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Residential demand-controlled heat recovery ventilation system and performance in the Canadian Arctic

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Executive Summary

A demonstration house was previously built and commissioned in Iqaluit, Nunavut, Canada. The purpose of the overall effort is to develop, integrate, and evaluate technologies in this high-performance building located in the Canadian Arctic, while considering the unique social, economic, and logistical challenges associated with its remote location. Previous work consisted of monitoring and reporting on the energy consumption due to heating between April 2016 and April 2017 and evaluating the thermal resistance of the building envelope measured in-situ over a full year. The purpose of this next stage of research is to contribute experimental data of the prototype residential CO_2-based demand-controlled ventilation (DCV) system with heat recovery. The experiments were performed in the cold climate of Iqaluit, where the average annual outdoor temperature is approximately -9 °C. This paper outlines the development, implementation, and monitoring of the ventilation system that took place between April 2017 and April 2019.

The ventilation system was equipped with two electric preheaters to enable the prevention of frost build-up in the heat recovery ventilator (HRV) while maintaining adequate ventilation according to the demand. An electric heater was included after the HRV to control the supply air temperature. The electricity consumption of the HRV, preheaters, and supply air heater was measured and the supply airflow rate and pertinent temperatures in the ventilation system were also monitored to enable assessment of the system's performance. On November 26th and 27th, 2018 the functionality of the demand-controlled aspect of the ventilation system was verified. Between December 2018 and April 2019 the ventilation system was operated using two separate frost prevention techniques, each for seven week periods: 1) with electrical preheat enabled and 2) with electrical preheat disabled, instead relying on recirculation as a defrost technique.

The two-day testing of the demand-controlled aspect of the ventilation system consisted of varying the occupancy from 0 to up to 5 occupants. During this time, the CO_2 concentration in the space and HRV fan operation was monitored to ensure the HRV fan was operating based on the measured CO_2 concentration. The testing proved the viability of using low-cost CO_2 sensors for ventilation control in a small residential dwelling. Further testing will be conducted on the DCV system's performance; however, the two-day study sufficed for verifying the functionality of the DCV strategy and showed that if occupants enter the space, sufficient ventilation will be supplied.

The seven-week experiment using electrical preheat as the frost prevention technique displayed that, on average, the sensible recovery efficiency (SRE) of the HRV and ventilation system was 70.1% and 38.4%, respectively. The average daily energy consumption of the preheaters, supply air heater, and HRV was 6.72 kWh, but when considering the intake duct gains, HRV case gains, and exhaust duct gains in the ventilation system, the typical energy consumed to provide outdoor air at 18 °C was actually 11.9 kWh per day, which translates to a daily energy use intensity of 0.49 kWh/day/m^2. Unsteady control of the electrical preheaters allowed the system to enter into defrost (recirculation) mode 2.6% of the time. The average daily volume of air supplied to the space was 923 m^3 (at T_{ref} = 20 °C), which translates to a volumetric energy use intensity of 12.9 Wh/m^3 of outdoor air.

The seven-week experiment using recirculation as the frost prevention technique displayed that, on average, the SRE of the HRV and ventilation system was 68.7% and 46.0%, respectively. The average daily energy consumption of the supply air heater and HRV was 4.27 kWh, but when considering the intake duct gains, HRV case gains, and exhaust duct gains in the ventilation system, the typical energy consumed to provide outdoor air at 18 °C was actually 14.5 kWh per day, which translates to a daily energy use intensity of 0.60 kWh/day/m^2. The HRV went into defrost (recirculation) mode 20.5% of the

time, reducing the average daily volume of air supplied to the space to 735 m^3 (at T_{ref} = 20 °C), which translates to a volumetric energy use intensity of 19.7 Wh/m^3 of outdoor air.

The ventilation system equipment consumed more absolute energy with electrical preheat than with recirculation as the frost prevention technique. However, when using recirculation as the frost prevention method, the ventilation system experienced more losses throughout the system, causing the operation of the ventilation system to consume more energy. Moreover, considering the quantity of outdoor air that was restricted when using recirculation to prevent frost accumulation, this study shows that electrical preheat is the superior option for this ventilation system design and operation – the energy use of the system with electric preheaters enabled was 35% lower on a per volume of outdoor air basis. Contrary to some belief that preheating is a poor approach for frost prevention in heat/energy recovery ventilators, this research finds that for the tested system design and operation, preheating is a more energy efficient method to provide ventilation.

1 Introduction

A demonstration house that uses a structural insulated panel (SIP) building envelope was built and commissioned in Iqaluit, Nunavut, Canada in April, 2016. Following commissioning of the demonstration house, National Research Council Canada (NRC) instrumented the facility to monitor the performance of the building envelope over multiple years. The in-situ R-value measurements between April, 2016 and April, 2017 are described by Banister, Swinton, Moore, & Krys (2018), while Banister, Swinton, Moore, Krys, & Macdonald (2018) evaluated the energy consumption of the demonstration house during that same period. Monitoring of the building envelope performance during the second year of operation is now in progress.

There are multiple challenges associated with housing in northern Canada. The cost of supplies and labour are increased in remote locations (Roszler, 2005) and the cost of heating is substantially higher due to the increased cost of fuel and very cold climate (Minich, et al., 2011). Furthermore, extremely cold climates result in durability challenges with houses and their systems, which can result in health problems (Minich, et al., 2011). An Inuit Health Survey evaluated the living conditions in a total of 1,901 households and concluded that approximately 40% of the homes were in need of major repairs, 30% were overcrowded and, 20% contained mould (Minich, et al., 2011).

The occurrence of mould growth in these homes suggests a need for better building envelopes and ventilation. The research outlined by Banister et al. (2018) presented a potentially advantageous building envelope solution in cold climates. However, previous research has provided limited experimental data regarding the operation of ventilation systems in extremely cold climates. It is valuable to control ventilation rates effectively and have a system that manages ventilation air heating energy well due to the large difference between outdoor and indoor temperatures during much of the year.

Equipment for air to air heat/energy recovery is an effective method of reducing the cost associated with ventilating buildings (Nasr, Kassai, Ge, & Simonson, 2015). Heat recovery ventilators (HRVs) were used by Kovesi et al. (2009) in Nunavut and successfully allowed for a reduction of indoor air pollutants and in the number of reported respiratory infection symptoms in Inuit children. However, the extreme temperature difference between the outdoor and indoor air temperature can cause problems, such as frost accumulation in the mechanical equipment, which can cause the performance of equipment to degrade (Nasr, Kassai, Ge, & Simonson, 2015) or system failure (Kragh, Rose, & Svendsen, 2005). A related concern is whether ventilation systems are properly installed and operated. Previous studies found that occupants' will turn off this equipment due to the high noise levels (van Holsteijn, Li, Valk, & Kornaat, 2016).

While these challenges are difficult to overcome, they need to be addressed for the living standards and occupant health to be increased in northern communities (Minich, et al., 2011). Therefore, this work addresses this gap in research through the development and implementation of a ventilation system within a demonstration house in Iqaluit which combines effective use of low-cost carbon-dioxide (CO_2) sensors for demand-controlled ventilation, ventilation system modifications to enable more fan speeds, preheaters to reduce the need for defrost cycles when desired, and a supply air heater to achieve satisfactory thermal comfort. This report outlines the development and implementation of the ventilation system that took place between April 2017 and April 2019. Moreover, this report provides experimental data regarding the operation of a ventilation system in an extremely cold climate, an area that has limited content in the literature.

2 Literature Review

Mechanical ventilation systems are necessary to ensure that acceptable indoor air quality (IAQ) is maintained in cold climates (Alonso, Liu, Mathisen, Ge, & Simonson, 2015). Due to the high energy costs of the Arctic it is very beneficial to ventilate on an as-needed basis. For this reason, literature related to residential demand control ventilation strategies was reviewed. In addition, the extreme cold and high energy costs result in several challenges when implementing and operation ventilation systems. The energy demands of ventilation systems without heat/energy recovery are very significant in cold climates (Alonso, Liu, Mathisen, Ge, & Simonson, 2015). Therefore, heat/energy recovery ventilators are necessary to reduce operational costs. However, a further consideration is that the extremely cold climate can cause frosting to occur, leading to system degradation and failure. Thus, previous literature pertaining to the deployment of ventilation systems in cold climates was reviewed.

2.1 Residential Demand Control Ventilation Strategies

Kim & Park (2009) studied eight ventilation strategies that consisted of constant air volume (CAV) and CO_2-based DCV systems, and varying locations of the supply and CO_2 sensors. Simulations showed that the CAV systems were better in terms of initial cost, percent of occupants dissatisfied, and CO_2 concentration. However, the DCV methods were shown to have lower energy costs, and shorter investment payback periods. Cho, Song, Hwang & Yun (2015) evaluated the energy saving potential of a CO_2- and formaldehyde-based DCV system. Compared to continuous ventilation, the proposed ventilation system decreased ventilation and energy use by approximately 50% and 20%, respectively. Guyot, Sherman & Walker (2018) reviewed 38 studies related to smart ventilation systems, which operate based on CO_2, humidity, combined CO_2 and total volatile organic compounds, occupancy, and outdoor air temperature. The literature review demonstrated that ventilation energy savings of up to 60% can be achieved without jeopardizing the IAQ of the space. However, the meta-analysis revealed some studies that showed an increase in energy consumption.

2.2 Ventilation Systems in Cold Climates

Liu, Alonso, Mathisen & Simonson (2016) built and tested a novel quasi-counter-flow membrane energy exchanger during cold operating conditions. The sensible and latent effectiveness values were found to vary from 88.5% to 94.5% and 73.7% to 83.5%, respectively. Nielsen, Rose & Kragh (2009) developed a dynamic model of a counter flow air-to-air heat exchanger in Simulink. The model took into account condensation, frosting and melting to accurately simulate performance in cold climates. Nasr, Kassai, Ge & Simonson (2015) tested two cross-flow heat/energy exchangers during frosting and defrosting cycles. The effect of two defrosting methods (preheating and bypassing) on energy consumption of ventilation in three locations was evaluated. It was determined that preheating the outdoor air consumes less energy than the heat exchanger bypass method. Table 1 provides a summary of nine techniques outlined in the reviewed literature that can be used to prevent frosting and was adapted based on Nasr, Fauchoux, Besant, & Simonson (2014).

