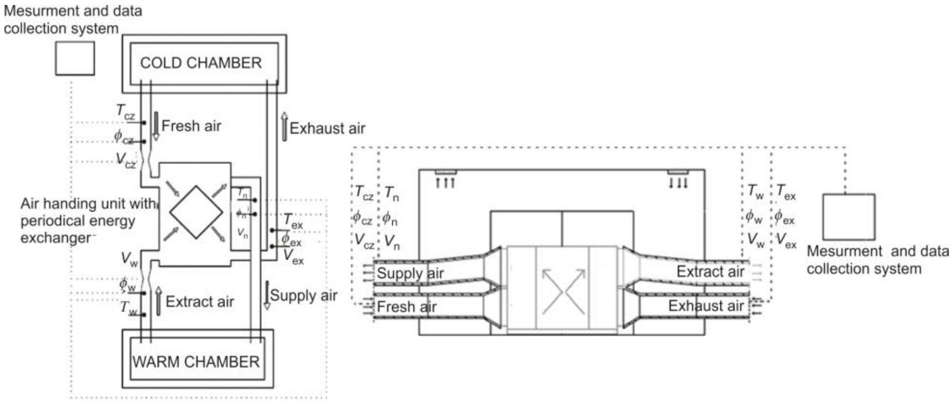


**Figure 1. Schematic diagram of the test section**

the direction of air on different part of the exchanger. The fourth tightness class according to EN-1751 prevent air mixing. The tests were done to determine the temperature, humidity and total efficiency of the air handling unit with a new generation of high performance periodic counterflow air-to-air heat exchanger with dampers. The experimental measurements were done under real conditions able to generate the possibility of occurrence of the phenomenon of heat exchanger frosting, tab. 1.

In the test chamber which simulated winter conditions, there was a connection of the outside and exhaust air to the unit with the heat exchanger, while in the warm chamber which simulated the conditions inside the ventilated rooms, there was a connection of the extract and supply air to the unit. A schematic diagram of the analysed system with the main unit is presented in fig.1. Measuring of air relative humidity ($\varphi_{cz}$, $\varphi_{ex}$, $\varphi_n$, $\varphi_w$) were performed by using Introl EE31 transducers with 1% data accuracy while the air temperatures ($T_{cz}$, $T_{ex}$, $T_n$, $T_w$) were measured by means Geneza GPE-D-A-160-Pt100-klA sensors with 0.1 K accuracy. Air-

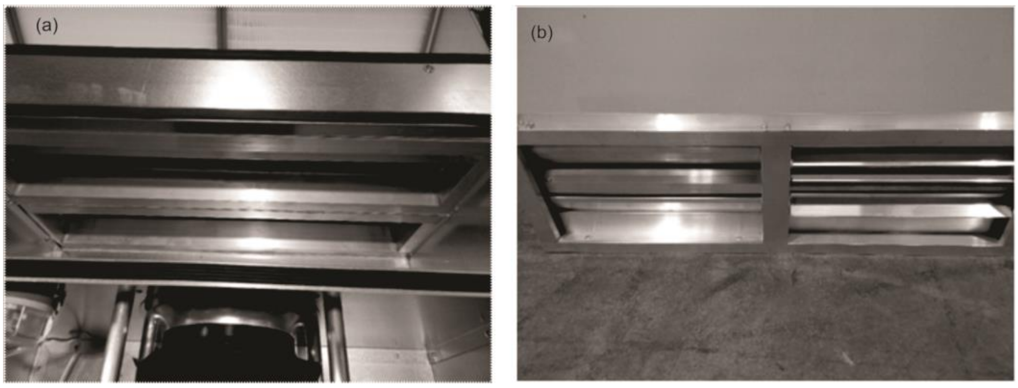

**Figure 2. Photo of the dampers on fresh air side (a) and whole counter-rotating damper (b)**

flow rates ($V_n$, $V_w$) were measured by using the Venturi tube in accordance with PN-EN ISO 5167-4:2005 standard. In order to record air thermal and fluid-flow parameters APAR AR207 data logger was used. All acquired thermal, and flow parameters were sampled with a time resolution equal to 5 seconds and recorded for a period equal to 5100 seconds. The temperature and humidity conditions of the outside and extract air were stabilized for the period about 1500 seconds.

