Investigation of prototype membrane based energy exchanger

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Abstracts

In order to minimize the energy use for heating, passive houses for residential use are constructed using heavy insulation. In addition, they have minimal air leakages and no vents in exterior walls for direct supply of fresh air, and thus, mechanical ventilation systems are a mandatory requirement in such buildings.

With the aim of reducing energy use, efficient energy recovery from used air will be of high importance. In residential buildings with several living units, centralized air handling units are regarded as the most energy efficient system. However, to prevent odours to transfer between apartments it is important to avoid carryover leakages of pollutants between the exhaust air and the supply air inside the heat exchanger.

Rotary heat exchangers (heat wheels) are very energy efficient (85 %), but have the drawback of transferring odours from exhaust air to fresh supply air. To avoid transfer of odours in apartment buildings, flat plate heat exchangers are commonly used instead. Nevertheless, the state-of-the-art flat plate heat exchangers may not handle properly water condensation and frost formation at low supply inlet temperatures. To avoid this problem, the efficiency must be reduced on cold days, causing an increase in yearly energy use for air heating.

An alternative to the flat plate heat exchanger are the so called quasi-counter flow membrane-based heat and mass recovery exchangers. In a membrane based exchanger, moisture is transferred from the humid exhaust air to the dry supply air. In this way, condensation and frosting should be avoided at the exhaust air side. In this work, a membrane energy exchanger was compared to a thin non vapour permeable plastic foil heat exchanger. The study focused on verifying condensation and freezing problems and how the membrane energy exchanger performs.

To compare the different plate materials, a test rig was built in the laboratory at the Department of Energy and Process Engineering at NTNU. The experiments showed that non permeable heat exchangers have problems with condensation and freezing during the tested conditions, such as metal plate heat exchanger experience in real conditions. For the same conditions, the membrane based exchanger did not experience the same problems. Yet, additional problems with swallowing of the membrane in high humidity conditions showed that the tested membrane type had drawbacks and needs further development to become commercially applicable.

Keywords: Sensible Heat, Latent Heat, Net-Zero Energy Buildings (NZEBs), Cold Climate, Thermal Comfort, Energy Recovery

Introduction

Reduced energy consumption is one of the most cost-effective ways of reducing CO_2 emissions from combustion of fossil fuels. Worldwide, the building sector accounts for 40 % of mainland energy use [EERE, 2009]. Airtight and well insulated houses are needed in order to reduce uncontrolled air infiltration and loss of thermal energy. Ventilation is required for removing or diluting airborne pollutants that even in low concentrations may become irritating or hazardous to humans. The main function of ventilation systems is to remove stale air from a room and to supply fresh air. For relatively cold climates such as the Nordic climate, mechanical ventilation systems are the state-of-the-art solution even in residential buildings for ensuring air quality, thermal comfort and reduced energy use [ASHRAE, 1997.]. Heat losses in ventilation systems without heat recovery are very significant in cold climates. In order to achieve a further reduction in energy use, the focus must be set on high energy efficient ventilation and heat recovery [Steimle, 1992]. Estimates show that 70 % of the heat lost through mechanical ventilation systems can be recovered by using heat recovery systems [Aristov, 2008]. This percentage may increase when recovering latent and sensible heat from exhaust air.

NZEB buildings

Net Zero Energy Buildings (NZEBs) lack a definition or common understanding. However, the guidelines to be followed in the definitions of suitable recovery heat exchangers in ventilation systems for NZEB, will consider that:

1. NZEBs aim at reducing the energy demand through technology measures,
2. NZEBs aim at reducing the energy demand though a better use and operation and
3. NZEBs aim at substituting non environmental-friendly energy sources in favour of renewable energies.

The present study relies on the two first conditions and is focused on NZEB apartment buildings placed in Nordic countries whose ventilation system is common and shared by all the apartments within a given building. The ideal air-to-air energy exchanger for use in NZEBs in Nordic countries has the requirements of high ventilation effectiveness and efficiency, induce proper IAQ and avoid odours spreading. These are major evaluation parameters since the first one affects the emissions share, while the second and third ones alter the habitability of the apartment.