Table 1: Frost control strategies in air-to-air heat/energy exchangers, adapted based on Nasr et al. (2014)

Strategy	Description	Advantages	Disadvantages	References
Preheating ventilation air	Ventilation air is preheated prior to the heat/energy exchanger to reduce the formation and accumulation of frost	Simple; Can prevent the occurrence of frost	Not economical for long cold seasons	(Kragh, Rose, & Svendsen, 2005) (Nasr, Kassai, Ge, & Simonson, 2015)
Reducing ventilation air flow rate	Ventilation air is reduced, while the exhaust airflow rate remains the same to defrost the core of the heat/energy exchanger	Simple	Can decrease IAQ; increases building infiltration	(Kragh, Rose, & Svendsen, 2005)
Recirculating return air	Ventilation air is temporarily stopped and the indoor air is recirculated to defrost the core of the heat/energy exchanger	Simple; Defrosting can be enhanced with an increase in flow rate; Suitable for extremely cold climates	No ventilation supply during defrosting	(Beattie, Fazio, Zmeureanu, & Rao, 2017)
Ventilation air bypass	Ventilation air bypasses the heat/energy exchanger either partially or entirely, while the exhaust airflow rate remains the same to defrost the core of the heat/energy exchanger	Simple	Energy recovery decreases during defrosting	(Nasr, Kassai, Ge, & Simonson, 2015) (Kragh, Rose, & Svendsen, 2005)
Reduce effectiveness	Decrease the effectiveness of the heat/energy exchanger by changing the working conditions. For example, changing the wheel speed of an energy wheel. This will reduce frost formation since the warm and humid exhaust air is not cooled as much	Simple;	Defrosting time is longer than other techniques; not applicable in plate heat/energy exchangers; energy recovery decreases during defrosting	
Auxiliary exchanger or double core heat exchanger	Ventilation air alternates between two heat/energy exchangers to either prevent frost formation or allow time for the unused exchanger to defrost	Continuous energy recovery and defrosting	High capital cost for large or redundant equipment; minimal experimental results available under very cold temperatures	(Nielsen, Rose, & Kragh, 2009) (Kragh J., Rose, Nielsen, & Svendsen, 2007)
Changing surface properties of energy wheels	Increasing the moisture adsorption capacity of the energy wheel will reduce the formation of frost	Frosting limit can be significantly reduced by surface material selection	Dependent on many parameters; each design is different; uncertainty in the long term performance and durability of the material	(Liu, Alonso, Mathisen, & Simonson, 2016)

2.3 Summary

The review on literature pertaining to DCV revealed that careful consideration to the control strategy is required to obtain appropriate IAQ and energy savings. CO_2 could have limitations as an indicator of occupancy in residential buildings as the increase in space CO_2 concentration could be minimal due to low occupant densities (a low number of occupants relative to the building volume). Although this could be problematic in residential buildings located in southern regions of Canada, it is believed that the periodic over-crowding associated with northern housing and extremely cold climate make DCV appropriate.

The literature revealed that the cost of defrosting methods can be high, the equipment can be bulky and there is limited experimental data (Zeng, Liu, & Shukla, 2017) and that some defrosting techniques cut off the supply of outdoor air for an extended period of time, defeating the purpose of a ventilation system. As a result, preheating was considered an effective method for preventing frosting in this research due to its low initial cost, non-invasiveness and the added benefit of a continuously available supply of outdoor air. In addition, the ventilation system performance when using electrical preheat was compared to one of the alternative methods that uses recirculation as a defrost technique, instead of electrical preheat.

3 Materials and Methods

This section describes the design of the ventilation system, the controls strategy for the DCV system, preheaters, and supply air heater, and the sensors used to monitor the performance of several aspects of the ventilation system.

3.1 Test Building

The building that the ventilation system was designed for and constructed in is shown in Figure 1. The building is a demonstration house located in Iqaluit, Nunavut, Canada and was built in collaboration with Qikitaaluk Corporation. The demonstration house was initially built to demonstrate the construction technique of a novel panelized construction system in Nunavut and to assess its energy performance in the Iqaluit climate. More detail regarding the initial design and construction of the demonstration house was documented in a report by Banister et al. (2017).

Figure 1: Qikitaaluk Demonstration House in Iqaluit

3.2 Ventilation System Design

The ventilation system was designed with two 1 kW preheaters to ensure an off-the-shelf heat recovery ventilator (HRV) would provide constant ventilation by preventing the restriction of outdoor air due to defrost cycling. The selected HRV is a cross-flow heat exchanger specified to have an apparent sensible effectiveness and sensible recovery efficiency (SRE) from 70% to 78% and 55% to 65%, respectively, depending on the outdoor air temperature and airflow rate. Table 2 shows the expected energy performance specified by the manufacturer.

Table 2: HRV energy performance specifications from the manufacturer

Outdoor Air Temperature (°C)	Net Air Flow (L/s)	Power Consumed (W)	Sensible Recovery Efficiency (%)	Apparent Sensible Effectiveness (%)
0	24	42	65	76
0	30	52	61	70
-25	23	42	60	78
-25	30	53	55	72

To avoid or remove frost build up, the HRV has a factory-set defrost strategy which switches the unit into recirculation mode for an increasing proportion of time as the HRV inlet temperature decreases away from -5 °C. Table 3 shows the standard (factory set) defrost cycles that can be expected at certain outdoor air temperatures.

Table 3: HRV standard (factory set) defrost cycle specifications from the manufacturer

Outdoor Air Temperature	Defrost in Minutes / Air Exchange in Minutes
Above -5 °C	No Defrost
Between -5 °C to -15 °C	6/40
Between -15 °C to -27 °C	5/21
Below -27 °C	8/15

The temperature of the supply air that exits the HRV varies based on the inlet temperatures to the HRV and could be low relative to the indoor air temperature. This can result in cold drafts within the spaces, as was found to be the case by Kragh, Rose, & Svendsen (2005). To ensure a comfortable supply air temperature, a 1 kW electric heater was included on the supply air stream. The intake and exhaust ducts were insulated with R4 reflective insulation to reduce heat gains from the space.

Figure 2 is a schematic of the ventilation system that shows the location of the HRV, two preheaters (EH1 & EH2), supply air heater (EH3), the routing of the ductwork, and airflow paths.

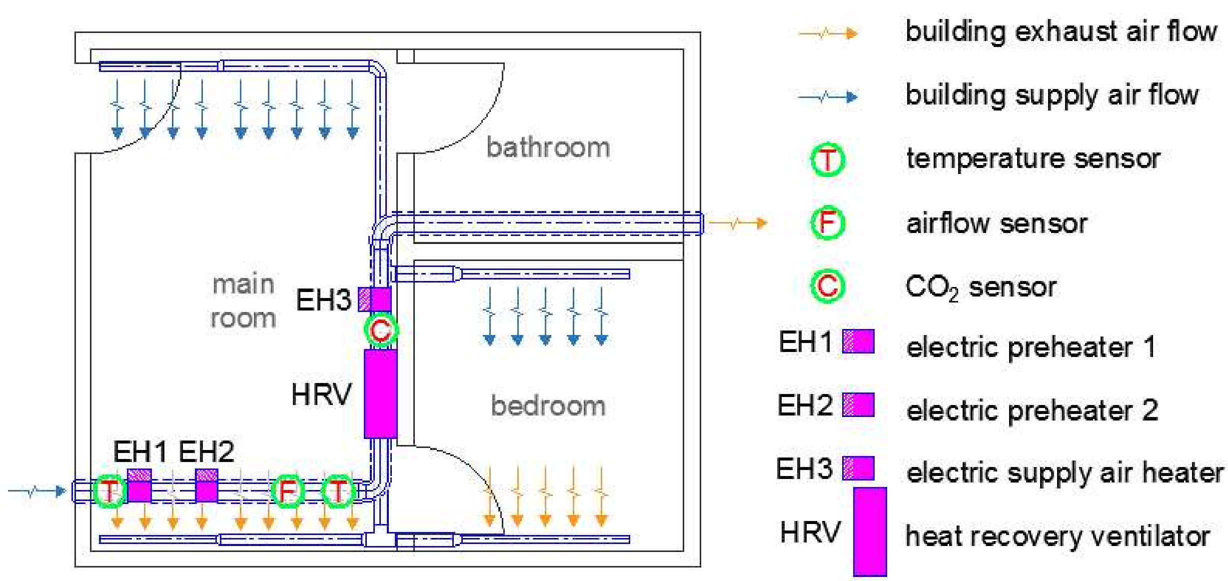

Figure 2: Schematic of the ventilation system in Iqaluit

Figure 3 is an image of the ventilation system in the demonstration house that shows the physical location of the HRV, two preheaters (EH1 & EH2), and supply air heater (EH3), as well as the physical location of the CO_2 (C), temperature (T), and airflow (F) sensor.

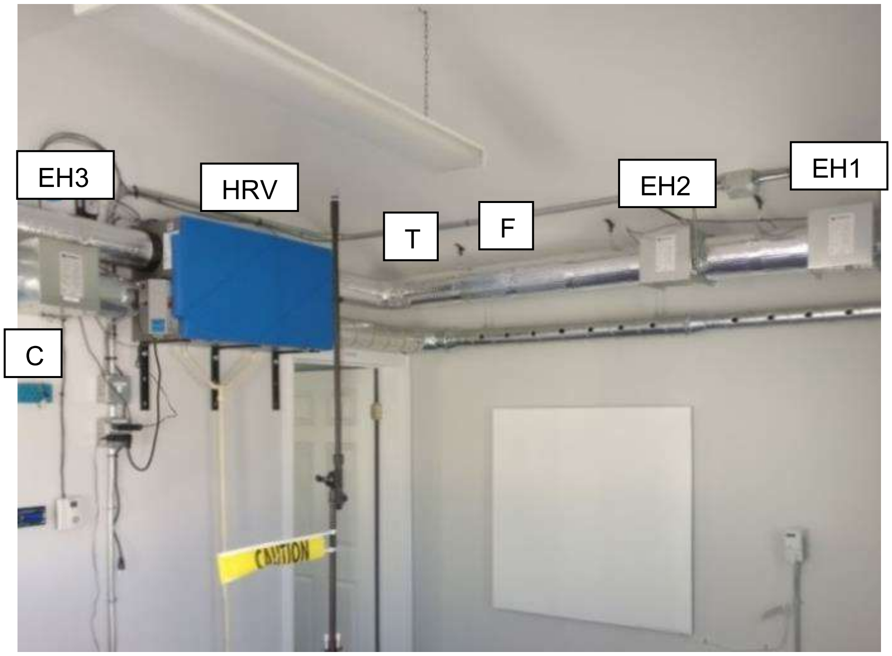

Figure 3: Picture of the ventilation system in Iqaluit

3.3 Ventilation System Controls

Having established the equipment required to achieve desired indoor conditions through ideal ventilation rates, a control strategy was developed for the DCV system and the preheaters.