**Table 1. Flow Conditions during the study**

| Parameter | $V_{cz}$ | $T_{cz}$ | $\varphi_{cz}$ | $V_w$ | $T_w$ | $\varphi_w$ |
|---|---|---|---|---|---|---|
| Unit | m$^3$/h | $^o$C | % | m$^3$/h | $^o$C | % |
| Value | 670 | –20 | 70 | 680 | 20 | 50 |

## Mathematical description

In order to perform an analysis of received measurements it was highly required to make additional calculations. It is known that relative humidity depends on temperature and the pressure. The same amount of water content results in lower relative humidity in warm air than cold air. To show the variation of mass water content in the air relative humidity was calculated to absolute moisture content.

One of the most important parameter for any heat exchanger is efficiency. In this paper due to the nature of presented unit, temperature, humidity, and total efficiency is presented.

### *Air moisture content*

The air moisture content is defined as a ratio of the mass of water vapour contained in the air to the mass of dry air. By transforming the gas state equations and using Dalton's law [21], the following equation can be used to determine the air moisture content by the value of the air temperature and relative humidity:

$$x = 0.622 \frac{\varphi p_{gs}}{p_b - \varphi p_{gs}} \tag{1}$$

where $x$ [gkg$^{-1}$] is the absolute moisture content, $\varphi$ [%] – thr relative humidity, $p_{gs}$ [hPa] – the water vapour pressure in saturation state:

$$p_{\mathrm{gs}} = 6.1121\mathrm{e}^{\frac{17.502T}{T+240.97}} \tag{2}$$

where $T$ [°C] is the temperature of air and $p_\mathrm{b}$ – the atmospheric air pressure (= 1013.25 hPa)

### *Air enthalpy*

Humid air enthalpy, $h$, with moisture content, $x$, is the enthalpy of a mixture of 1 kg dry air and $x$ kg water vapour. Assuming that for such a mixture of dry air and the total moisture content in the liquid form at 0 °C, the enthalpy equals zero; the following relationship is obtained:

$$h = c_{\mathrm{pg}}T + x\left(c''_{\mathrm{pp}}T + r_o\right) \tag{3}$$

where $h$ [Jkg$^{-1}$] is the humid air enthalpy, $T$ [°C] – the temperature of air, $x$ [kgkg$^{-1}$] – the absolute moisture content, $c_{\mathrm{pg}}$ – the specific heat of dry air (= 1005 J/kgK), $c''_{\mathrm{pp}}$ – the specific heat of water vapour (= 1965 J/kgK), and $r_\mathrm{o}$ – the heat of water evaporation (= 2500000 J/kg).

### *Temperature efficiency of the heat exchanger*

The temperature efficiency of a heat exchanger is calculated as the ratio of the heat flux recovered by the heat exchanger transferred from the extracted air to the supply air in reference to the total heating power required to heat the outside air to indoor air temperature. The efficiency of the recovery of sensible heat can be determined using the following equation:

$$\eta_\mathrm{t} = \frac{\left(T_\mathrm{n} - T_{\mathrm{cz}}\right)}{\left(T_\mathrm{w} - T_{\mathrm{cz}}\right)}100\% \tag{4}$$

### *Humidity efficiency of the heat exchanger*

The humidity efficiency of a heat exchanger can be defined as the ratio of the moisture flux recovered by the heat exchanger transferred from the extracted air to the supply air compared to the total moisture demand that would have to be met in order for the humidity of the outside air to be increased to the level of humidity of indoor air. The efficiency of the recovery of latent heat can be determined:

$$\eta_\mathrm{w} = \frac{\left(x_\mathrm{n} - x_{\mathrm{cz}}\right)}{\left(x_\mathrm{w} - x_{\mathrm{cz}}\right)}100\% \tag{5}$$