Theoretical background

Frosting in heat exchangers in the Nordic countries is a matter of primary concern [Phillips, 1989]. In rotary wheel exchangers frosting occurs when exhaust moisture is condensed in a cross-sectional zone in which the mean temperature of the rotor during one revolution is lower than 0 °C [Holmberg, 1989]. According to [Incropera, 1996], due to the link between frosting conditions and effectiveness, strategies of control based on the freezing point or fixed time period are inadequate. In [Holmberg, 1989] practical limits when excess water is drained out of the rotor were lab-tested, and concluded that: “the supply air limiting temperature can be assumed to be approximately -10 °C”. According to the experience in Norway, frosting is not an usual problem in rotary heat exchangers since frosting process is very slow and indoor humidity conditions usually change before this becomes a problem[Bilodeau, 1999].

Figure 1 shows that frost will start growing only during a few hours. However, this will not cause problems due to few successive hours with growth.

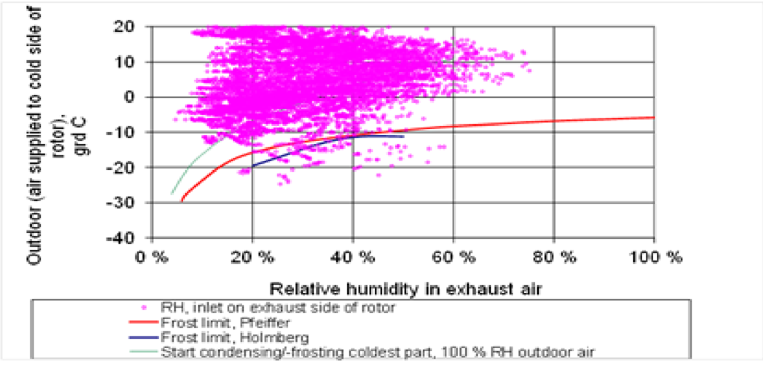

Figure 1 Frost limits threshold from [Pfeiffer, 1987]; red line, and [Holmberg, 1989] blue line. Purple dots: relative humidity (RH) for a heat wheel given a constant 21° C indoor temperature and normal year outdoor conditions in Oslo. Represented humidity production from a family with school children and non-home job parents.

Flat plate heat exchangers do not profit of the positive "unfreezing" effect of the rotation. Therefore they have a low annual energy recovery due to the need of energy for defrosting in cold climates [Drivsholm, 2005]. Condensation and frost problems in flat plate heat exchangers may be avoided by dehumidifying air before saturation temperature is reached. Aarnes [Aarnes, 2012] calculated that the necessary moisture efficiency in order to avoid freezing in Oslo climate is 70 %.

Experimental setup

A test rig was built to compare the frost formation of the laboratory at the Department of Energy and Process Engineering at NTNU. The frost growth in a recovery system based on two different plastic sheets was compared to the one with membrane sheet. For this experiment, the test conditions should be as stable as possible over the testing periods. The test should be preferably run at quite large humidity and temperature differences between the supply air and the exhaust air.

Supply Air Side

The supply air was taken from the laboratory and blown through cooling coils by means of a 12V axial computer fan. Two cooling coils in series with glycol as refrigerant were used to cool down the air on the supply air side. The typical air temperature when reaching the heat exchanger was about -4 to -10 °C depending on the air flow rates. The ducts worked as air straighteners before and after the heat exchanger, ensuring supply of a fully developed flow. Static pressure over the heat exchanger was measured to determine ice formation inside the cooling coil.

Exhaust Air Side

Conditioning air to simulate the exhaust air side was done inside a “climate chamber” built of 40mm thick polystyrene plates. By taking some of the cold air from the cooling coil, the inlet air to the “climate-chamber” was stable regarding temperature and humidity. A bucket of water with a 300W heating element and a thermocouple connected to a PID regulator was used to humidify the air. By regulating the water temperature in the water bucket both temperature and humidity were controlled. The inlet temperature and humidity were nearly constant. Figure 2 shows the configuration of the exhaust air side.

