

Analysis of the anti-icing system used in air handling units with a counterflow heat exchanger

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Abstract. The present paper examines one of the most popular anti-icing solutions on the system performance. Different ratios of flowing fresh air to exhaust air were applied in order to heat the exchanger's iced surface with warm air. The analysed system demonstrated that the system efficiency significantly dropped down under unfavourable conditions in winter. The results show that this type of solution is insufficient and should not be applied. A long operation with the iced surface can cause irreversible permanent damage to the heat exchanger unit and a serious system failure.

1 Introduction

In 2010 the European Union introduced the EPBD [1] directive on the energy performance of buildings, which assumes that from 2019 all public buildings, and from 2021 all newly constructed buildings, will have to meet the requirements for nearly zero-energy buildings (nZEB) [1]. The ventilation, heating, hot water and cooling systems of buildings account for almost 50% of the total final energy consumption in the construction sector. Because of the national requirements defining the nZEBs, these systems have to be designed to meet the requirements for the maximum use of renewable energy for ventilation, heating, domestic hot water and cooling. With the growing share of heating loads and ventilation the heat recovery appears to be one of the key solutions to save large amounts of energy [2-4], however, a new ventilation type is necessary to maintain a desired indoor air quality [5-8].

Heating, ventilation and air conditioning systems are increasingly applicable in modern industry as well as for domestic purposes [9-12]. Skyscrapers, public buildings, hospitals or even houses have been equipped with ventilation and air conditioning systems [13,14]. A heat exchanger is the essential part of such system, responsible for effective and efficient energy recovery from the exhaust air. Parallel-plate heat exchangers were analysed by Vera and Linan [15] to provide formulations for heat exchanger design. Authors developed a 2-dimensional model to find analytical expressions to analyse counter-flow, parallel-plate heat

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exchangers. Kragh et al. [16] examined new geometry for counterflow heat exchange used in ventilation systems in cold climates. The efficiency of the proposed heat exchanger was calculated based on the theoretical model and validated using experimental measurements. Another paper [17] analysed the influence of channel geometry on the performance of a counter flow heat exchanger.

The most commonly used heat exchangers have efficiency up to about 90% [18]. Nevertheless, a relatively high value is received under laboratory conditions and may differ significantly under real conditions, especially when conditions are adverse [19]. Catalogue sheets of heat exchangers typically provide the temperature efficiency under optimum or standard conditions. However, the speed of the medium flowing through the exchanger is an additional parameter, affecting the equipment temperature efficiency, apart from temperature conditions. What is important, it is not standardised in any way, whatsoever. Because of that the manufacturers, based on their know-how, appropriately adapt the air flow speed to a specific piece of equipment to obtain as high as possible efficiency. When during the winter months the air temperature falls below -5°C a danger of heat exchanger icing, and at a later stage of total freezing, appears in such exchangers. This phenomenon 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. This effect occurs as a result of the air temperature decline below the dew point [20,21]. 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 significant increase in the flow resistance. In adverse situations, such an effect can lead to irreversible mechanical changes in the exchanger and its damage.

Various technical solutions are used to defrost exchangers. The most popular of them consists in the installation of an electric heater in the supply air duct to heat it to an appropriately high temperature before it is let in directly to the exchanger. Also, a special fresh air bypass can be applied, which makes that the air bypasses the heat exchanger. As a result, cold air is supplied to the building, and the exhaust air retains its enthalpy and thereby warms up the heat exchanger and melts the ice layer. Certain manufacturers apply in their units a solution, in which to warm up the heat exchanger they entirely switch off the fresh air fan. The application of a heater to warm up the supplied air substantially increases the electricity consumption, while the stopping of supply fans delivering the fresh air entirely disturbs the air flow not ensuring acceptable heat comfort or causing total deactivation of the ventilation system [22]. Heat exchangers can also be defrosted by changing the ratio of air supplied to exhausted from premises, thereby causing the exchanger defrosting. However, this solution significantly reduces the number of fresh air exchanges in the building and results in the origination of substantial negative pressure in the building. Then the deficit of air is supplemented through leaks in the building envelope elements causing the building cooling.

The temperature efficiency of heat exchangers used in recuperators and air handling units is determined in Poland according to the Polish regulations PN-EN 308 standard [23]. It declares a method of efficiency calculation based on parameters described below. Coefficients defining the temperature η_t and humidity η_x efficiency for a specific part of equipment are calculated according to the following equations:

$$\eta_t = \frac{t_{22} - t_{21}}{t_{11} - t_{21}} \quad \text{and} \quad \eta_x = \frac{x_{22} - x_{21}}{x_{11} - x_{21}} \quad (1)$$

where: t_{22} , t_{21} are the temperature of supply & outdoor air, t_{21} is exhaust air temperature; x_{22} , x_{21} stand for the humidity in the supply & outdoor air, x_{21} is humidity in the exhaust air.

