Experimental investigation of frost formation on air to air counter flow heat exchanger in air handling unit and climatic influence on dry, wet, frost operation condition

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ABSTRACT

The work presented in this paper investigates frosting problem on high efficient air to air counter flow heat exchanger. The presented investigation consists of two main activities.
Firstly, experimental tests on a counter flow plate heat exchanger in a climatic chamber have been performed to better understand under which exhaust condition frost starts forming in the exit region of the exhaust port. A special experimental set-up to investigate frost formation on the air to air heat exchanger was developed. Set up consists of a cold chamber that could maintain freezing temperatures and a warm chamber imitating indoor condition. Both temperature and relative humidity could be controlled. Experimental tests have indicated that the temperature gradient on the exhaust port is very significant. While in one part of the exhaust port occurs frosting the other part has still positive temperature. Observed temperature difference across exhaust port was up to 6.8 ^{O}C. From the experiments, it is observed that the frost starts to form when mean temperature of the exhaust air is at 0°C. Furthermore, it can be stated that even though frost formation depends greatly on the outdoor temperature, the level of indoor humidity also plays some role in location of frost formation.
Secondly, the percentage of operation condition of the exchanger – dry, wet and frost - has been analyzed for four geographical locations – Southern Scandinavia, Central Europe, Southern Germany and Austria, and Scotland. Regarding indoor conditions, data has been created using dynamic simulation software BSim with loads representative for classrooms and offices. This investigation has shown that for the respected regions risk of frosting can occur but would not be higher than 5.8 % of operational time in the most severe investigated location. Finally, for the investigated high heat recovery efficient exchanger, it is discovered that heat recovery efficiency drops below required by Ecodesign 73% when outdoor temperature would drop below -14 °C.

KEYWORDS

Air to air heat exchanger, frost formation, climate chamber, dry, wet, frost operation

1 INTRODUCTION

Legal requirements regarding energy efficiency and rational exploitation of the energy resources [1] are getting stricter in countries with cold climates due to high space heating consumption [2] and rising environmental awareness. For space heating in cold climates is consumed between 40-60 % of the total building energy demand [2] and 30-60 % from the total space heating consumption is due to heating up the incoming fresh air [3]. For that reason, from January 2018 the minimum demand for heat recovery in ventilation units has been increased from 67 % to 73 % by [4] in EU Member States. Although there is a big energy saving potential in ventilation units with high heat recovery, in cold climates it can lead to frost formation on the plate surface of the warm side. During winter, when the plate surface temperature falls below freezing point, there is a risk of condensation and frost formation. Consequently, when the highest heat recovery efficiency is needed, it cannot be provided. Frosting leads to reduction of the heat recovery efficiency, increase in pressure drop in the return airflow channels, higher electricity consumption for the fans, and draught in the space due to low supply air temperature. In case of severe frosting, there is also a risk of damage of the heat exchanger [5] [6]. Even though the topic has raised concerns for the past 35 years, the published papers related to frost formation in heat exchangers are limited. Frost formation has been detected by many researchers, but only few have investigated frost prevention or defrost methods. Due to that, literature states that the problem of frost is still unresolved and even the most commonly used defrost methods face challenges in cold climates [5].

In counter flow heat exchangers, the supply and return airstreams flow along each other, separated by thin aluminum/ plastic plates. Heat is transferred from the return to the supply air. The heat exchanger is sealed, and therefore, the two flows are unmixed. They are very popular in the northern countries because of their reliability, long service-life due to no moving parts, operation with unmixed fluids and high heat recovery efficiency [6].

The purpose of the paper is:

- To investigate where and under what condition frost starts forming in the exit region of the exhaust port.
- To investigate percentage of operation condition of the exchanger – dry, wet and frost for preselected geographical locations – Southern Scandinavia, Central Europe, Southern Germany and Austria, and Scotland.