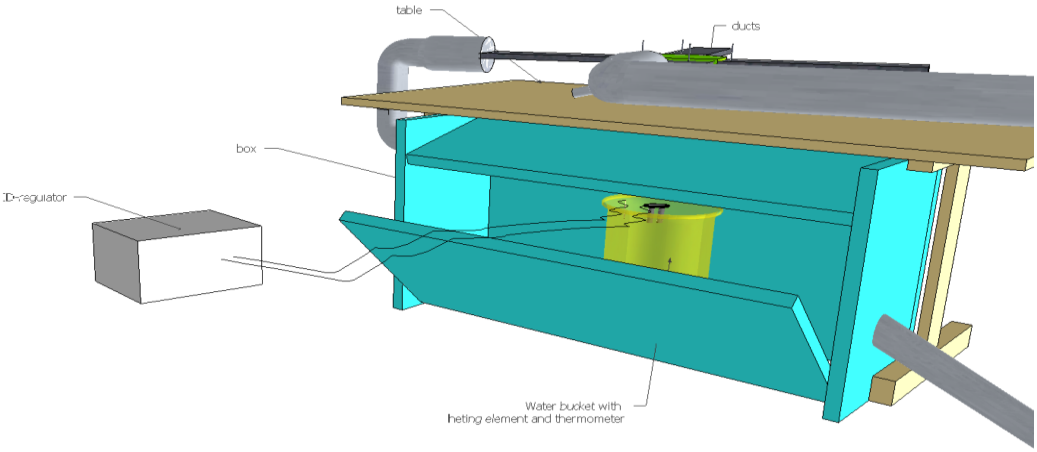

Figure 2 **Sketch of the exhaust air side of the test set-up**

The heater was dimmed to give 34 °C water temperature. This gave an exhaust temperature of approximately 22 °C and humidity of approximately 42 % RH. The PID-regulator had a set point at 40 °C to protect the bucket from drying-out. A 12V axial fan was attached at the inlet of the exhaust air supply duct.

Heat Exchanger Prototype

The prototype heat exchanger was built with a frame made of two hexagonal shaped 6 mm transparent acrylic plastic plates. Two 3 mm bars were glued to each hexagonal plate using epoxy glue. Four more bars of 6 mm Lexan were cut out as well. The first membrane layer was connected to the frame base layer using double sided tape. The construction was built like a sandwich with Lexan bars and membranes as seen in the Figure 3. For the first test this prototype was used. However, for the rest of the experiments eight new bars, now of 3 mm Lexan were made to increase the number of membrane layers from 3 to 5. Figure 3 shows the membrane heat exchanger appearance.

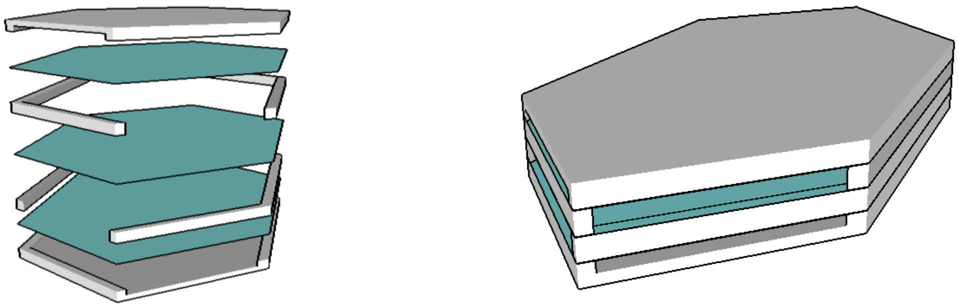

Figure 3 Heat exchanger made of sandwich construction. Drawing of layer prototype.

Three different plate type materials were tested. Two non-permeable "plastics" and one membrane with high permeability to water vapour. Properties of the materials are shown in Table 1

Table 1 . Properties for the different tested plate materials.

Material	Water permeable	Elastic	Crumples in high humidity
Wrapping plastic	No	No	No
PP (polypropylene)	No	yes	no
Membrane X	yes	yes	yes

The transparent wrapping plastic sheet was non-elastic and relatively stiff. This made it simple to construct the heat exchanger prototype. The PP-sheets and membrane were more difficult to handle in the construction of the prototype exchanger. Permeability values were taken from provider's datasheets. The PP-sheets had similar thickness to the membrane. The heat exchanger was placed as shown in Figure 4.