3.3.1 Demand-Controlled Ventilation Strategy

New HRV fan speeds were enabled to allow the CO_2-based DCV system to meet a wider range of occupant densities, without excessive over-ventilation. The HRV was modified by replacing the standard transformer with an alternative supplied by the manufacturer. A relay control board was also integrated to allow the ventilation system to switch between the speeds of the new transformer. A microcontroller was used to control the ventilation system based on CO_2 concentration measurements in the main room of the building. The microcontroller collects the data from a CO_2 sensor and controls the speed of the fan in the HRV according to the predefined CO_2 concentration thresholds. Table 4 shows the operational modes of the HRV, the corresponding airflow rate, and the CO_2 concentrations that trigger each flow rate when the CO_2 concentration is rising and falling.

Table 4: HRV control modes and corresponding airflow rates

HRV mode	Airflow rate (L/s) at T_{ref} = 20 °C	CO_2 concentration thresholds (ppm)	
		When rising	When falling
Fan speed 1	11.0	-	550
Fan speed 2	15.5	700	850
Fan speed 3	29.0	1000	-

Volumetric flow rate is dependent on the density of the fluid under analysis. For the volumetric flow rate of an ideal gas at a constant pressure, the density of the gas only depends on the operating temperature. As a result, all measured volumetric flow rates need to be converted to a similar reference temperature to ensure accurate findings. Although converting the measured volumetric flow rate to the mass flow rate is a valid alternative method to deal with this issue, common standards and guidelines quantify the outdoor air delivery as a measure of volumetric flow rate. Therefore, measured volumetric air flow rates were converted to a common reference temperature. In this research, the reference temperature was selected as 20 °C, a temperature that is near the middle of common heating and cooling supply air temperature setpoints for buildings.

The CO_2 concentration thresholds corresponding to the airflow rates in Table 4 considered the American Society for Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) guideline of 700 ppm above ambient (ASHRAE, 2016). The ambient CO_2 concentration in 2018 was approximately 400 ppm; therefore, ASHRAE's recommended guideline currently correlates to 1100 ppm. The threshold for speed 3 considers the 1100 ppm guideline and was conservatively selected to ensure that inaccurate sensor measurements do not jeopardize occupant comfort.

3.3.2 Preheater and Supply Air Heater Control Strategy

The analysis of measurements from an existing ventilation system with heat recovery used in single-family houses in Denmark and a test of a standard heat recovery unit in a laboratory have clearly shown that problems occur when the outdoor air temperature gets below approximately –5 °C (Kragh, Rose, & Svendsen, 2005). For this reason, and to avoid the built-in defrost (recirculation) mode of the HRV, the preheater strategy was set to a static minimum temperature of -5 °C.

The supply air heater was controlled using the factory supplied thermostat, which was set to a static value of 18 °C, 2 °C lower than the heating system set point of 20 °C.

3.4 Monitoring of Ventilation System Performance

The ventilation system was equipped with sensors to allow for the performance of components to be monitored. This section outlines the parameters and corresponding sensors that were used to monitor the performance of the CO_2-based DCV strategy and the performance of the HRV, preheaters and supply air heater.

3.4.1 CO_2-based DCV Control Strategy Performance

The performance of the CO_2-based DCV system was monitored during two days with dynamic occupancy, November 26th and 27th, 2018. Occupancy was recorded manually and the operation of the DCV system was monitored using two CO_2 sensors located in the space and an airflow sensor in the intake duct. The HRV mode and airflow measurements were then analyzed to verify if the ventilation system was operating as expected.

3.4.2 HRV Performance

The current and voltage supplied to the HRV was measured by a WattsOn power meter. Six thermocouples located on the HRV's core enabled calculation of the HRV's performance. Figure 4 shows the locations of the thermocouples and outlines the terminology used to describe the quadrants of the core.

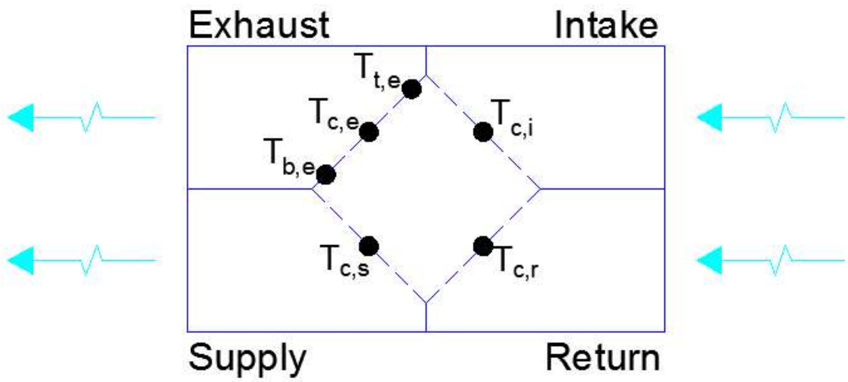

Figure 4. HRV thermocouple locations and core air stream denotations

At the lowest operational airflow rate the Reynolds number is approximately 6,900, which relates to turbulent flow (Re > 4000). Therefore, the airflow through the duct will be turbulent for all operational airflow rates, allowing for the intake and return air streams to be assumed as well mixed. The application of this assumption allowed for one thermocouple to be used to accurately represent the intake and return air temperature within the HRV core. Through the heat exchanger, a temperature distribution arises for each air stream. For this reason, thermocouples were placed on the bottom, center and top of the core's exhaust side. This allowed for a quadratic temperature distribution to be fit to the exhaust air stream to determine the average exhaust temperature. Due to similar air flow rates and temperature gradients, the core's supply side temperature distribution shape was assumed to be the same as the exhaust side, allowing for an approximation of the average supply air stream temperature.

The literature review revealed that it is common to examine the performance of HRVs and ERVs by experimentally determining the corresponding apparent sensible effectiveness. However, the apparent sensible effectiveness of the HRV was not used to assess the performance of the HRV in this research as this performance metric assumes the HRV operates as an isolated system and neglects several heat and mass transfers through the enclosure. This assumption can make the unit look more efficient than it is (PHIUS Passive House Institute US, 2015). Determining the significance of these assumptions by comparing the methods currently used to quantify the performance of an HRV is an area that will be examined in future work. However for the purpose of this research, the performance of the HRV was assessed based solely on its SRE.

CSA-439-18 defines Equation 1 for calculating the sensible heat-recovery efficiency of an HRV during the heating season (CSA Group, 2018). SRE is a metric of the sensible energy recovered by the supply air as adjusted by the fan heat gains, case heat loss or heat gains, case air leakage, and the energy used to defrost, as a percent of the potential sensible energy that could be recovered (CSA Group, 2018).

$$SRE_{HRV} = \frac{\dot{m}_S \times c_p \times (T_s - T_I) - \dot{Q}_{F,S} - \dot{Q}_C - \dot{Q}_L - \dot{Q}_D}{\dot{m}_{max} \times c_p \times (T_R - T_I) + \dot{Q}_{F,E}}$$

Equation 1

where

SRE_{HRV} = sensible recovery efficiency of heat recovery ventilator
$\dot{m}_S$ = mass flow rate of supply air stream (kg/s)
$\dot{m}_{max}$ = the higher mass flow rate between the supply and exhaust air stream (kg/s)
c_p = specific heat capacity of air at average temperature (J/kg·K)
T_s = supply air stream temperature (K)
T_I = inlet air stream temperature (K)
T_R = return air stream temperature (K)
$\dot{Q}_{F,S}$ = rate of energy added to the supply stream by the supply fan (W)
$\dot{Q}_{F,E}$ = rate of energy added to the exhaust stream by the exhaust fan (W)
$\dot{Q}_C$ = energy transfer through unit enclosure (W)
$\dot{Q}_L$ = energy transfer due to casing leakage (W)
$\dot{Q}_D$ = energy transfer used for defrost (W)

Accurately measuring the heat gains and losses in Equation 1 can be difficult; however, by evaluating the control volume and first law of thermodynamics for the two air streams through the HRV, Equation 1 can be simplified. Figure 5 shows a schematic of the thermodynamic process for both air streams through the core of the HRV, as well as the two corresponding control volumes used for this analysis.