### *Total efficiency of the heat exchanger*

The total efficiency of the heat exchange can be defined as the ratio of the enthalpy flux recovered in the enthalpy exchanger system to the potentially recoverable heat flux:

$$\eta_\mathrm{c} = \frac{(h_\mathrm{n} - h_{\mathrm{cz}})}{\left(h_\mathrm{w} - h_{\mathrm{cz}}\right)}100\% \tag{6}$$

## Results and discussion

The first analysis was carried out based on the equipment manufactured on a large scale and using one of the widely used anti-frost systems (variable control of the air-flow depending on the exhaust temperature or on the value of pressure drop in the heat exchanger). In the catalogue sheet for the analysed unit and for the flow $V_{\mathrm{cz}} = V_\mathrm{w}$ = 1800 m$^3$/h and the air parameters: $T_{\mathrm{cz}}$ = –15 °C, $T_\mathrm{n}$ = 14.7 °C, $T_\mathrm{w}$ = 20 °C, the recuperation efficiency was around 90.5%. The exchanger surface was checked before the test start. The surface was clean, without any traces of damage and entirely dry, see fig. 3(a).

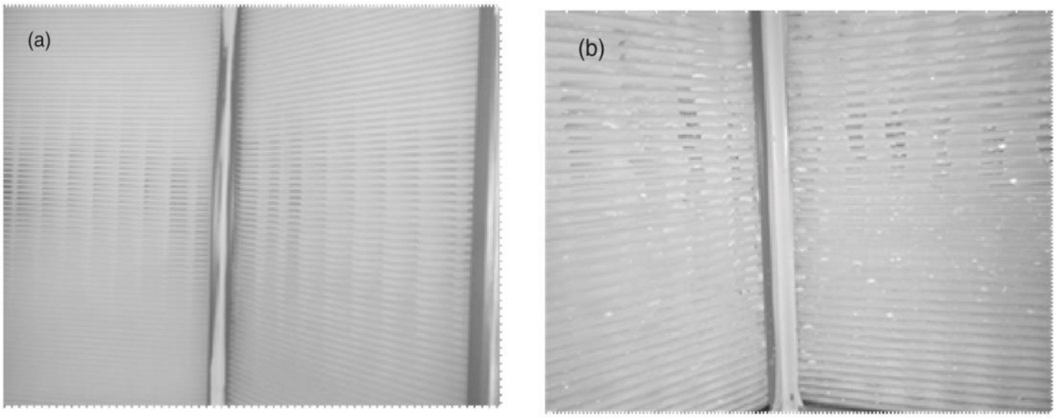


**Figure 3. The heat exchanger surface at the initial state (a) and the end of measurements (b)**

Figures 4-6 present results of measurements for the air-flow, temperature and humidity made on the supply and exhaust side during the test. The main unit control system was intended to prevent the exchanger icing through appropriate adjustment of the fans rotational speed. The equipment also featured an implemented second anti-icing system – an outside air bypass, which was deactivated during these tests. The co-operation of both systems should protect the exchanger, however, at the price of worsening the heat comfort and resulting in a substantial plummet in the system efficiency.