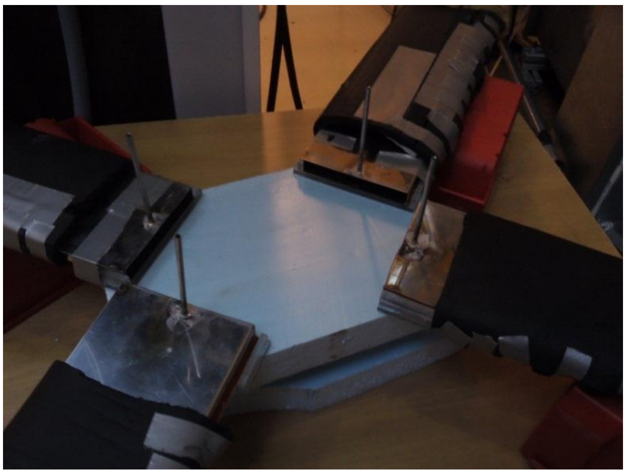

Figure 4 Supply and exhaust channels to connect to the heat exchanger

Measurement and Instruments

Vaisala temperature and humidity instruments were used. They were mounted perpendicular to the air streams in each of the rectangular ducts receiving and supplying air from and to the heat exchanger. The pressure drop over the heat exchanger was measured every second with micro manometers from DPM. The relative humidity and temperature of the surroundings and the supply air outlet velocity were logged with the TSI Velocicalc 9555-P instrument every second as well. The surrounding humidity and temperature were logged by the Velocicalc every 10 minutes (averaging by the instrument) [Aarnes, 2012]. The placement of the sensors is shown in Figure 5.

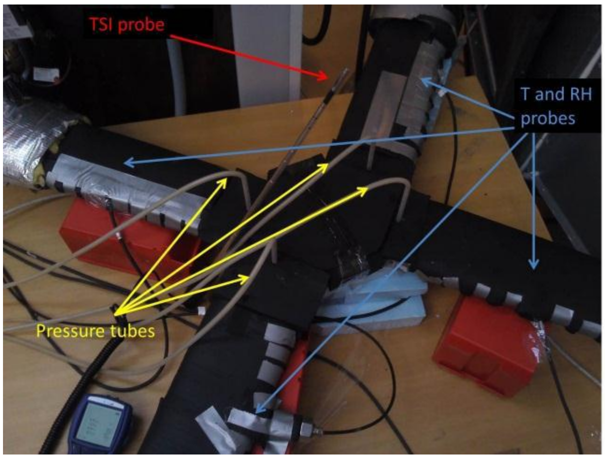

Figure 5 Positioning of sensors

Results

Flow Pattern inside the Heat Exchanger

A heat exchanger with colored copy paper as heat transfer plates was built to study the flow pattern inside the heat exchanger by means of a “smoke pen”. A fan was used to move the air through the heat exchanger and the smoke pen was ignited and moved along the inlet end of the heat exchanger. The colored paper made the smoke visible and it was possible to identify stream lines inside the heat exchanger. The picture series showed in Figure 6 were taken starting on the left with dry paper; the center after the heat exchanger was exposed to moist from the humidifier for approx. 5 minutes and to the right after approximately 10 more minutes exposed to moist. The paper started to crumple while exposed to humidity and for the last picture this effect was clearly visible.

Figure 6 Flow patterns at different humidity rates

Experimental Investigation

Table 2 shows the mean results for eight different experiments with different plate material.

Table 2 Overview of all experiments with mean values for inlet temperatures, relative humidity, pressure drops, measured air flow rates and calculated efficiencies based on the exhaust air side.

	Plate Material	$T_{s,in}$(c)	$T_{E,in}$(c)	$\Phi_{s,in}$(%)	$\Phi_{E,in}$(%)	ΔPs [Patel]	ΔPe [Patel]	Vs(m^3/h)	Ve(m^3/h)	η_T	η_T
1	Wrap plastic	-5.27	23.81	27.4	43.6	2.4	2.55	1.58*	1.58	0.27	no moisture transfer
2	Wrap plastic	-8.05	20.85	33.6	39.3	3.95	5.79	1.66	1.05	0.37	no moisture transfer
3	Mem	-4.96	21.25	27.1	42.86	9.248	8.96	0.74	1.38	0.41	0.37
4	PP	-8.41	21.04	35.17	46.15	6.16	6.73	1.38	1.3	0.35	No moisture transfer
5	Mem	-0.23	22.91	39.02	45.25	10.47	9.85	1.53	1.33	0.54	0.49
6	Mem	-4.32	22.77	29.54	43.27	11.13	10.48	1.55	1.2	0.54	0.58
7	Mem	-10.5	23.21	41.04	37.27	27.19	25.6	2.6	0.6	0.6	0.91
8	Mem	-9.62	22.9	34.22	46.6	25.25	24.8	1.4*	0.6	0.61	0.88

*) Flow rates not measured, but assumed from velocity and humidity-temperature diagram lines.