Additionally, the influence of different indoor conditions on frost formation and temperature distribution across the ports is analyzed. Loads typical for office and class room are investigated.. Furthermore, the required minimum heat recovery efficiency is analyzed taking account for efficiency in freezing conditions.

2 METHODOLOGY

2.1 Experimental investigation

A schematic plan view of the experimental setup is shown in Fig.1. The setup is designed to achieve outdoor temperatures corresponding to winter conditions and typical indoor conditions. It consists of an insulated shipping container divided into two environmental chambers: a warm chamber where the temperature and humidity is controlled by a duct electric heater and a humidifier and a cold chamber where the temperature is controlled by a cooling unit.

To achieve indoor conditions of a typical classroom or office, the temperature for the extracted air should be maintained at 22 °C. For that reason, a circular electric duct heater is used to maintain the air temperature at the desired set point [7]. The heater that is installed in the warm chamber is connected to a fan, that has the purpose to circulate the air in the room so that stratification does not occur. Moreover, in terms of control, the fan also helps to better control the temperature, as the sensor is measuring a uniform temperature of the well mixed chamber. In Fig.1, the placement of the fan (no. 4) and the duct with the electric heater (no. 5) is shown. Both are placed at the ceiling level and the duct has a 90-degree bend towards the floor, so that the flow does not influence the extraction.

The humidification system consists of a domestic water pump (no. 1), a water heater (no. 2) and the Vapac Minivap Humidifier (no. 3), as shown in Fig. 1. The pump provides water to the small water heater. After the water gets heated it is pumped to the humidifier, where it gets heated further until it boils. The steam is released into the ventilation duct after the circular electric duct heater (no. 5) [8]. The desired relative humidity that should be maintained stable is in the range 25 - 65 %.

The cooling unit consists of an evaporator placed in the cold chamber. The compressor and condenser are placed outside - on the roof of the container. The evaporator is connected to a duct system (no. 4) that provides cold air to the room, without influencing directly the air that is being supplied to the warm room. In addition, a circular electric duct heater is installed (no. 5) to defrost the evaporator and to maintain the ambient temperature in case higher set points are desired. The cooling unit is shown in Fig. 1.

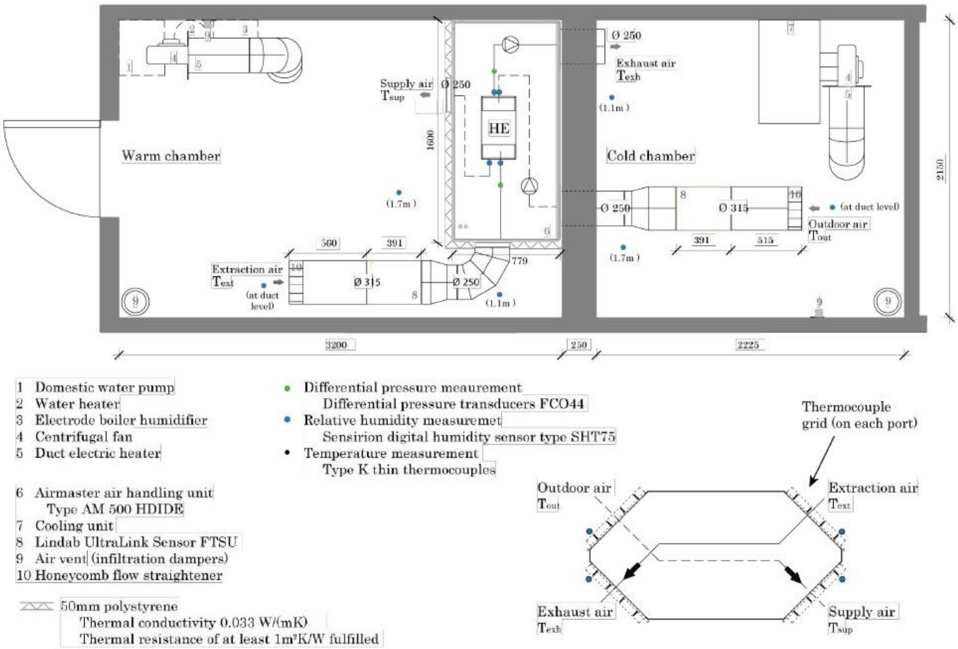

Figure 1: Schematic of test chamber layout and equipment.