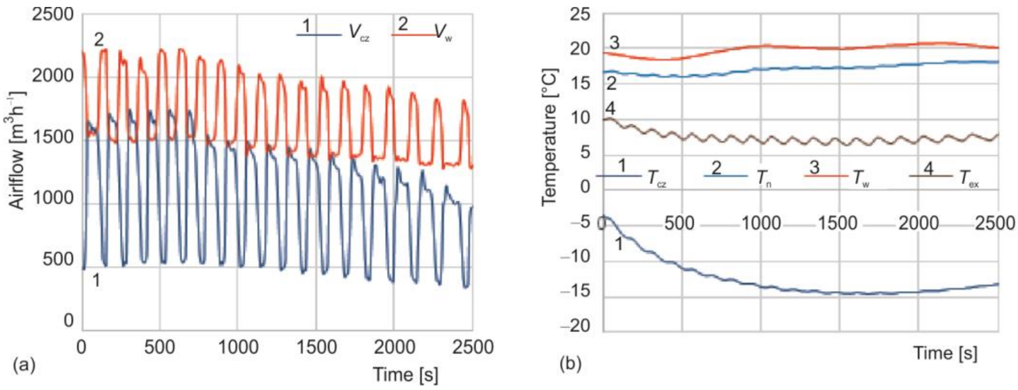


**Figure 4. The air-flow through the heat exchanger (a) and air temperature over time (b)**

Figure 4(a) shows changes in the air-flow rate, both on the supply and exhaust side. A gradual icing of the heat exchanger surface was observed during the test. This results in decreasing (with each cycle) the maximum value of the hot air-flow rate from 2220 $m^3$/h to 1780 $m^3$/h. Observed an increase in air-flow resistance is due to the freezing of the dropped out condensate. The measurement has been stopped when the mass of frozen condensate became a danger for the heat exchanger fins. It is clear that over time a new layer of ice will grow on the exchanger, making the further operation impossible.

Figure 4(b) presents the temperature over time for all air streams in the analysed unit. Average temperature efficiency based on measurements and eq. (4) was about 91.7% and was higher than that declared by the manufacturer. However, it should be noticed that calculated efficiency does not take into account phase transitions proceeding in the heat exchanger nor the fluctuation in the air-flow rates. When determining the total temperature efficiency, considering the air-flow, based on the measurement data new value of total temperature efficiency can be obtained equal $\eta_t$ = 66.8%. Moreover, calculating the ratio of outside air-flow rate to the extract air-flow rate, we obtain the value $V_{cz}/V_{ex}$ = 0.625. That shows a significant deficit of the air in the building and hence a considerable under pressure. During the operation under such parameters, the air in the building is supplemented by leaks in the building structure (drawing cold air from the environment).

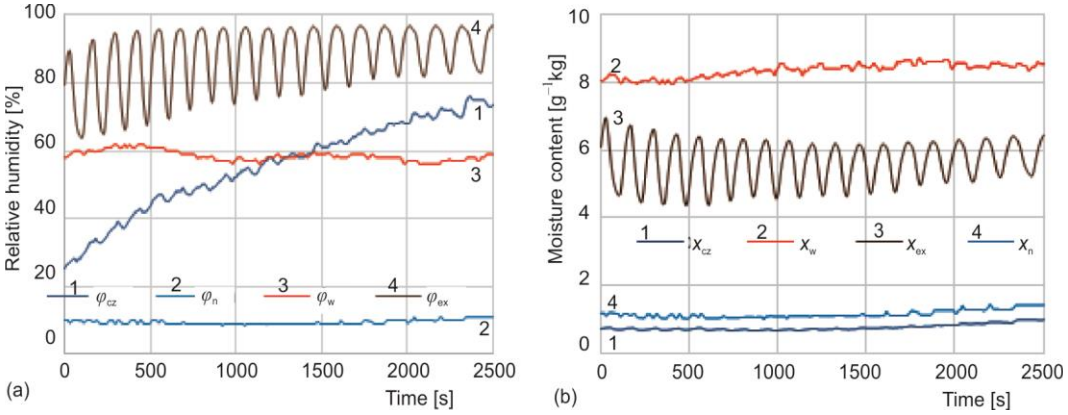


**Figure 5. Relative humidity (a) and water content (b) over time**

Figure 5(a) shows the value of relative humidity during the test. One may infer from this figure a change in humidity which results from a gradual decrease of the air temperature. However, the actual water content in the air did not change, see fig. 5(b) for reference. On the other hand, as shown in fig. 5(b) with the increasing flow rate of humid air exhausted from the room the condensation of water on the heat exchanger internal surface grows, and this phenomenon proceeds cyclically with output changes. The total efficiency calculated based on measurements and eq. (6) is very low $\eta_c$ = 42.7%.