The temperature efficiencies for the different experiments are shown in Figure 7. The temperature efficiencies are quite stable over time for all experiments except experiment two. The development of the pressure drops are shown in Figure 8 for experiment one to six.

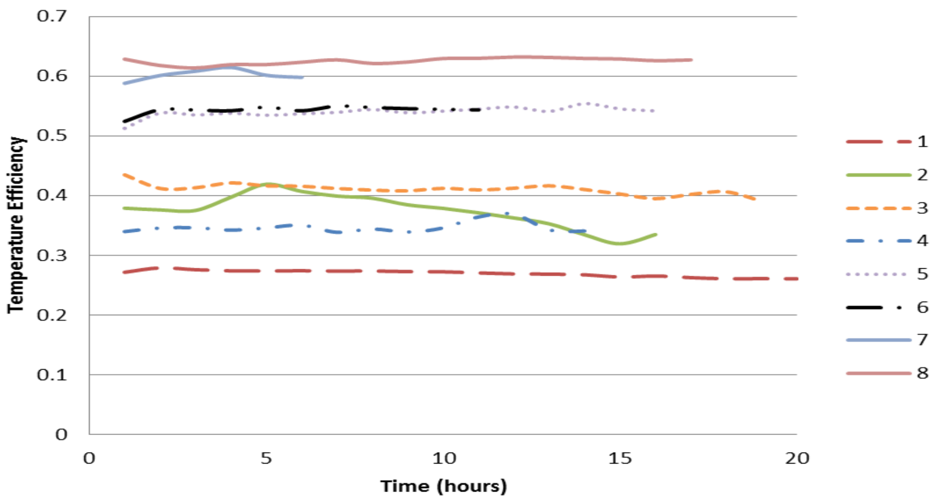

Figure 7 Temperature efficiency for all experiments.

The correlation between the temperature and moisture transfer efficiency in experiment 5,6,7 and 8 shows an increasing efficiency trend for lower temperatures. Experiment 3 does not follow this trend. However this was the only test where the exhaust air flow rate was greater than the supply air flow rate.

The development of the pressure drop are shown in Figure 8 for the experiments 1 to 6.

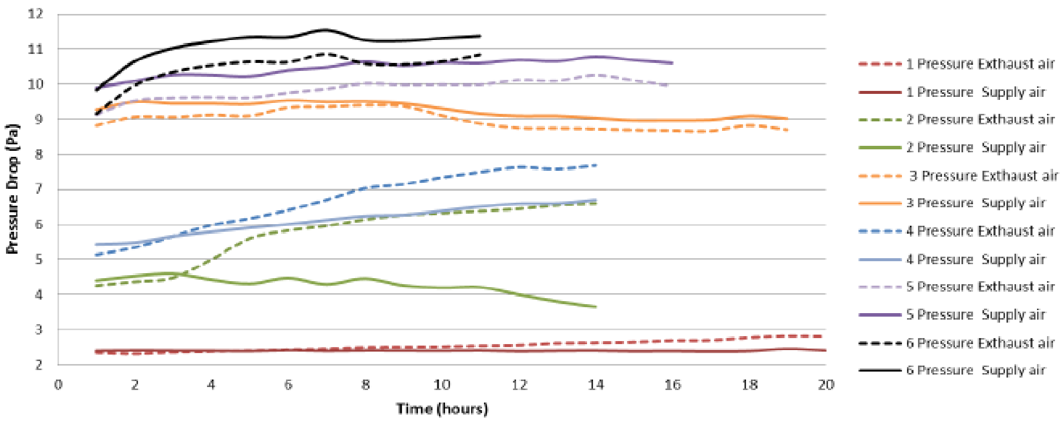

Figure 8 Pressure drop Experiments 1, 2 and 4 are plastic based while 3, 5, 6, 7, 8 are membrane based.

Ice and condensed water were found in all plastic exchangers (experiment 1, 2 and 4) plus in the last membrane based Experiment 8. The ice was formed in different areas in the plastic based and membrane based exchanger. For the plastic based heat exchangers, ice formation occurred in the exhaust air channels near the supply air inlet. For the membrane based heat exchanger, it was near the supply air outlet.