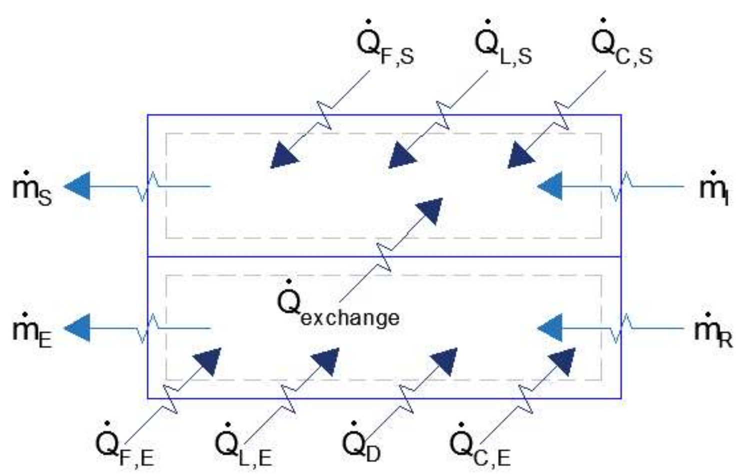

Figure 5: Thermodynamic process and control volume for the two air streams in the HRV

Equation 2 is a general version of the first law of thermodynamics which accounts for all work, heat, and mass transfer entering (in) and exiting (out) a control volume. Equation 2 shows that all parameters entering (in) a given control volume are considered to be positive, while all parameters exiting (out) a given control volume are considered to be negative.

$$\frac{dE_{cv}}{\partial t} = \sum\dot{m}_{in}\left(h_{in} + \frac{V_{in}^2}{2g_c} + \frac{g}{g_c}\right) - \sum\dot{m}_{out}\left(h_{out} + \frac{V_{out}^2}{2g_c} + \frac{g}{g_c}\right) + \sum\dot{W}_{in} - \sum\dot{W}_{out} + \sum\dot{Q}_{in} - \sum\dot{Q}_{out}$$

Equation 2

Equation 3 can be obtained by applying the following assumptions to Equation 2: air behaves as an ideal gas, the process is in steady state, and kinetic and potential energy are negligible.

$$0 = \dot{m}c_p(T_{in} - T_{out}) + \sum\dot{Q}_{in} - \sum\dot{Q}_{out}$$

Equation 3

Equation 4 can be used to describe the thermodynamic process the exhaust air stream undergoes through the HRV by applying Equation 3 to the exhaust air stream control volume.

$$0 = \dot{m}_E c_p(T_R - T_E) + \dot{Q}_{L,E} + \dot{Q}_D + \dot{Q}_{C,E} + \dot{Q}_{F,E} - \dot{Q}_{exchange}$$

Equation 4

Equation 5 can be obtained by rearranging Equation 4.

$$\dot{Q}_{exchange} = \dot{m}_E c_p(T_R - T_E) + \dot{Q}_{L,E} + \dot{Q}_D + \dot{Q}_{C,E} + \dot{Q}_{F,E}$$

Equation 5

Equation 6 can be used to describe the thermodynamic process the supply air stream undergoes through the HRV by applying Equation 3 to the supply air stream control volume.

$$0 = \dot{m}_S c_p(T_I - T_S) + \dot{Q}_{L,S} + \dot{Q}_{C,S} + \dot{Q}_{F,S} + \dot{Q}_{exchange}$$

Equation 6

Equation 7 can be obtained by rearranging Equation 6.

$$\dot{Q}_{exchange} = \dot{m}_S c_p(T_S - T_I) - \dot{Q}_{L,S} - \dot{Q}_{C,S} - \dot{Q}_{F,S}$$

Equation 7

Equation 8 can be obtained by equating Equation 5 and Equation 7 based on their similar term, $\dot{Q}_{exchange}$.

$$\dot{m}_E c_p(T_R - T_E) + \dot{Q}_{L,E} + \dot{Q}_D + \dot{Q}_{C,E} + \dot{Q}_{F,E} = \dot{m}_S c_p(T_S - T_I) - \dot{Q}_{L,S} - \dot{Q}_{C,S} - \dot{Q}_{F,S}$$

Equation 8

Equation 9 can be obtained by rearranging Equation 8.

$$\dot{m}_S c_p(T_S - T_I) - (\dot{Q}_{L,S} + \dot{Q}_{L,E}) - (\dot{Q}_{C,S} + \dot{Q}_{C,E}) - \dot{Q}_D - \dot{Q}_{F,S} = \dot{m}_E c_p(T_R - T_E) + \dot{Q}_{F,E}$$

Equation 9

The summation of the leakage and conduction through the case from the intake and exhaust air stream equal the total leakage and conduction through the case, allowing Equation 9 to be simplified to Equation 10:

$$\dot{m}_S c_p(T_S - T_I) - \dot{Q}_L - \dot{Q}_C - \dot{Q}_D - \dot{Q}_{F,S} = \dot{m}_E c_p(T_R - T_E) + \dot{Q}_{F,E}$$

Equation 10

Equation 11 can be obtained by substituting Equation 10 into Equation 1.

$$SRE_{HRV} = \frac{\dot{m}_E c_p(T_R - T_E) + \dot{Q}_{F,E}}{\dot{m}_{max} c_p(T_R - T_I) + \dot{Q}_{F,E}}$$

Equation 11

If the HRV is balanced then the mass flow rates of the supply and exhaust streams will be equal, allowing for Equation 11 to be simplified to Equation 12.

$$SRE_{HRV} = \frac{\dot{m} c_p(T_R - T_E) + \dot{Q}_{F,E}}{\dot{m} c_p(T_R - T_I) + \dot{Q}_{F,E}}$$

Equation 12

Equation 13 can be obtained by implementing $\dot{m}c_p(T_I\text{-}T_I)$ into the numerator Equation 12, which is effectively equal to 0.

$$SRE_{HRV} = \frac{\dot{m} c_p(T_R - T_E) + \dot{Q}_{F,E} + \dot{m} c_p(T_I - T_I)}{\dot{m} c_p(T_R - T_I) + \dot{Q}_{F,E}}$$

Equation 13

Equation 14 can be obtained by rearranging Equation 13 in a manner that displays a portion of the numerator that equals the denominator.

$$SRE_{HRV} = \frac{\dot{m} c_p(T_R - T_I) + \dot{Q}_{F,E} + \dot{m} c_p(T_I - T_E)}{\dot{m} c_p(T_R - T_I) + \dot{Q}_{F,E}}$$

Equation 14

Equation 15 can be obtained by separating the terms in the numerator Equation 14.

$$SRE_{HRV} = \frac{\dot{m}c_p(T_R - T_I) + \dot{Q}_{F,E}}{\dot{m}c_p(T_R - T_I) + \dot{Q}_{F,E}} - \frac{\dot{m}c_p(T_E - T_I)}{\dot{m}c_p(T_R - T_I) + \dot{Q}_{F,E}}$$

Equation 15

Equation 16 can be obtained by simplifying Equation 15.

$$SRE_{HRV} = 1 - \frac{\dot{m}c_p(T_E - T_I)}{\dot{m}c_p(T_R - T_I) + \dot{Q}_{F,E}}$$

Equation 16

where

SRE_{HRV} = sensible recovery efficiency of heat recovery ventilator
$\dot{m}$ = mass flow rate of supply and exhaust air stream (kg/s)
c_p = specific heat capacity of air at average temperature (J/kg·K)
T_E = exhaust air stream temperature (K)
T_I = inlet air stream temperature (K)
T_R = return air stream temperature (K)
$\dot{Q}_{F,E}$ = rate of energy added to the exhaust stream by the exhaust fan (W)

In this research, it could not be determined whether the HRV was balanced; however, due to the implementation of only one flow sensor it was necessary to assume equal flows between both streams and thus, Equation 16 was used when calculating the SRE for the HRV.

The HRV power data is used to determine when the HRV was operating in defrost (recirculation) mode. The HRV operates at the maximum fan speed when it is in defrost (recirculation) mode, and thus will consume the most electricity at this time. For this reason, it was assumed that the HRV is in defrost (recirculation) mode when it is consuming over 50 W. When the HRV entered into defrost (recirculation) mode the SRE was assumed to be 0 as the ventilation system recirculates air and thus is not recovering any sensible energy. The SRE of the HRV was calculated every minute and averaged to approximate the performance over longer periods of time.

3.4.3 Preheater and Supply Air Heater Performance

The current and voltage supplied to the two preheaters and one supply air heater were measured by a WattsOn power meter, allowing for the power and energy consumption of the HRV to be recorded. In addition to the power and energy consumption information, the outdoor air temperature and air temperature after the electric preheaters are measured, allowing for the functionality of the preheater control strategy to be monitored.

3.4.4 Ventilation System Performance

The performance of the ventilation system was assessed by evaluating its SRE. Similar to that of an HRV, Equation 17 can be used to calculate the SRE of the ventilation system and is a metric of the sensible energy recovered by the supply air as adjusted by the fan heat gain, case heat loss or heat gains, air leakage, cross flow leakage, the energy used to defrost, the duct heat loss or heat gain, the energy used to preheat, and the energy used to increase the supply air temperature after the HRV, as a percent of the potential sensible energy that could be recovered.

$$SRE_{Sys} = \frac{\dot{m}_S c_p (T_S - T_O) - \dot{Q}_{P,S} - \dot{Q}_{F,S} - \dot{Q}_{D,S} - \dot{Q}_C - \dot{Q}_L - \dot{Q}_D - \dot{Q}_{H,S}}{\dot{m}_{max} c_p (T_{Room} - T_O) + \dot{Q}_{F,E} + \dot{Q}_{D,E}}$$

Equation 17

where

SRE_{Sys} = sensible recovery efficiency of the ventilation system
$\dot{m}_S$ = mass flow rate of supply air stream (kg/s)
$\dot{m}_{max}$ = the higher mass flow rate between the supply and exhaust air stream (kg/s)
c_p = specific heat capacity of air at average temperature (J/kg·K)
T_s = supply air stream temperature (K)
T_O = outdoor air temperature (K)
T_{Room} = room air temperature (K)
$\dot{Q}_{F,S}$ = rate of energy added to the supply stream by the supply fan (W)
$\dot{Q}_{F,E}$ = rate of energy added to the exhaust stream by the exhaust fan (W)
$\dot{Q}_C$ = rate of energy added to the supply stream due to heat transfer through casing (W)
$\dot{Q}_L$ = rate of energy added to the supply stream due to cross flow leakage (W)
$\dot{Q}_D$ = rate of energy transferred to the HRV core during defrost, from the space (W)
$\dot{Q}_{P,S}$ = rate of energy added to the supply stream by the preheater (W)
$\dot{Q}_{H,S}$ = rate of energy added to the supply stream by the post HRV heater (W)
$\dot{Q}_{D,S}$ = rate of energy added to the supply stream due to heat transfer/leakage from the duct (W)
$\dot{Q}_{S,E}$ = rate of energy added to the exhaust stream due to heat transfer/leakage from the duct (W)

Similar to monitoring the performance of the HRV, accurately measuring the heat gains and losses required to determine the performance of the ventilation system (Equation 17) can be difficult; however, by evaluating the control volume and first law of thermodynamics for the two air streams through the entire ventilation system, Equation 17 can be simplified. By following the same methodology outlined for the HRV, Equation 17 can be simplified to Equation 18.

$$SRE_{Sys} = \frac{\dot{m}_E c_p (T_{Room} - T_E) + \dot{Q}_{F,E} + \dot{Q}_{D,E}}{\dot{m}_{max} c_p (T_{Room} - T_O) + \dot{Q}_{F,E} + \dot{Q}_{D,E}}$$

Equation 18

Similar to the methodology outline for the HRV, Equation 18 can be simplified to Equation 19, provided the ventilation system is balanced.

$$SRE_{Sys} = 1 - \frac{\dot{m} c_p (T_E - T_O)}{\dot{m} c_p (T_{Room} - T_O) + \dot{Q}_{F,E} + \dot{Q}_{D,E}}$$

Equation 19

In this research, it could not be determined whether the ventilation system was balanced; however, due to the implementation of only one flow sensor it was necessary to assume equal flows between both streams and thus, Equation 19 was used when calculating the SRE for the ventilation system.