The decentralized ventilation unit is placed in the warm chamber and it is tightly connected to the cold chamber through ducts. The operation principle of the unit is as following: air at outdoor conditions (tout) moves by suction in the ventilation unit passing through the heat exchanger and then is provided to the room (tsup), while the warm extraction air (text) moves by suction in the ventilation unit passing through the heat exchanger and then is exhausted (texh) on the cold side, as shown in the 3D drawing in Fig 2.

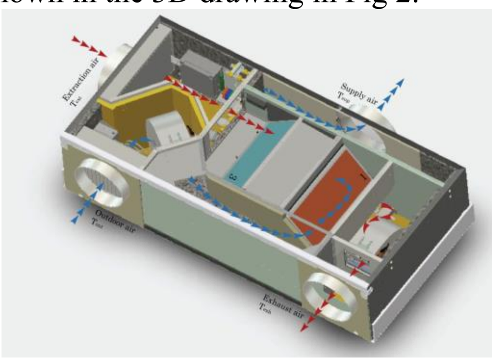

Figure 2: 3D graphic of tested heat exchanger in air handling unit.

The parameters that are measured in order to test the performance of the heat exchangers are air temperature, relative humidity, air flow and differential pressure. For both air streams of the heat exchanger, the parameters are measured at different locations, as shown in Fig. 1.

The air temperature is measured with a type K thermocouples (Chromel /Alumel) that are gold coated, through sputtering technique. In this case, radiative heat exchange that can influence the measurement is decreased by using coating [9]. The temperature is measured with four grids of thermocouples on all ports of the heat exchanger. Since there is a risk of uneven temperature distribution, 16 temperature measuring points are distributed evenly across each grid. The recommended number of temperature sensors for rectangular ducts is 9, according to ASHRAE [10].
The relative humidity is measured with a Sensirion digital humidity sensor type SHT75. In the ventilation unit, as shown in Fig. 1, the relative humidity is measured with the Sensirion sensor. To monitor the temperature and relative humidity in the two chambers, 3 sensors are used, in each room.
The airflow rate is measured on both airstreams: for the outdoor - supply airstream the measurement is performed on the outdoor side, while for the extraction - exhaust airstream the measurement is performed on the extraction side, see Fig. 1. The device used is an UltraLink FTSU with a diameter of 315 mm and a length of 391 mm. The flow is measured with and angled ultrasonic beam that provides high accuracy.
The pressure difference is measured using a pressure transducer FCO44 manufactured by Furness Control Limited that has a range of ±500 Pa. The measurement is performed on the extraction - exhaust side (warm side) of the heat exchanger. The aim is to detect pressure drop increase caused by frost growth on the plates of the exhaust side of the heat exchanger. Pressure taps are drilled on the surfaces on which flow passes, therefore measuring static pressure.
Before running the experiments, analysis on indoor relative humidity - outdoor temperature relation was done. The purpose was to find out which indoor air relative humidity ranges (for typical indoor loads) correspond to outdoor negative temperature during the working hours. The findings from this investigation helped defining the relative humidity ranges that need to be tested for the frost formation experiments. The results were obtained using dynamic simulation tool BSim and provide the relation between the indoor relative humidity (at 22 °C) and outdoor temperature. For classrooms and offices, indoor relative humidity does not reach above 41 % and 37 %, respectively, for the outdoor temperatures below 0 °C. This relative humidity limits are the same for the investigated locations – Gothenburg (Southern Scandinavia), Groningen (Central Europe), Innsbruck (Sounth Germany and Austria) and Leuchar (Scotland). The last location, Leuchar, has almost no negative temperatures, therefore this location and the area it is representing (Scotland) can be considered as out of risk for frost formation.