The temperature efficiency of heat exchangers is calculated as the ratio of heat recuperated in the exchanger transferred from the exhaust air to the supply air, to the total demand for heating power, which should be delivered to heat the outdoor air to the indoor air temperature. The humidity efficiency of heat exchangers is defined as the ratio of moisture flow recuperated in the exchanger transferred from the exhaust air to the supply air, to the total demand for the amount of moisture, which should be delivered to humidify the outdoor air to the indoor air humidity. During tests, the ambient temperature of the equipment should range between 17 and 27°C. Tables 1 and 2 present parameters of the air supplied to the heat exchanger unit.

Table 1. Temperature applied during tests without condensation.

Symbol	Temperature, °C
t_{11}	25
t_{w11}	< 14
t_{21}	5

Table 2. Temperature applied during tests with condensation.

Symbol	Temperature, °C
t_{11}	25
t_{w11}	18
t_{21}	5

When the condensation occurs, it is necessary to perform additional tests using a set of parameters presented in Table 2. When the equipment is intended to operate in a cold climate, an additional 6-hour test is applied to another set of temperature values, consisting in the visual inspection of the recuperator.

As it can be noticed, the insufficient specification of equipment operation parameters gives manufacturers a lot of room for manoeuvre and enables applying by them parameters optimal for their equipment. The presented studies are aimed at presenting the parameters of the equipment operation under actual operational conditions and the effect of exchanger icing. The given results of the main unit tests under adverse weather conditions and at a closed bypass show the impact of exchanger icing and resulting implications.

2 Methodology

The study was aimed to verify the performance of one of the most popular methods preventing the counter-flow exchanger icing in an air handling unit. The object of tests consisted of the main unit with a standard control system, which activates the anti-icing mode when the pressure of air flowing through the exchanger falls, or when the exhaust air temperature t_{12} goes down below the predefined value. The growing flow resistance means that ice crystals start forming in the exchanger, making a fluid flow of air difficult. The air handling unit was also equipped with a fresh air bypass, which was closed for the current test. The main unit was placed in an appropriately insulated chamber, inside which the air temperature was about 20+/-1°C. The air was supplied to the air handling unit by two ducts equipped with glycol heaters/coolers, as appropriate. Fig. 1 presents the measuring section of the heat exchanger. The temperature of the exhaust air from the room was maintained on a constant level of 20°C, while the temperature of the fresh air was gradually decreasing from -5°C to -15°C. JUMO EE650-T2L200 airspeed converters were used to record the volume flow rate ($V_{11}, V_{12}, V_{21}, V_{22}$). JUMO EE210-HT3xPBFxB hygroscopic humidity and

temperature converters were used to measure the air temperature and humidity ($t_{11}, t_{12}, t_{21}, t_{22}$ and $\varphi_{11}, \varphi_{12}, \varphi_{21}, \varphi_{22}$). The data acquisition from the measuring sensors was carried out using a Mitsubishi Electric FX5U module and a MAPS HMI 750 panel. The measured values were recorded with 1 second intervals during 2500s. The data recording was activated at the moment of automatic switching on the defrosting mode by the unit. The measurement data was averaged with an interval of 10s and presented in the following figures.

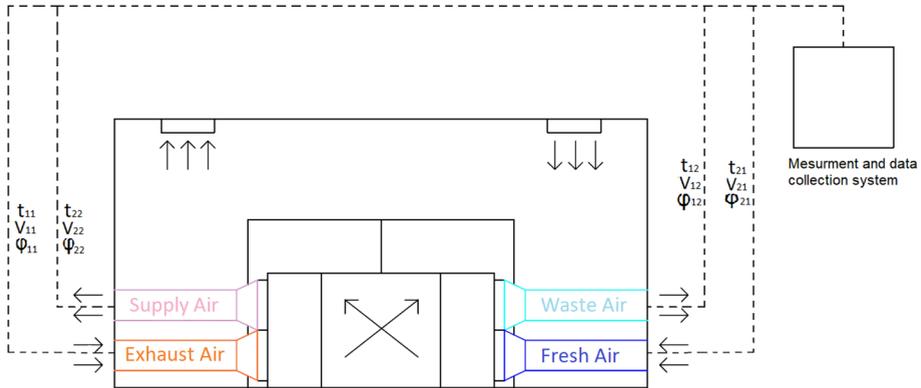


Fig. 1. Diagram of the test section.