The exhaust air stream loses energy due to exhaust duct gains from the space. This needed to be quantified to determine the SRE of the ventilation system. To quantify the heat loss through the duct the exhaust outlet temperature of the building and HRV were required. However, the temperature of the air exiting the ventilation system to the outdoors was not measured during the two test periods under analysis. Instead, the building exhaust outlet temperature was approximated by correlating the exhaust stream temperature rise to the intake stream temperature rise, by assuming lumped capacitance for both streams.

For both Equation 16 and 19, electrical and heat energy are treated equally; however, if fuel is considered to be directly available as the heat source, the calculation would change to account for primary energy consumption of the electricity used.

Table 5 presents a summary of all of the sensors used to monitor the performance of the ventilation system and their purpose.

Table 5: Summary of sensors used to monitor the performance of the ventilation system

Measured Parameter	Sensor	Quantity	Purpose
HRV core temperature: • center of the inlet side • center of the supply side • center of the return side • center of the exhaust side • top of the exhaust side • bottom of the exhaust side	Thermocouples	6	The six thermocouples are used to monitor the performance of the HRV. In addition, the thermocouple on the core's intake side is used to monitor the performance of the preheating control strategy.
Amperage and voltage of the HRV and electrical heaters	WattsOn power meter	4	The four current transducers are used to measure the current supplied to the electrical heater and the HRV. The current measurements are automatically converted to a voltage by the module and used to determine the power and energy consumption of each piece of equipment. This allows for the performance/functionality of the HRV and electrical heaters to be monitored.
Outdoor air supply flow rate	Ampliflow airflow sensor & Setra pressure transducer	1	This measurement allows for the determination of the HRV mode: speed 1, 2, and 3 or defrost (recirculation) mode. It allows for the performance of the HRV (SRE) to be monitored. In this research, it is assumed that there is negligible cross flow leakage in the HRV and that the ventilation system is well balanced, allowing for the outdoor air supply flow rate to be used to calculate the mass flow rate in both the intake and exhaust air streams. In addition, the flow measurement allows for the preheater control strategy and CO_2-based DCV strategy to be monitored.
Space CO_2 concentration	Gravity SEN0219	2	The measurement of the space CO_2 concentration allows for the performance of the CO_2-based DCV strategy to be monitored.

4 Results

The functionality of the demand-controlled aspect of the ventilation system was verified on November 26th and 27th, 2018. Afterward, the performance of the ventilation system was monitored when preheat was enabled for 7 weeks in December 2018 and January 2019 and when electrical preheat was disabled for 7 weeks in February 2019 and March 2019. For these two 7 week periods, the building was unoccupied and the ventilation flow rate was at its minimum according to fan speed 1.

4.1 CO_2-based DCV System Performance Verification

A two-day experiment was conducted on November 26th and 27th, 2018 to verify the functionality of the CO_2-based DCV system. These two days were chosen to verify the functionality of the DCV system because occupancy varied from no occupancy to up to five occupants. Figure 6 shows the measured CO_2 concentration, ventilation rate, and number of occupants present in the space.

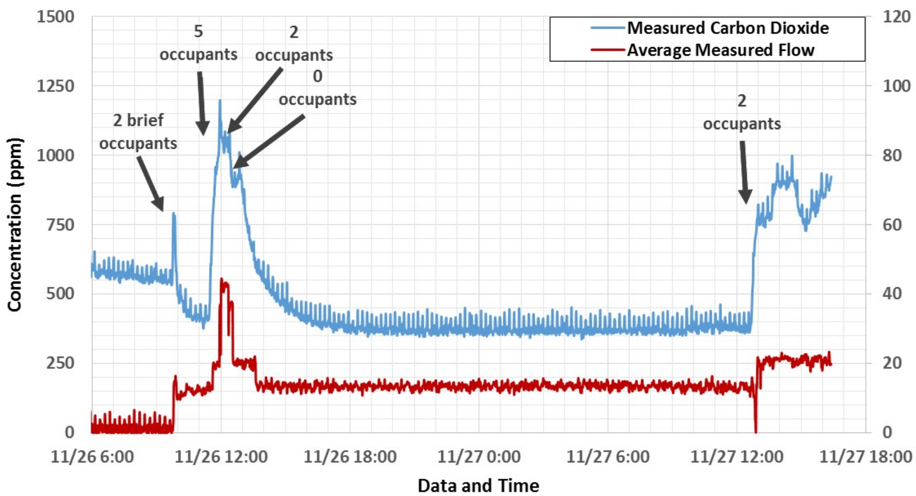

Figure 6: Sample data of demand-controlled ventilation system operation

Initially, the measured CO_2 concentration was above outdoor ambient as a result of previous occupancy and no outdoor air supply. Two occupants briefly entered the space to enable the ventilation system. Their presence caused the CO_2 levels to increase, and enabling the ventilation system caused the HRV to enter fan speed 1 (11.0 L/s of outdoor air). The departure of the occupants, combined with the increase in outdoor air supply, caused the CO_2 concentration to decrease back to outdoor levels.

When five occupants entered the space the CO_2 levels increased, causing the HRV to enter fan speed 2 (15.5 L/s of outdoor air), followed soon after by fan speed 3 (29.0 L/s of outdoor air). The CO_2 concentration stabilized at approximately 1050 ppm until three occupants left. The three occupants leaving, combined with 29.0 L/s of outdoor air supply decreased the space concentration to below 850 ppm, causing the fan to switch to speed 2. After the two occupants left, the CO_2 levels decreased below 550 ppm, which caused the fan to switch to speed 1. Ambient CO_2 concentrations were reached in the space after approximately 4 hours. The next day two occupants entered the space, increasing the CO_2 concentration above 700 ppm. This caused the HRV to enter speed 2, stabilizing the CO_2 levels below 1000 ppm.

4.2 Ventilation System Energy and SRE Performance

The performance of the heat recovery ventilation system with and without preheat was monitored between December 2019 and March 2019. The ventilation system had preheat available for 7 weeks in December 2018 and January 2019. For 7 weeks in February 2019 and March 2019 preheat was not used, instead the ventilation system would use the manufacturer's built-in recirculation method to defrost the core. Measurements were collected during this time period which allowed for the operation of the HRV, preheaters, and supply air heater to be analyzed.

The performance of the ventilation system for each operational mode was evaluated and compared. However, it is not possible to directly compare the results obtained from the in-situ testing since the two 7 week periods endured varying environmental conditions. To account for the varying conditions, the measured performance of the ventilation system with preheat was adjusted to account for the difference in outdoor air temperature during the two 7 week periods. Since the setpoint of the electrical preheaters remained constant throughout the 7 week period only the energy performance of the preheaters will change due to the varying outdoor air temperature. As a result, the performance of the ventilation system when using preheat was adjusted by scaling the energy consumption of the two electrical preheaters based on the temperature difference between the preheater setpoint and the measured outdoor air temperature for both test periods. In this report the original and adjusted preheat measurements will be presented; however, all comparisons between preheat and recirculation will be completed using the temperature adjusted values for the preheat methodology.

4.2.1 Energy performance monitoring

Table 6 shows the average daily energy consumption that was recorded during the 7 weeks of operation with and without outdoor air preheating.

Table 6: Average daily energy consumption of ventilation system components

	Average outdoor air temperature (°C)	HRV (kWh/day)	Preheaters (kWh/day)	Supply air heater (kWh/day)	Total (kWh/day)
Preheat (adjusted)	-28.0 (-29.4)	0.55 (0.55)	3.42 (3.88)	2.54 (2.54)	6.51 (6.72)
Recirculation	-29.4	0.67	0.00	3.60	4.27

Although 7 weeks of monitoring data is available, one week will be presented graphically for better clarity. The first week of December and the third week of February were selected for analysis of the system during times with and without preheat, respectively. These weeks were selected because they include outdoor air temperature fluctuations. Evaluating the functionality of the ventilation system during periods of changing outdoor air temperatures allows for a better understanding of the relationship between the parameters of interest and the outdoor air temperature. Figure 7 and Figure 8 show the energy consumption of the preheaters, supply air heater, and the HRV, as well as the HRV inlet and outdoor air temperature during the first week of December, when preheat was available and during the third week of February, when preheat was not available, and instead recirculation was used as a defrost technique. The energy data provided in Figure 7 and Figure 8 consist of a 10-minute moving average of 1-minute data.