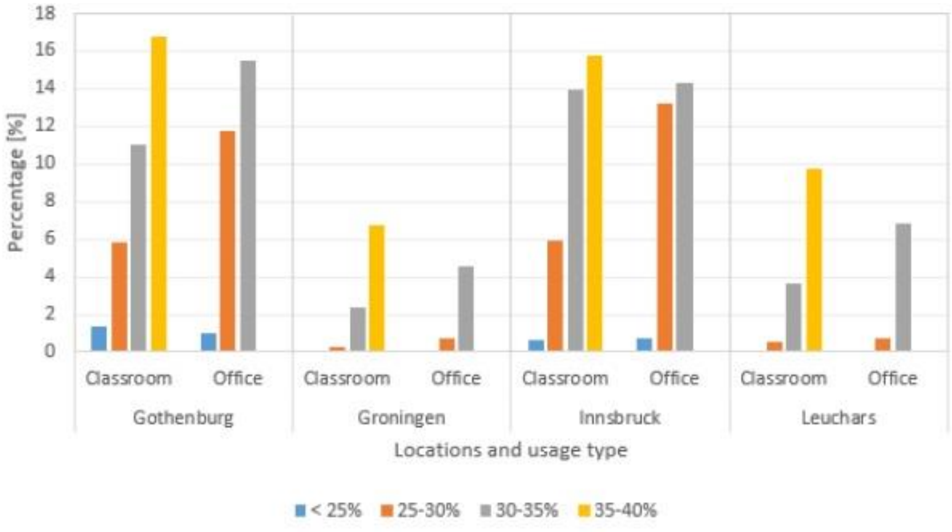

Figure 3: Percentage of critical indoor relative humidity ranges from the total number of yearly working hours with 22 °C as indoor temperature.

The pre-defined extraction temperature for the tests is 22 °C and the relative humidity range is determined based on the results presented in Fig 3. Due to the discovered not sufficient cooling capacity on the cold site of the chamber the indoor temperature had to be decreased from 22 °C and as consequence indoor relative humidity had to be adjusted to compensate for that. Table 1 presents both indoor conditions as scheduled based on loads and climate analysis (temperature and relative humidity) and equivalent indoor conditions for lowered temperature matching the available cooling capacity that corresponds to indoor temperature at 22 C.

Table 1: Frost formation test condtion - actual tested and equivalant to them.

Experiment	**Equivalent conditions to ind. temp. 22 °C**	**Indoor scheduled conditions**
Condition 1	22 _C and 22 %	RH 22 _C and 22 % RH
Condition 2	16 _C and 38 %	RH 22 _C and 27 % RH
Condition 3	15 _C and 46 %	RH 22 _C and 30 % RH
Condition 4	16 _C and 58 %	RH 22 _C and 40 % RH

2.2 Operation condition – dry, wet, frost

Due to the varying outdoor conditions during the year, the dynamic temperature efficiency of the heat exchanger is looked into. It is compared with the minimum heat recovery efficiency required by the Ecodesign [4], which is currently at 73 %. Moreover, the duration of the periods with dry, wet and frost operation conditions for the heat exchanger are also investigated for the chosen four geographical locations. Yearly (hourly) data for outdoor, supply, extraction and exhaust air temperature, based on BSim simulations and validated software to calculate performance of investigated heat exchanger is analyzed. Only working hours are investigated. Depending on whether the exhaust temperature is below the dew point for extraction air or below the frost limit temperature of 0 °C, the periods with dry, wet and frost conditions are sorted.