To protect the heat exchanger surface against damage, the measurements were stopped when the amount of ice accumulated on the surface threatened its construction. Once the test was stopped, the exchanger surface condition was analysed. Figure 3(b) shows a heat exchanger surface when ice crystals are formed between the exchanger fins. Ice occupy a significant area of the heat exchanger, making the air-flow very difficult (an increase of the airflow resistance through the exchanger).

Figures 6 and 7 show the results of the flow rate measurements of supply air, $V_n$, and extract air, $V_w$, temperatures, $T$, and relative humidity, $\varphi$, of the air on each side of the heat exchanger. On the basis of experimental measurements, the time step of air damper position changes was set-up 300 seconds. Changing the position of the dampers results in the direction of airflow through the exchanger channels, enabling condensate evaporation from the heat exchanger walls in both operating modes. Evaporated condensate is then absorbed by the outside air stream. The results in fig. 6(a) shows that switching the air damper position resulting in a change in the direction of air-flow through the heat exchanger and the resistance of the flowing air on both sides of the heat exchanger is changed. This creates a decrease in the extract air-flow rate and an increase in the supply of air-flow rate by a constant value of about 60 m$^3$/h. This values equal to about 8% of the initial rates value. The total difference between the supply and extract air balance increases from the initial value of 5% up to 15%. As can be seen the temperature of the supply air presented in fig. 6(b) did not fall below the value of 18 °C during the entire period of the study, with the temperature values for the intake and extract air equal to –20 °C and 20 °C (+/–0.5 °C), respectively. The maximum and minimum temperature of the supply air was 18.9 °C and 18.2 °C, respectively, which results in the temperature efficiency in the range of $\eta_t$ = 95-97.5%. The maximum and minimum exhaust temperature was –6.2 °C and –9.1 °C, respectively. Despite critical air conditions, no frost has been observed on the heat exchanger surfaces. This was caused by the complete evaporation of the condensate as a result of periodic changes in the direction of air-flow.

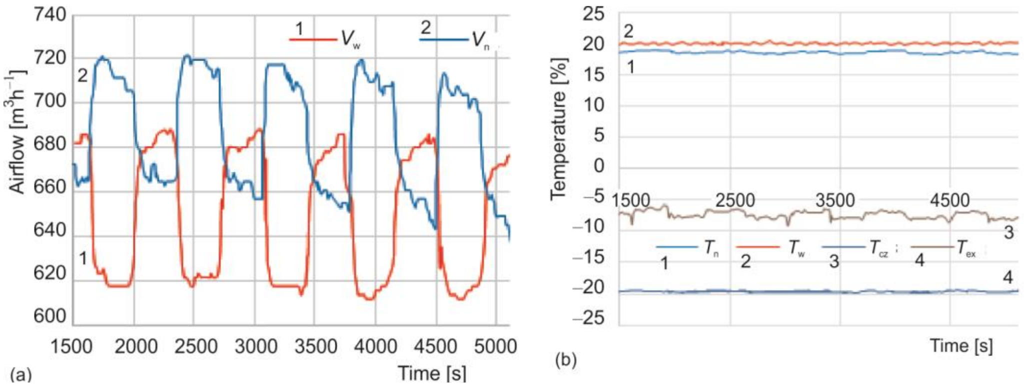

**Figure 6. Supply $V_n$, extract $V_w$ air volume flow rates (a) and outside $T_{cz}$, supply $T_n$, extract $T_w$, exhaust $T_{ex}$ air temperature (b)**