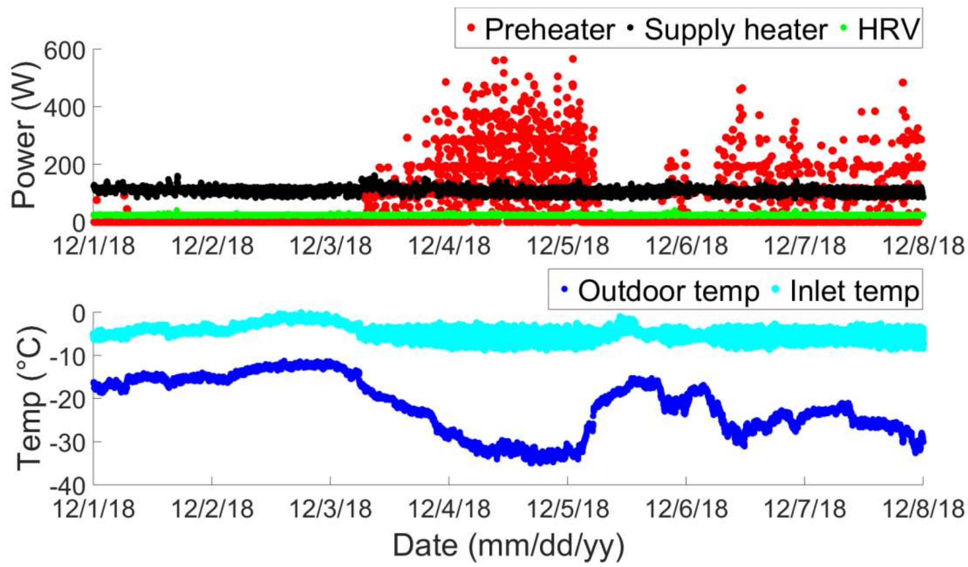

Figure 7: Energy consumption of ventilation system with preheat and temperatures of outdoor air and inlet

Figure 7 shows the supply air heater constantly provided heat to maintain a supply air temperature of 18 °C. During the first week of December, the HRV only operated in fan speed 1, the lowest available outdoor air supply rate. However, in January even though preheat was available the HRV operated in defrost (recirculation) mode a few times and therefore operated at its highest fan speed. Figure 7 shows the preheater control strategy was functional; however, minimal preheat was used during the week. Figure 7 shows that as a result of an increase in outdoor air temperature, the preheaters stopped supplying heat to the outdoor air. The increase in outdoor air temperature did not exceed -10 °C. However, the outdoor air was warmed up by the space through the duct before the HRV. This was a result of the low 11.0 L/s average flow rate (at T_{ref} = 20 °C) and 3 m length of outdoor air duct, which was found to heat the outdoor air by as much as 12 °C without the use of a preheater.

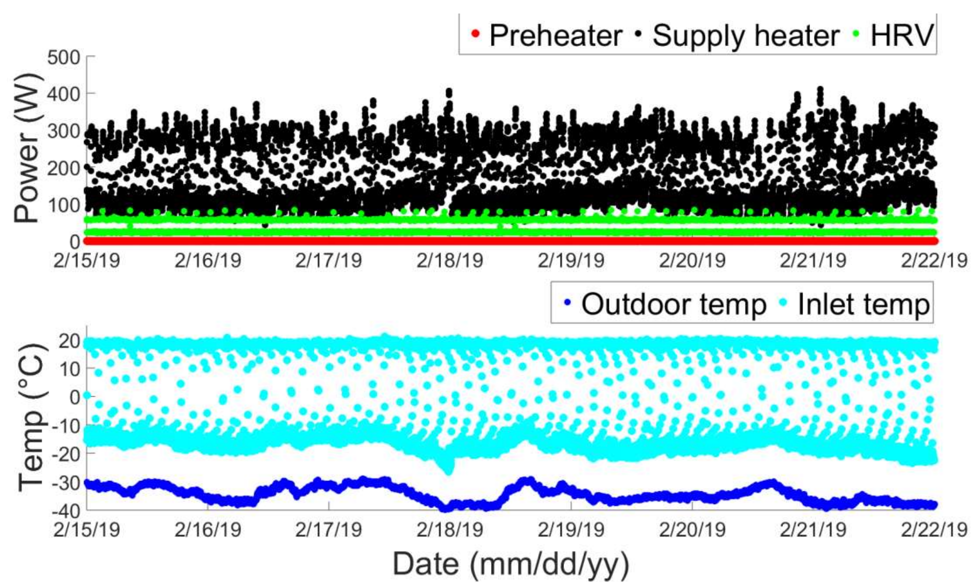

Figure 8: Energy consumption of ventilation system without preheat and temperatures of outdoor air and inlet

Figure 8 shows that without preheat there was a significant rise in temperature between the outdoor inlet and the HRV inlet. As mentioned earlier, this was a result of the low 11.0 L/s average flow rate (at T_{ref} = 20 °C) and 3 m length of intake duct that was required to accommodate the design of the building. In addition to this temperature rise, the HRV inlet temperature sometimes increased to approximately 20 °C. This was a result of the HRV recirculating air from the space, which was maintained at 20 °C, through the HRV core's inlet. The HRV operated almost exclusively in alternation between defrost (recirculation) mode and delivering outdoor air, causing the electricity consumption of the HRV to vary. When the HRV is operating at its low flow rate it consumes 24 W. For the periods when the HRV operates in defrost (recirculation) mode, the maximum flow rate is used and electricity consumption increases to 57 W. In addition, the supply air heater power consumption fluctuated based on the HRV operation. Similar to when the ventilation system was operating with electrical preheat, the supply air heater was providing 100 W to the air when the HRV operated in ventilation mode. However, when the HRV operated in defrost (recirculation) mode the supply air heater provided 300 W to the recirculation air to make up for the energy lost inside the HRV's core.

4.2.2 Sensible recovery efficiency monitoring

Table 7 shows the average sensible recovery efficiency (SRE) of the ventilation system and HRV and the outdoor air delivery rate of the ventilation system during the 7 weeks of operation with and without outdoor air preheating.

Table 7: Sensible recovery efficiency of HRV and ventilation system

	Average outdoor air temperature	SRE of HRV	SRE of Ventilation System	Outdoor air delivery rate
Preheat (adjusted)	-28.0 °C (-29.4 °C)	70.1% (70.1%)	39.6% (38.4%)	97.4% (97.4%)
Recirculation	-29.4 °C	68.7%	46.0%	79.5%

Although 7 weeks of monitoring data is available, one week will be presented graphically for better clarity. The same periods as before, the first week of December and third week of February, were selected for analysis. Figure 9 and Figure 10 show the SRE of the ventilation system and HRV, and the HRV inlet and outdoor air temperature for the first week of December, when preheating was available, and for the third week of February, when preheating was not available. The SRE of the ventilation system and HRV was not calculated when the HRV was operating in defrost (recirculation) mode (i.e., when there was no outdoor air supply) to ensure validity and accuracy of the calculations; instead, values at these times are displayed as 0 in Figure 9 and Figure 10.

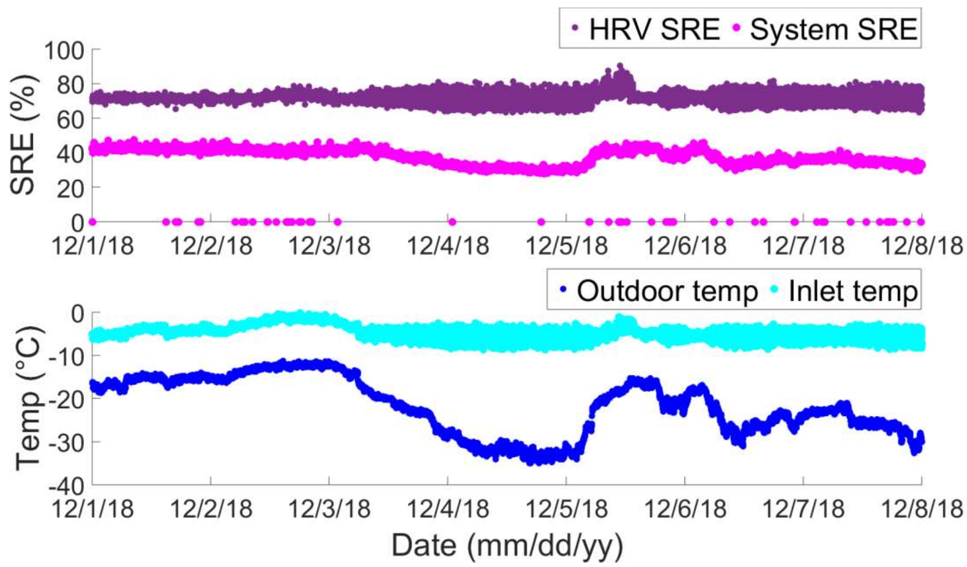

Figure 9: SRE of the ventilation system and HRV with preheat

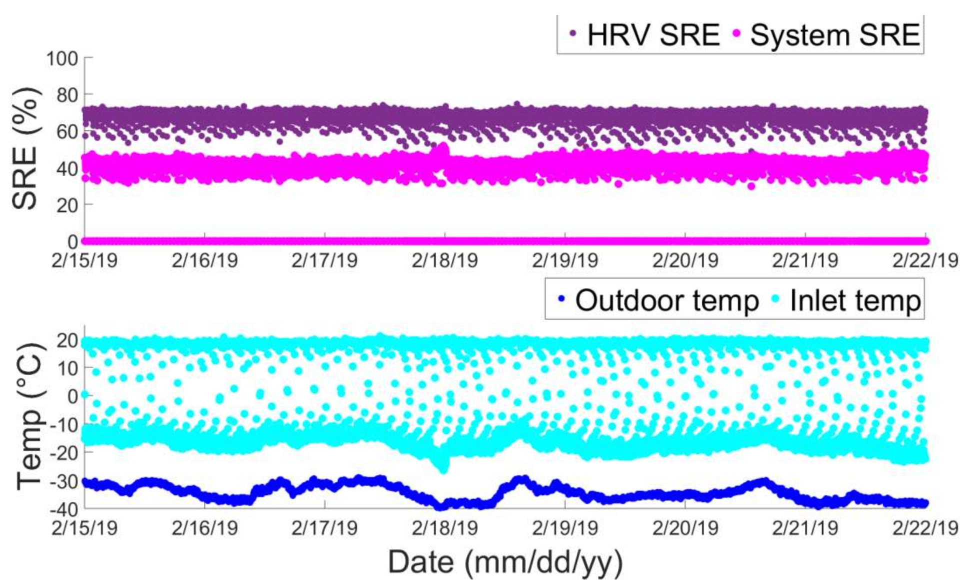

Figure 10: SRE of the ventilation system and HRV without preheat

Figure 9 and Figure 10 show a noticeably lower ventilation system SRE compared to the SRE of the HRV. The large discrepancy is a result of the extremely low outdoor air supply temperatures causing excessive energy added to the system via duct heat gains, HRV case heat gains, the supply air heater, and the preheaters (when operating). The SRE of the overall ventilation system when operating with preheat has more variation than the SRE of the HRV due to the preheater decreasing the ventilation system SRE. Figure 9 shows this trend and that the SRE of the ventilation system is dependent on the outdoor air temperature as a result. While Figure 10 does not display the same dependency of the SRE of the ventilation system on the outdoor air temperature, it does still depend on it somewhat, as a result of the intake duct gains from the space.