The temperature efficiency for the different conditions is calculated using the formula in equation 1. When it comes to calculating the efficiency under freezing conditions, the supply temperature is calculated taking into account the maximum possible heat transfer before frost occurs. The formula which is used can be seen in equation 3.

$$\eta_t = \frac{t_{sup} - t_{out}}{t_{ext} - t_{out}} \; [-] \tag{1}$$

t_{out} - outdoor air temperature [°C]
t_{sup} – supply air temperature [°C]
t_{ext} – extraction air temperature [°C]

$$Q_{\max possible} = m_h \cdot (h_{ext} - h_{exh'}) \tag{2}$$
$$t_{sup,frost} = \frac{Q_{\max possible}}{m_c \cdot Cp_c} + t_{out} \; [°C] \tag{3}$$

$Q_{max\ possible}$ – maximum possible heat flow based on freezing point threshold [W]
m_h – mass flow rate on the warm side [kg/s]
m_c – mass flow rate on the cold side [kg/s]
h_{exh} – exhaust air enthalpy [kJ/kg]
$h_{exh'}$ – freezing limited threshold air enthalpy [kJ/kg]
Cp_c – air heat capacity on the cold side [(J*kg)/K]

3 RESULTS

3.1 Experimental investigation

Fig. 4 shows the test conditions on an I-x diagram. The lines for the different conditions indicate the air treatment process and correspond to mean temperature values. During the experiments, it was observed that for all the relative humidity levels, frost on the exit region of the exhaust side appears at 0 °C mean exhaust temperature. Still, it is not possible to tell whether frost starts to form inside the heat exchanger sooner. Moreover, significant temperature gradient on the exhaust port for all the test conditions was observed. As a consequence on the heat exchanger exhaust port that is below 0 °C is formed frost and the part with positive temperatures remains free of frost.

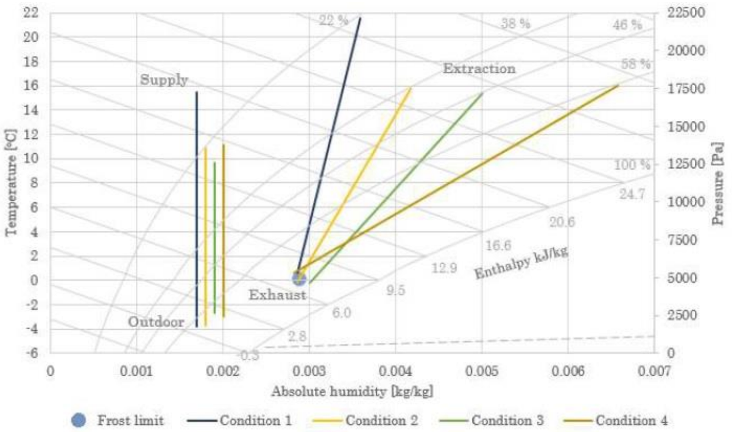

Figure 4: Frost formation test conditions presented on an I-x diagram - for mean temperatures.

Condition 1 - balanced flows
Temperature gradient across exhaust port

-1.2	-2.3	-2.5	-2.3
0.1	-0.3	-0.5	-0.5
1.8	1.5	1.2	1.2
3.8	3.1	3.1	3.3

Condition 2 - unbalanced flows
Temperature gradient across exhaust port

-1.7	-2.2	-2.4	-2.4
-0.3	-0.6	-0.8	-1.0
1.0	0.8	0.3	0.5
2.4	1.6	1.6	1.7

Condition 3 - unbalanced flows
Temperature gradient across exhaust port

-1.6	-1.8	-2.2	-1.9
-0.4	-0.9	-1.4	-0.7
0.8	0.4	-0.1	0.2
2.0	1.5	1.3	1.6

Condition 4 - unbalanced flows
Temperature gradient across exhaust port

-1.6	-2.3	-2.7	-2.3
0.5	-1.8	-2.0	0.1
3.3	-0.3	-0.6	2.0
5.2	4.0	3.7	4.1

Figure 5: Temperature gradient on exhaust port for all 4 test condition.