Moisture content in the air - the absolute humidity is defined as a ratio of water vapour mass in the air to the dry air mass. Using Dalton's law [24] and the gas equation of state we obtain a formula allowing to determine the moisture content in the air based on its temperature and relative humidity:

$$x = 0,622 \cdot \frac{\varphi \cdot p_{gs}}{p_b - \varphi \cdot p_{gs}} [gH_2O / kg \cdot ps] \quad (2)$$

where

$$p_{gs} = 6,1121 \cdot e^{\frac{17,502T}{T+240,97}} [hPa] , \quad p_b = 1013,25 [hPa]$$

Enthalpy of air: The enthalpy of humid air with the moisture content x can be defined as the enthalpy of x kg of water vapour mixture with 1kg of dry air. Assuming that for such a mixture at the temperature of $0^\circ C$ the enthalpy equals zero we obtain the relationship:

$$i = c_{pg} \cdot t + x(c_{pp} \cdot t + r_0) \left[\frac{kJ}{kg} \right] \quad (3)$$

Total temperature efficiency: The heat exchange efficiency was determined based on the formula taking into account the various values of the flow and then the arithmetic mean was calculated from the obtained results:

$$\eta_t = \frac{V_{21} \cdot (t_{22} - t_{21})}{V_{11} \cdot (t_{11} - t_{21})} \cdot 100\% \quad (4)$$

where V_{21} and V_{11} are volume flows of the exhaust and outdoor air.

Total exchanger efficiency: The efficiency of the total exchange of energy contained in the air flowing around the exchanger considers the latent heat contained in the air moisture. Also, in this case, the end result is an arithmetic mean of obtained calculations, as follows:

$$\eta_i = \frac{V_{21} \cdot (i_{22} - i_{21})}{V_{11} \cdot (i_{11} - i_{21})} \cdot 100\% \quad (5)$$

3 Results and discussion

Studies were carried out based on the equipment manufactured on a large scale and using one of the widely used anti-frost systems, i.e. 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, at the flow equal to $V_{21}=V_{11}=1800\text{ m}^3/\text{h}$, for the air parameters: $t_{21} = -15^\circ\text{C}$, $t_{22} = 14.7^\circ\text{C}$, $t_{11} = 20^\circ\text{C}$, the recuperation efficiency was given as 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. 2(a)).

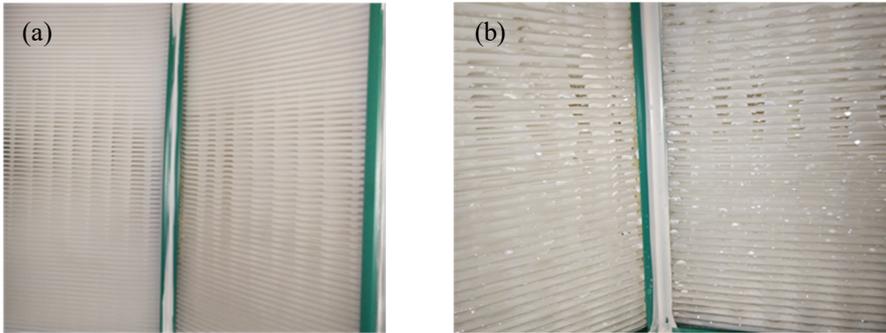


Fig. 2. The heat exchanger surface condition: at the initial state (a) and the end of measurements (b).

Figs. 3-5 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 - a fresh air bypass, which was closed, for the period of the performance of this test. It should be emphasised that the collaboration of both systems would protect the exchanger, however, at the price of worsening the heat comfort and resulting in a substantial decline in the system efficiency.