4.2.3 Results Summary

Figure 11 displays the energy source distributions that were measured throughout the ventilation system as a percentage of the total energy required to heat the outdoor air to the supply air temperature when using either preheat or recirculation as the frost prevention method. This figure shows the average measured energy flows as the process that the outdoor air gains its energy from the point that it enters the ventilation system (left) to the point that it exits the ventilation system and enters the space (right). Figure 11 also includes the energy that is lost through the exhaust stream as a result of either the HRV case or exhaust duct. These two values are displayed in parallel with the energy recovered by the HRV as they are influenced by the operation of the HRV and offset the energy recovered by the HRV.

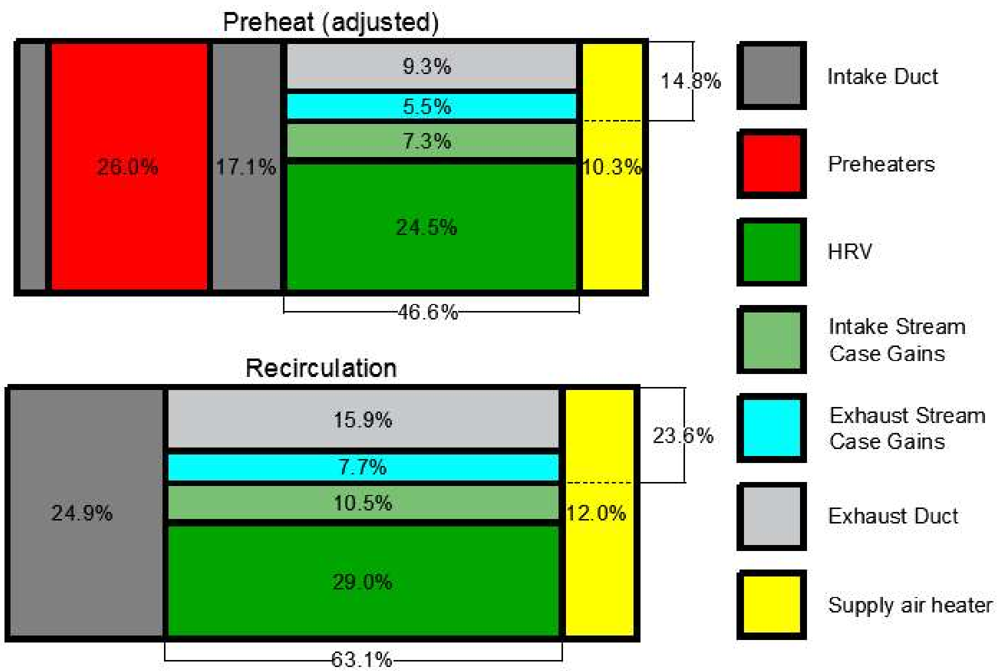

Figure 11: Average corresponding energy flow for ventilation system using preheat and recirculation

The average energy added by the preheaters was determined based on the measured power data and grouped together for visualization purposes. The energy addition through the intake duct work was determined based on available measurements: the outdoor air temperature, HRV core inlet temperature, and air flow rate, while also considering the input from the electrical preheaters. When using preheat, it was determined that the supply air gained a total of 17.1% of its energy through the intake duct from the space. In reality, a portion of the intake duct gains occur before the preheaters, as well as after. This is reflected in Figure 11; however, for clarity the 17.1% is displayed solely on the portion after the preheaters. The average energy gained by the supply air stream through the HRV was calculated using the air flow rate measurement and HRV supply and intake temperature measurements. This value is displayed below each schematic: 46.6% and 63.1% when using preheat and recirculation, respectively. These are valuable amounts of heat recovery by the HRV; however, a portion of the supply air stream gains its energy from the space, through the HRV casing. Using a control volume around the HRV based on the four measured temperatures for each stream and the air flow rate, the total energy transferred through the casing was determined. By applying a ratio of the measured temperature differences between each stream and the space, the portion of energy transferred to the supply and exhaust streams could then be approximated. The portion that is transferred to the exhaust stream is lost to the outdoors, whereas the portion that is transferred to the supply stream is brought back into the space. Although it is not lost out of the building, the space heating system does need to make up for this energy use and thus, it is still energy consumed to operate the ventilation system. In addition, the exhaust stream also loses energy due to exhaust duct gains from the space. The average energy added by the supply air heater was determined based on the measured power data; however, the supply air heater was determined to have consumed more energy than was required for both scenarios. This was likely due to unsteady flow and temperature control. Therefore, for the purpose of visualization in this figure it was assumed that the supply air heater only consumed the energy required to increase the supply air temperature to 18 °C. The remaining energy was assumed to be recovered by the HRV. The reason the recovered energy in Figure 11 does not align with the aforementioned SRE is because the SRE accounts for exhaust losses in the denominator, whereas this figure deducts the energy lost through the exhaust on the numerator.

4.2.3.1 **System performance with electrical preheat for frost prevention**

The system operated well during the 7 week period using electrical preheat to prevent frost accumulation in the HRV. The preheaters consistently maintained HRV inlet temperatures around -5 °C, resulting in an average of 3.88 kWh of daily heating energy after adjusting for outdoor temperature. The heat addition of 3.88 kWh per day by the preheaters only added 26.0% of the total energy gained by the outdoor air supply. However, as a result of the length of the intake duct, the outdoor air supply gained an additional 17.1% of heat energy prior to entering the HRV. Therefore, outdoor air supply heating prior to the HRV accounted for 43.1% of the total energy gained by the outdoor air supply.

Although preheat was used, the HRV entered defrost (recirculation) mode 2.6% of the time, which shows the preheaters allowed the inlet temperature to drop below -5 °C, as the HRV's defrost settings initiate at temperatures below -5 °C. The reason for this was a combination of unsteady airflow and fluctuating control of the preheaters from the PID loop response. The energy consumed by the HRV was much lower than when operating without electrical preheat, at an average of 0.55 kWh per day. To limit drafts of cool air in the building, the supply air was heated to 18 °C after the HRV, resulting in 2.54 kWh average daily heating energy use. In addition to the supply air heater, the heat gains through the HRV enclosure and duct work after the HRV increased the outdoor air supply temperature. While the outdoor air supply temperature did increase by 46.6% of the total required to reach 18 °C, 14.8% of the total energy required was then lost by the ventilation system through the exhaust duct gains and by the exhaust stream gaining energy through the HRV casing at the cost of the building's heating system. In addition, 7.3% of the total energy required was obtained by the supply air stream through the HRV casing. As a result, the HRV only recovered 24.5% of the total energy required to increase the outdoor air supply temperature to 18 °C.

Although the three electrical heaters and HRV fan only consumed 6.72 kWh per day, when considering the intake duct gains, HRV case gains, and exhaust duct gains in the ventilation system, the typical energy consumed to provide outdoor air at 18 °C was actually 11.9 kWh per day. The average daily volume of air supplied to the space was 923 m^3 (at T_{ref} = 20 °C) during the 7 week period with electrical preheat. The total energy consumed by the electric heaters to temper and supply the building's outdoor air was 7.3 Wh/m^3 during the 7 week testing period. However, the total energy consumed by the ventilation system was 12.9 Wh/m^3 when considering the additional energy that was consumed by the ventilation system, as a result of the intake duct gains, HRV case gains, and exhaust duct gains. Ultimately, when using preheat to prevent frost accumulation, the HRV recovered sensible energy at an efficiency of 70.1%, which translated to a ventilation system sensible recovery efficiency (adjusted) of 38.4% because of energy additions elsewhere in the ventilation system.

4.2.3.2 System performance with recirculation for frost prevention

When the ventilation system operated using recirculation to prevent frost accumulation, the HRV went into defrost approximately 20.5% of the time. This percentage of time was less than what would be expected when considering the average outdoor air temperature during the 7 week period was -29.4 °C. The manufacturer's defrost strategy consists of 8 minutes of defrost per every 15 minutes of outdoor air supply at temperatures below -27 °C. Therefore, operating closer to 30% of the time in defrost (recirculation) mode would be expected. However, even though the ventilation system did not use electrical preheat, the duct heat gains from the space acted as a means of preheat for the outdoor air supply. This undesired means of preheating the outdoor air supply accounted for 24.9% of the total heat energy added to the outdoor air by the ventilation system. This finding confirms the need for proper installation of a high level of insulation on the intake duct in cold climates. In addition, better insulation should be installed on the exhaust duct to avoid unnecessary system losses, which are exacerbated in extremely cold climates. Moreover, the lengths of these ducts should be minimized as much as possible. Following completion of this portion of the study, duct insulation was increased from R4 to R10.

The energy consumed by the HRV was higher when operating without electrical preheat, at an average of 0.67 kWh per day. This was a result of entering defrost (recirculation) mode more frequently, causing the fan speed to increase. To limit drafts of cool air in the building, the supply air was heated to 18 °C after the HRV, resulting in 3.60 kWh average daily heating energy use. In addition to the supply air heater, the heat gains through the HRV enclosure and duct work after the HRV increased the outdoor air supply temperature. While the outdoor air supply temperature did increase by 63.1% of the total required to reach 18 °C, 23.6% of the total energy required was then lost by the ventilation system through the exhaust duct gains and by the exhaust stream gaining energy through the HRV casing. In addition, 10.5% of the total energy required was obtained by the supply air stream through the HRV casing at the cost of the building's heating system. As a result, the HRV only recovered 29.0% of the total energy required to increase the outdoor air supply temperature to 18 °C.