During the experiments it was observed that due to the low level of water vapour in the air with 22 % relative humidity (condition 1), frost formation started without condensation. The water vapour in the air turned directly into a very thin layer of frost on the coldest part of the port. The reason for that is because the dew point for that condition (-0.8 °C) is below the triple point (0 °C). The triple point (gas-liquid-solid point) corresponds to the pressure at 0 °C below which liquid cannot exist [11]. Due to low vapor content in the air it would take more time for frost to build up with relative humidity ranges corresponding to dew point value of below 0 °C.

The low temperature on the outdoor port (intake) decreases the temperature on the top part of the exhaust port. Additionally, the temperature gradient increases when there is more condensation, as with condition 4 (58 % relative humidity). Compared to the other cases, the temperature at the bottom of the port is significantly higher. In this case, the greater condensation rate keeps the plate surface warmer and maintains higher air temperature values.

Depending on the relative humidity, frost formation starts in different places on the port. With low relative humidity (22 %), it starts on the coldest part of the port (the top side). With higher relative humidity, it starts a bit lower, though still in a spot with a local negative temperature value. As condensation occurs, it drips down along the plates of the heat exchanger. The more condensation there is, the lower on the port the condense droplets reach before they freeze. Fig. 6 indicates where frost starts forming with the different relative humidity ranges. The frost formation location is marked with a red circle to indicate where frost starts to occur.

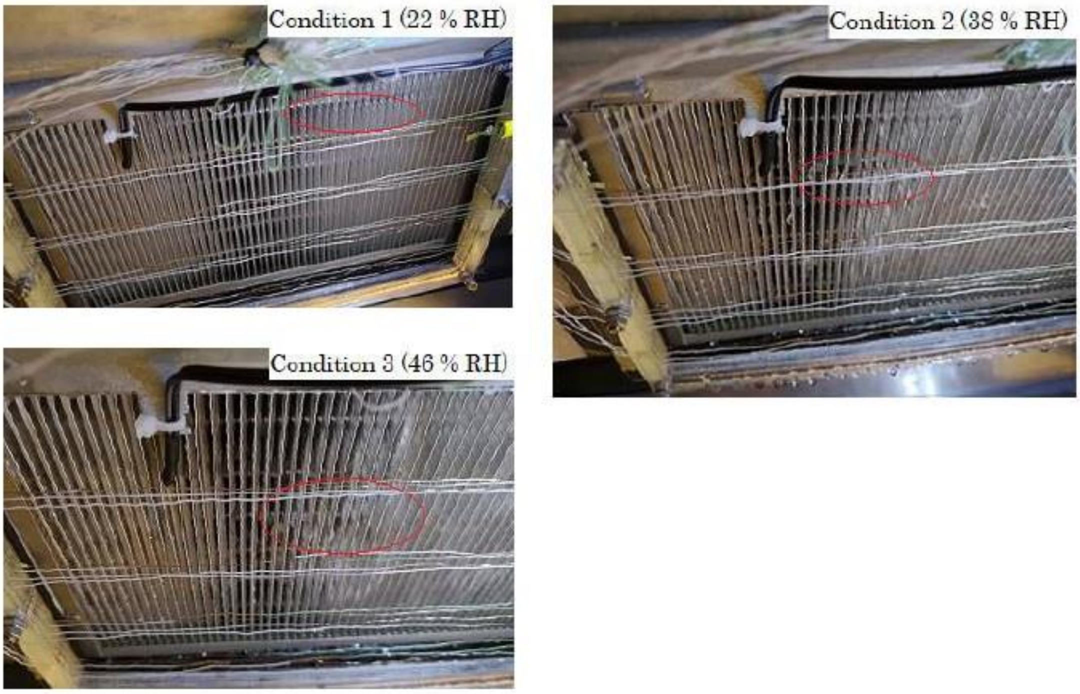

Figure 6: Frost formation location at 1-3 condition.