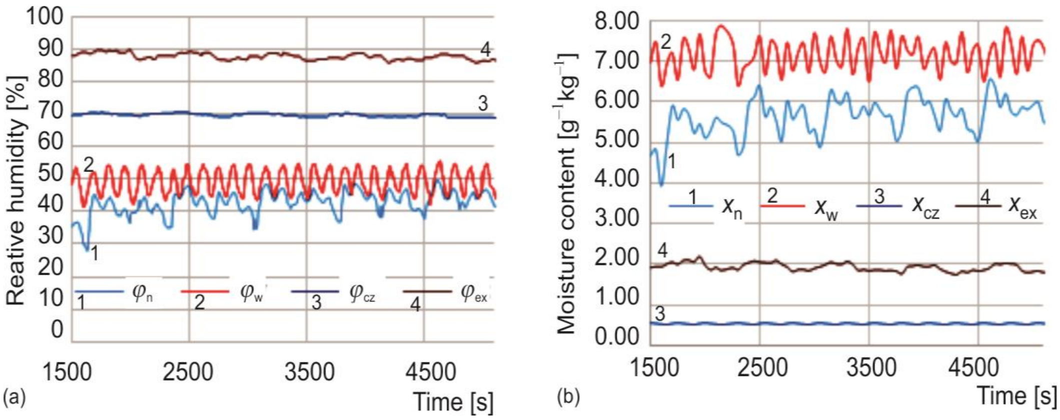

**Figure 7. Outside, supply, extract, exhaust relative humidity (a) and air moisture content (b)**

The humidity of the supply and extract air is presented in fig. 7(a) and during the measurements, it changed dynamically between 30-48% and 44-54%, respectively. The supplied air had the humidity level higher than 30%, which effectively prevented the drying of the ventilated rooms, contributing to maintaining the feeling of thermal comfort for the users of the building.

On the basis of the experimental measurements of temperature and relative humidity, the moisture content changes in each of the four air streams were evaluated. These values, which contribute to the heat and moisture exchange processes, are presented in fig. 7(b). For the first 150 seconds, the most intensive process of evaporation of the condensate accumulated on the heat exchanger plates is observed. During the following 150 seconds, the moisture content, $x_w$, in the supply air decreases gradually as a result of the complete evaporation of moisture accumulated on one side of the heat exchanger.

Based on the measurement data the average humidity efficiency is about 79.8%, while the average humidity capacity of the air handling unit with a periodic-flow heat exchanger is 2.96 kW. The detailed results for this case are presented in tab. 2-4. The average overall efficiency of the analysed in this work heat exchanger is 91.4%, which translates into a total capacity of 11.76 kW.

**Table 2. Average temperature capacity and efficiency of the heat exchanger**

| Parameter | $V_n$ | $V_w$ | $T_{cz}$ | $T_n$ | $T_w$ | $T_{ex}$ | $\eta_t$ | $Q_t$ |
|---|---|---|---|---|---|---|---|---|
| Unit | m$^3$/h | m$^3$/h | °C | °C | °C | °C | % | kW |
| Value | 685 | 651 | –19.8 | 18.6 | 20.1 | –7.3 | 96.24 | 8.80 |

**Table 3. Average humidity capacity and efficiency of the heat exchanger**

| Parameter | $V_n$ | $V_w$ | $x_{cz}$ | $x_n$ | $x_w$ | $x_{ex}$ | $\eta_w$ | $Q_u$ |
|---|---|---|---|---|---|---|---|---|
| Unit | m$^3$/h | m$^3$/h | g/kg | g/kg | g/kg | g/kg | % | kW |
| Value | 685 | 651 | 0.56 | 5.82 | 7.15 | 1.93 | 79.82 | 2.96 |

**Table 4. Average total capacity and efficiency of the heat exchanger**

| Parameter | $V_n$ | $V_w$ | $h_{cz}$ | $h_n$ | $h_w$ | $h_{ex}$ | $\eta_c$ | $Q_c$ |
|---|---|---|---|---|---|---|---|---|
| Unit | m$^3$/h | m$^3$/h | kJ/kg | kJ/kg | kJ/kg | kJ/kg | % | kW |
| Value | 685 | 651 | -18.5 | 33.5 | 38.4 | -2.51 | 91.4 | 11.76 |

## Conclusions

This article presents the results of measurements conducted to determine the temperature, humidity, and total efficiency of an air handling unit equipped with an innovative solution for the periodic parallel-plate counterflow heat exchanger.