Although the three electrical heaters and HRV fan only consumed 4.27 kWh per day, when considering the intake duct gains, HRV case gains, and exhaust duct gains in the ventilation system, the typical energy consumed to provide outdoor air at 18 °C was actually 14.5 kWh per day. The average daily volume of air supplied to the space was 735 m^3 during the 7 week period without electrical preheat. The total energy consumed by the electric heaters to temper and supply the building's outdoor air was 5.8 Wh/m^3 during the 7 week testing period. However, the total energy consumed by the ventilation system was 19.7 Wh/m^3 when considering the additional energy that was consumed by the ventilation system, as a result of the intake duct gains, HRV case gains, and exhaust duct gains. Ultimately, when the ventilation system did not use electrical preheat to prevent frost accumulation the SRE of the HRV was 70%, which translated to a ventilation system SRE of 46.0%.

5 Discussion

This section begins with a discussion of the performance of the demand-controlled aspect of the ventilation system and the benefits of its implementation in ventilation systems in the North. Afterwards, the performance of the ventilation system with and without electrical preheat is discussed and the advantages and disadvantages of each method are outlined.

5.1 CO_2-based DCV System Performance

The two-day testing of the DCV system proved the viability of using low-cost CO_2 sensors for ventilation control in a small residential dwelling. Further testing will be conducted on the DCV system's performance; however, the two-day study sufficed for testing the DCV strategy and showed that if occupants enter the space, sufficient ventilation will be supplied. The detection of occupancy is important to ensure appropriate ventilation is supplied to the space and for energy conservation purposes. CO_2-based DCV for residential applications has had some challenges in past work as a result of low occupant densities being difficult to detect; however, CO_2-based DCV is more easily justified in the Arctic, as occupant densities can be much higher in the north due to housing shortages and the greater need for shelter from the extreme climate.

5.2 Ventilation System Energy and SRE Performance

Although it is often thought that preheat consumes more energy than other frost prevention methodologies, experimental results showed that when using electrical preheat (adjusted) and recirculation, the ventilation system consumed 12.9 Wh/m^3 and 19.7 Wh/m^3 of outdoor air, respectively. This finding indicates that preheating is a more energy efficient method to provide ventilation if controlled well. However, the results display that the performance of the two methods to prevent frost accumulation depend on the entire ventilation system design or operation. For example, a system designed with different duct insulation levels or duct lengths, as well as a ventilation system operated with a different flow rate or in a different climate could impact the performance of these two frost prevention methods. Determining how various frost prevention methods perform with various ventilation system designs or operation is an aspect that needs to be explored more in the future.

When preheating, the daily energy consumption of the preheaters, supply air heater, and HRV was approximately 6.72 kWh (temperature adjusted). Without preheating, the daily energy consumption of the supply air heater and HRV was approximately 4.27 kWh. However, when considering the additional energy that was consumed by the ventilation system, as a result of the intake duct gains, HRV case gains, and exhaust duct gains, the daily energy consumption with electrical preheat (adjusted) and recirculation was 11.9 kWh and 14.5 kWh, respectively. The absolute energy consumption of each system operation, when considering the intake and exhaust duct heat gains, as well as the HRV case gains are similar in part due to undesired preheating of the outdoor air by the space, when the ventilation system does not use electrical preheat. Although this amount of energy addition is a result of an intake duct length that is longer than ideal to accommodate the flow sensor and HRV mounting within the building layout. The results display the importance of minimizing the intake duct length and insulating it to a high degree. This finding should also be considered and applied to the exhaust duct.

If the intake duct length was reduced or the insulation on the duct was improved, the daily energy consumption of the ventilation system when using recirculation would be reduced. However, the system would supply even less air without preheat due to increased defrost cycling. In an ideal situation, the space would not lose any energy to the outdoor air duct, which would allow for the HRV to recover more energy from the exhaust stream. This would mean that the undesired heat energy that preheated the air (24.9% of the overall energy) would be partially absorbed by the HRV. The rest would be made up by the supply air heater after the HRV. However, limiting the intake duct gains impacts the performance of the rest of the ventilation system as well. The HRV inlet temperature would be lower, in turn causing the supply air temperature exiting the HRV to be lower. This would cause the supply air heater to consume more energy to heat up the supply air to 18 °C. The lower HRV inlet temperature would also cause the HRV case gains to increase. In addition, the exhaust air temperature of the HRV would be lower, leading to a higher temperature gradient between the exhaust and room air temperature. The higher temperature gradient would drive more heat transfer through the exhaust duct from the room, ultimately increasing exhaust duct losses.

The purpose of the ventilation system is to provide the building and occupants with outdoor air. Relying on recirculation as a defrost technique cuts the supply of outdoor air for a significant proportion of time, which goes against the purpose of the ventilation system. For periods of high occupancy or humidity, preheating is an important technique to ensure sufficient ventilation can be delivered to a space to maintain acceptable IAQ. Preheat also has the added benefit of reducing or avoiding cycles of condensation or frost build-up and removal from the core, which could result in less required maintenance. Although the noise levels were not measured in this study, another reason for using preheating instead of recirculation is that it reduces noise from the ventilation system. For periods of moderate occupancy, it is likely more preferred by occupants for the fan to operate at a medium speed rather than the maximum. If preheat is not used, the HRV fan speed will alternate to a maximum flow rate during recirculation, which can occur every 23 minutes for the particular HRV model used for this study. These frequent alternations in fan speed and high levels of noise during recirculation could lead occupants to disable the ventilation system. Additionally, a system relying on recirculation for defrost would be operating at higher fan speeds during ventilation at all occupancy levels, whereas the system using preheat would be able to run at lower fan speeds at the same occupancy levels.

The obtained experimental results show the value of using preheat as a frost prevention methodology. However, given the excessive cost of electricity in the north, the use of electrical preheaters is not a practical solution without renewable sources. Instead, hydronic-based preheaters could be used and would be a more economical solution where and while fuel oil remains the main source of energy, if controlled effectively.

6 Conclusion

An existing demonstration house in Iqaluit was equipped with an experimental DCV system consisting of an off-the-shelf heat recovery ventilator, two preheaters, a supply air heater, and a carbon-dioxide sensor, temperature sensors, and a microcontroller for custom control. The ventilation system was instrumented with thermocouples, an airflow sensor, and current transducers to monitor ventilation system performance. While long-term monitoring is ongoing, the short-term experimental data provided in this paper demonstrated the viability of low-cost residential DCV.

Although it is often thought that preheat consumes more energy than other frost prevention methodologies, experimental results showed that this method can be acceptable if controlled well. It was experimentally shown that preheating consumed approximately 18% less energy daily (11.9 kWh/day compared to 14.5 kWh/day), than recirculating to defrost the HRV. Moreover, as a result of recirculation restricting outdoor air for approximately 20% of the time, the energy use of the ventilation system with electric preheaters per unit of outdoor air was actually 35% lower (12.9 Wh/m^3 compared to 19.7 Wh/m^3). However, the results also display that the performance of the two methods to prevent frost accumulation depend on the entire ventilation system design and operation, an aspect that needs to be explored more in the future.

Relying on recirculation as a defrost technique can cause the outdoor air supply to be cut off for a high proportion of time, which goes against the purpose of the ventilation system. For periods of high occupancy or humidity, preheating is an important technique to ensure sufficient outdoor air can be delivered to a space to maintain acceptable IAQ. Preheat also has the added benefit of reducing or avoiding cycles of condensation or frost build-up and removal from the core, which could result in a cleaner core between maintenance. Moreover, it is important to design building systems as efficient as possible; however, these systems must function and effectively serve their purposes. In the case of a ventilation system, the purpose is to provide outdoor air to the building and defrosting reduces the proportion of time when outdoor air is delivered, possibly to a point that is unacceptable to the occupants.

7 FUTURE WORK

The development of efficient and comfortable residential buildings in cold climates will continue to be a focus; comparing different defrost mitigation techniques, testing of various preheater equipment and control strategies, and adding sensors for additional testing and monitoring of ventilation components and system performance will be completed in the future.

7.1 Comparison of defrost mitigation techniques

Additional experimentation will be conducted to evaluate the performance of two frost prevention strategies at various outdoor airflow rates. The work will consist of a comparison of using preheat to prevent frost build up and recirculation to defrost. The energy consumption of each methodology per unit of outdoor air delivered to the space will be used to measure the performance of each technique at the respective operational outdoor air rate. How the performance of each technique varies based on the operation outdoor air rate will allow for the techniques to be compared on an application basis. For example, if one strategy performs better at low outdoor air flow rates it will be better for small residential applications with minimal occupants. If it performs better at high outdoor air flow rates, it will be more applicable for buildings with higher occupant densities. Alternatively, one technique could be proven to be a more effective technique at all operational flow rates.

7.2 Preheater equipment/control strategy

In the future, measured indoor CO_2 concentrations could be used to influence the setpoint of the preheaters. When the building is unoccupied and measured CO_2 concentrations are low, the preheaters could be disabled and a lower effective outdoor air delivery rate could be accepted, i.e., permit recirculation. When occupants are present, the preheater setpoint would be re-enabled, avoiding the need for defrost cycles. This would restore the effective outdoor air delivery rate to the supply air rate, i.e., avoid recirculation. Moreover, noise levels would be reduced during occupancy.

This strategy allows for the effectiveness of the HRV to be maximized and the energy consumption minimized, as the air will only be preheated when the space is occupied and frosting may occur. During times of no occupancy, the relative humidity indoors will be very low as the indoor air will have similar moisture content as the outdoor air. As a result, the exhaust air may not reach saturation at times, and thus, the preheater will not need to temper the outdoor air to ensure frost build up does not occur.

In addition, ventilation system design and control with hydronic pre- and post-heaters should be explored, as this is presently the only viable method for heating in many northern communities due to a reliance solely on fuel oil for heating and electricity generation.

7.3 Additional sensors for performance monitoring

The ventilation system was equipped with several sensors that allowed for the performance to be monitored. Most data was collected through a data acquisition (DAQ) system, allowing for the data to be collected remotely. However, some data was used to control the ventilation system and was stored locally via an SD card. Given the remoteness of the location it is extremely important to implement some redundancy to these measurements and collect the data through the DAQ system. Thermocouples will need to be added in the outdoor air intake stream, before and after the preheaters, before and after the supply air heater, and in the exhaust and return air stream.

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