3.2 Operation condition – dry, wet, frost

Yearly (hourly) data for outdoor, supply, extraction and exhaust air temperature, based on BSim simulations and outcomes from validated software to calculate performance of investigated heat exchanger, was analyzed for three selected locations with the frost risk. Only working hours were investigated. Depending on whether the exhaust temperature is below the dew point for extraction air or below the frost limit temperature of 0 °C, the periods with dry, wet and frost conditions are sorted. When it comes to calculating the efficiency under freezing conditions, the supply temperature was calculated taking into account the maximum possible heat transfer before frost occurs.

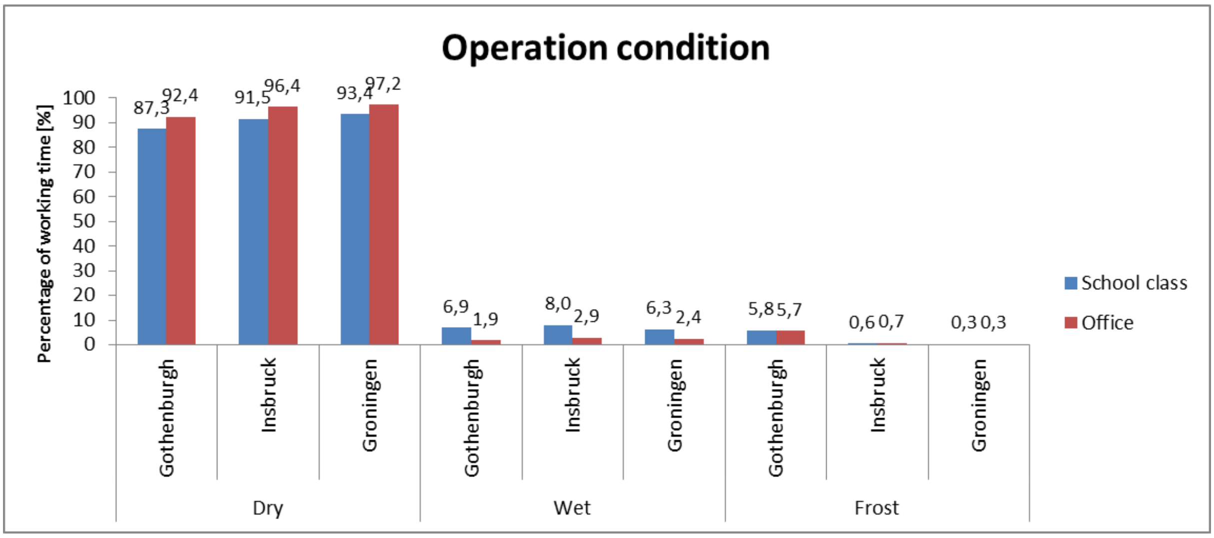

Figure 7: Operation condition – dry , wet, frost.

4 CONCLUSIONS

The focus of this paper is the frost formation on air to air counter flow heat exchangers. To investigate this topic an experimental setup with a decentralized ventilation unit containing the high efficiency counter flow heat exchanger was built and equipped with sensors. The setup consists of a cold room that could maintain freezing temperatures and a warm room where both temperature and relative humidity could be controlled.

To observe when frost occurs in the heat exchanger, a video camera was mounted inside the ventilation unit as well as pressure sensors to measure the pressure increase on the warm side of the heat exchanger. Even though frost formed, it was not possible to detect it with pressure increase nor with the camera. Therefore, visual inspection of frost formation was necessary.

From the experiments it can be concluded that the mean temperature of the exhaust air is 0 °C when frost occurs on the heat exchanger. Furthermore, it can be stated that even though frost formation depends greatly on the outdoor temperature, the level of indoor humidity also plays a significant role. It has been observed that the temperature gradient of the exhaust port can be 6.8 °C from warmest to coldest area. The coldest point is the area close to the outdoor port and the gradient is greater when indoor humidity is also higher.

At last, the risk of frost in the investigated geographical locations has been evaluated. Investigation indicates that the highest risk of all investigated areas is in Southern Scandinavia, but still do not exceed approx. 6% of working time. In the rest of the investigated locations the risk of frost formation is insignificant.

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