Based on the carried out studies it is possible to state clearly that the traditional system does not work correctly at low temperature conditions and cannot be applied without an additional usually energy consuming protection method. Despite a relatively high temperature efficiency and a high temperature of the air supplied to premises, the recuperator did not fulfil its primary role, consisting of proper ventilation of the building. The deficit of outside air can result in significant growth of $CO_2$ in the room and also can lead to the drop in the air temperature inside the building. During 40 minutes of traditional heat exchanger operation, a significant part of the surface was covered with ice, and further operation of the unit could result in total flow blocking, and hence in permanent building ventilation.

On the other hand it is shown that due to the appropriate design of the heat exchanger equipped with a system of air dampers, it is possible to achieve very high values – not available by any other commercial unit – of sensible and latent heat recovery at extremely unfavourable simultaneous values, equal to 96.3% and 79.8%, respectively. Obtained results correspond to an overall heat recovery value of 91.4% and this value is unprecedented on the ventilation market. An advantage of the proposed solution (periodic-flow design of the heat exchanger) is its unique property. This results in the fact that during all tests even when the exhaust heat exchanger side air temperature was very low, there was frost forming on the heat exchanger surfaces. With the proposed solution, it is possible to remove energy consuming anti-freeze systems completely. In addition to the above, to achieve the average air temperature of 18.6 °C and to supply air moisture content of 5.82 g/kg, the use of energy consuming air humidifiers becomes unnecessary. For these reasons, systems with periodic-flow heat exchangers are ideally suited to the requirements of nearly zero-energy buildings.

The optimisation of the periodic operation of the heat exchanger should be sought primarily in the optimisation of its design to equate the air resistance on both sides of the exchanger and to reduce the difference in the balance of the supply air and extract air-flow while changing the air damper position. It has been found that it is important to determine carefully the time of changing the positions of the air dampers with regard to the outside and indoor conditions. For this reason, it becomes important enabling a controlled recovery of moisture from the extract air.

## Acknowledgment

This research is supported in the Smart Growth Operational Programme 2014-2020 by the Polish National Centre for Research and Development and by the Polish Ministry of Science 11.11.100.005.

## Nomenclature

$c_{pg}$ – specific heat of dry air, [Jkg$^{-1}$]
$c^{*}_{pp}$ – specific heat of water vapour, [Jkg$^{-1}$]
$h$ – air enthalpy, [kJkg$^{-1}$]
$h_{cz}$, $h_{ex}$ – outside, exhaust air enthalpy, [kJkg$^{-1}$]
$h_n$, $h_w$ – supply, extract air enthalpy, [kJkg$^{-1}$]
$p_b$ – atmospheric air pressure, [hPa]
$p_{gs}$ – water vapour pressure in saturation state, [hPa]
$Q_t$, $Q_u$, $Q_c$ – sensible, latent and total heat exchanger capacity, [kW]
$r_o$ – heat of water evaporation, [Jkg$^{-1}$]
$T$ – air temperature, [°C]
$T_{cz}$, $T_{ex}$ – outside, exhaust air temperature, [°C]
$T_n$, $T_w$ – supply, extract air temperature, [°C]
$V_{cz}$, $V_{ex}$ – outside, exhaust air volume flow rate, [m$^3$h$^{-1}$]
$V_n$, $V_w$ – supply, extract air volume flow rate, [m$^3$h$^{-1}$]
$x$ – air moisture content, [gkg$^{-1}$]
$x_{cz}$, $x_{ex}$ – outside, exhaust air moisture content, [gkg$^{-1}$]
$x_n$, $x_w$ – supply, extract air moisture content, [gkg$^{-1}$]

*Greek symbols*

$\eta_t$, $\eta_w$, $\eta_c$ – temperature, humidity and total heat exchanger efficiency, [%]
$\varphi$ – relative humidity, [%]
$\varphi_{cz}$, $\varphi_{ex}$ – outside, exhaust air relative humidity, [%]
$\varphi_n$, $\varphi_w$ – supply, extract relative air humidity, [%]