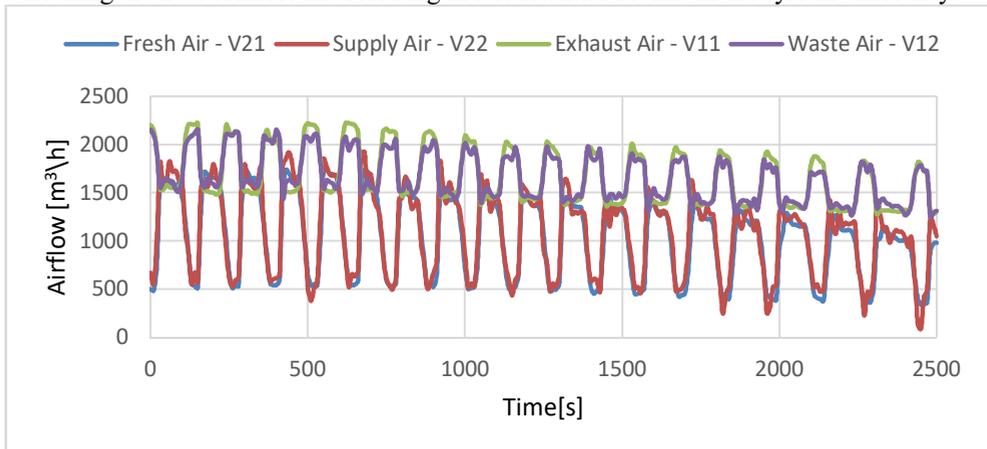


Fig. 3. The air flow through the heat exchanger over time.

Fig. 3 shows changes in the air flow rate, both on the supply and exhaust side. The system cyclically increases the rotations of the hot air exhausting fan and reduces the rotation of fan responsible for supplying the fresh air to the building. A gradual icing of the heat exchanger surface was observed during the test. It results in decreasing, with each cycle, the maximum

value of the hot air flow rate from 2220 m³/h to 1780 m³/h. Increased airflow resistance causes this phenomenon due to the freezing of the dropped out condensate. The test was stopped at the moment when the amount of frozen condensate became a threat of the heat exchanger damage. It is possible to presume that over time a new layer of ice will accumulate on the exchanger, ultimately making the further unit's operation impossible.

Fig. 4 presents the value of temperature change over time for the analysed unit. Average temperature efficiency determined based on measurements was about 91.7% and was slightly higher than that declared by the manufacturer. However, such efficiency does not take into account phase transitions proceeding in the heat exchanger as well as fluctuation in the air flow rate. The analysed unit features a built-in control panel presenting the temperature and the mode of unit operation, which does not allow to state properly whether the equipment operation is correct because the information received is that the temperature of the supplied air is around 17°C. When we determine the total temperature efficiency, considering the air flow, based on the measurement data we obtain the value of total temperature efficiency $\eta_t = 66.8\%$. Moreover, calculating the ratio of fresh air flow to the exhaust air flow we obtain the value $\frac{V_{21}}{V_{11}} = \frac{260766}{417005.8} = 0.625$. That means a significant deficit of air in the building and hence a considerable negative pressure. During the main unit operation at such parameters the air in the building is supplemented by leaks in the structure, so most frequently by drawing cold air from the environment.

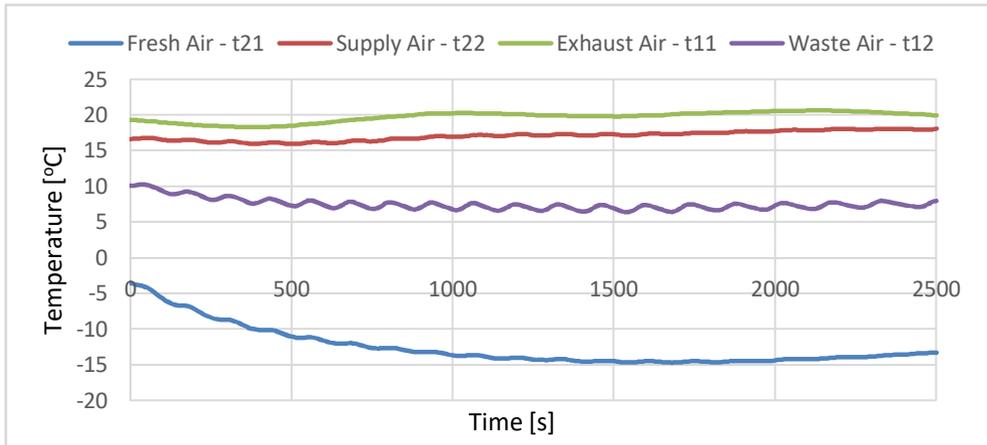


Fig. 4. Air temperature over time.

Fig. 5(a) shows the value of relative humidity during the measurements. One may infer from this Figure a change of value for the fresh air results from a gradual decrease of the air temperature. However, the actual water content in the air did not change (see Fig. 6 for reference). On the other hand, as shown in Fig. 5(b) with increasing flow of humid air exhausted from the room (RH approx. 60%), the condensation of water on the heat exchanger internal surface grows, and this phenomenon proceeds cyclically with output changes. The total enthalpy efficiency evaluated based on measurements is as low as $\eta_i = 42.7\%$.