## References

[1] ***, Directive 2012/27/EU of the European Parliament and of the Council of 25 Oct., 2012
[2] Simonson, C., Energy Consumption and Ventilation Performance of Naturally Ventilated Ecological House in a Cold Climate, *Energy Build., 37* (2005), 1, pp. 23-35
[3] Liu, X. P., Niu, J. L., An Optimal Design Analysis Method for Heat Recovery Devices in Building Applications, *Appl. Energy, 129* (2014), Sept., pp. 364-372
[4] El Fouih, Y., *et al.,* Adequacy of Air-to-Air Heat Recovery Ventilation System Applied in Low Energy Buildings, *Energy Build, 54* (2012), Nov., pp. 29-39
[5] Calautit, J. K., *et al.,* CFD Analysis of a Heat Transfer Device Integrated Wind Tower System for Hot and Dry Climate, *Appl. Energy, 112* (2013), Dec., pp. 576-591
[6] Jaszczur, M., *et al.,* Experimental Analysis of the Velocity Field of the Air-Flowing through the Swirl Diffusers, *Journal of Physics: Conf. Series, 745* (2016), June, pp. 1-9
[7] Borowski, M., *et al.,* An Analysis of the Innovative Exhaust Air Energy Recovery Heat Exchanger, *MATEC Web Conf., 240* (2018), Nov., 02003
[8] Hughes, B. R., *et al.,* Passive Energy Recovery from Natural Ventilation Air Streams, *Appl. Energy, 113* (2014), Jan., pp. 127-140

[9] Vera, M., Linan, A., Laminar Counterflow Parallel-Plate Heat Exchangers: Exact and Approximate Solutions, *International Journal of Heat and Mass Transfer, 53* (2010), 21-22, pp. 4885-4898
[10] Oclon, P., *et al.*, Experimental and Numerical Investigation of Flow Distribution within the Heat Exchanger with Elliptical Tubes, *Procedia Engineering, 157* (2016), Dec., pp. 428-435
[11] Lopata, S., *et al.*, Experimental Stand for Investigation of Fluid-Flow in Heat Exchangers with Cross-Flow Arrangement, *E3S Web of Conferences, 13* (2017), Feb., 02001
[12] Jaszczur, M., Numerical Analysis of a Fully Developed Non-Isothermal Particle-Laden Turbulent Channel Flow, *Archives of Mechanics, 63* (2011), 1, pp. 77-91
[13] Kragh J., *et al.*, New Counter Flow Heat Exchanger Designed for Ventilation Systems in Cold Climates, *Energy and Buildings, 39* (2007), 11, pp. 1151-1158
[14] Hasan M. I., *et al.*, Influence of Channel Geometry on the Performance of a Counter Flow Microchannel Heat Exchanger, *Int. J. Therm. Sci., 48* (2009), 8, pp. 1607-1618
[15] Mijakowski, M., *et al.*, Heat Recovery Efficiency and Reduction of Heat Consumption by Ventilation – Effect of Different Strategies of Heat Recovery (in Polish), *Civil Eng., 4* (2010), pp.125-132
[16] Besler, M., Skrzycki, M., Laboratory Tests of Efficiency of Heat Recovery (in Polish), *Rynek Inst., 3* (2013), pp. 26-29
[17] Hughes, B. R., *et al.*, Passive Energy Recovery from Natural Ventilation Air Streams, *Appl. Energy, 113* (2014), Jan., pp. 127-140
[18] Seker, D, *et al.*, Frost Formation on Fin-And-Tube Heat Exchangers., *Int. J. of Refrigeration, 24* (2004), 27, pp. 367-374
[19] Huang, L., *et al.*, Experimental Study on Frost Release on Fin-And-Tube Heat Exchangers, *Exp. Therm. Fluid Sci., 33* (2009), 7, pp. 1049-1054
[20] Arundel, A. V., *et al.*, Indirect Health Effects of Relative Humidity in Indoor Environments, *Environ, Health Perspect., 65* (1986), Mar., pp. 351-361
[21] Donald, B. J. H., *Chemical Thermodynamics at a Glance*, Wiley-Blackwell, New york, USA, 2008

Paper submitted: January 15, 2019
Paper revised: February 26, 2019
Paper accepted: March 21, 2019