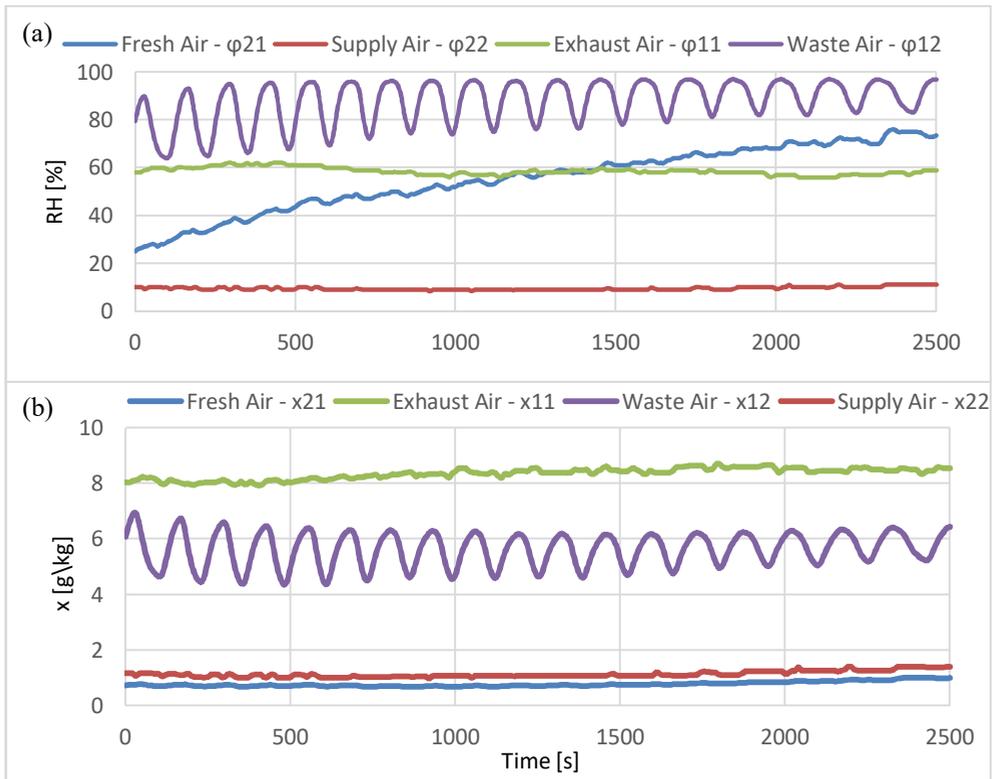


Fig. 5. Relative humidity (a) and water content (b) over time.

To protect the heat exchanger structure, 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. Fig. 2(b) shows a heat exchanger surface when ice crystals are formed between the exchanger layers and are visible. They occupy a significant surface of the heat exchanger, making the air flow very difficult and thereby substantially increasing the pressure drop.

4 Conclusions

The paper presents the results of temperature, humidity and airflow measurements during the operation of the anti-icing system of the air handling unit equipped with a counter-flow heat exchanger. Based on the carried out studies it is possible to state clearly that this system does not work well at low-temperature conditions and should not be applied without additional support in the form of another 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 proper ventilation of the building. The deficit of fresh air can result in significant growth of CO₂ in the room and also can lead to the drop in the air temperature inside the building. During 40 minutes of 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. It is possible to expect that in the case of implementing the second anti-icing system, like a fresh air bypass, the exchanger icing could be avoided.

Cyclical bypassing of cold air would result in defrosting the exchanger at the cost of temperature decrease of air supplied to premises. As standard such equipment is also equipped with an electric heater (or with another air heating device) to ensure appropriate thermal comfort. The temperature efficiency of the unit during studies was 91.7%. This is a value very close to that provided in the catalogue sheet. However, attention should be drawn to the fact that tests of the exchanger efficiency according to standard PN-EN 308 [23] should be carried out at appropriate temperatures and at equal air flow rates. For the sake of comparison, the efficiency considering an uneven air flow rate through the exchanger was calculated and the value obtained was 66.8%. This efficiency could be satisfactory in extreme temperature conditions if the air flow ratio was 1:1. It is worth to notice that the total enthalpy efficiency of the exchanger was only 42.7%. Taking into account the presented results it is possible to state that the unit operating within the mode, in which measurements were performed, should not be used in a cold climate, where the air temperatures can fall below -5°C.

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