Membrane Crumpling

The membranes tend to expand and crumple in very humid conditions. This happened in Experiment 8 and was also observed when the cooling coil was turned off between the experiments. However, the membranes tightened when the humidity level went down again.

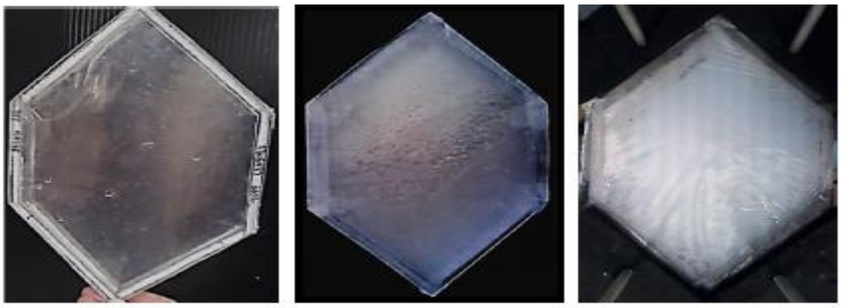

Figure 9 Expansion of the membrane in high humidity

The exchanger was dipped in water for one second. The membranes expanded and stuck together, as showed in Figure 9. After several hours, the membranes dried out and recovered their shape. Figure 10 shows the difference between the dry and the wet heat exchanger:

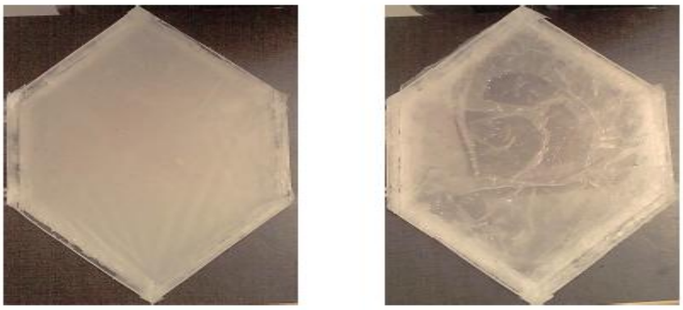

Figure 10 Left: Dry heat exchanger. Right: Wet heat exchanger

Discussion

The objective of this study was to investigate the difference between the plate materials regarding condensation and freezing. The change in pressure drop over the heat exchanger was used as an indicator of frost. The heat exchanger prototype top and bottom were made of transparent acrylic plates. This made it possible to do a visual investigation of possible formation of ice and its location. The streamlines in the dry exchanger were assumed to be of the same form in both exhaust and supply air streams. Figure 11 shows the streamlines for both streams overlaid.

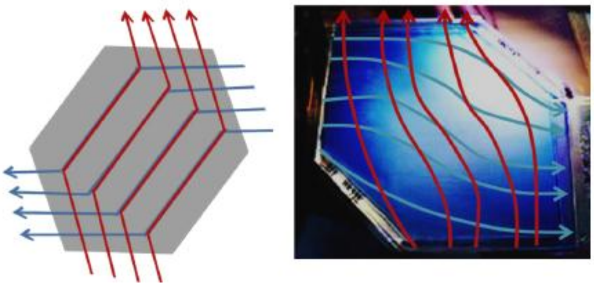

Figure 11 Left: Ideal quasi-counter flow heat exchanger flow. Right: Derived streamlines from smoke-pen test

In the ideal quasi-counter flow headers, the flows are perpendicular. However, in reality they are tilted about 30°. This means that numerical calculations based on counter flow assumption will probably give overpredicted values.

Pressure Drop and Flow Rates through the Exchanger

Program HXcalc (Aarnes,2011), calculates the pressure drop with the correlation:

$$\Delta P = \frac{1}{2}\rho_{air}V^2\left(\frac{4fL}{D_h} + 0.8775\right)$$

An increase in the flow rate will cause an increase in the pressure drop. Increasing flow rates will also cause a decrease in the U-value for the heat exchanger. This means that a rise in pressure drop should mean a reduction in temperature efficiency. However, the experimental results show the opposite result. No evident correlation was found between the measured pressure drop over the exchanger and the flow rate, as shown in Figure 12:

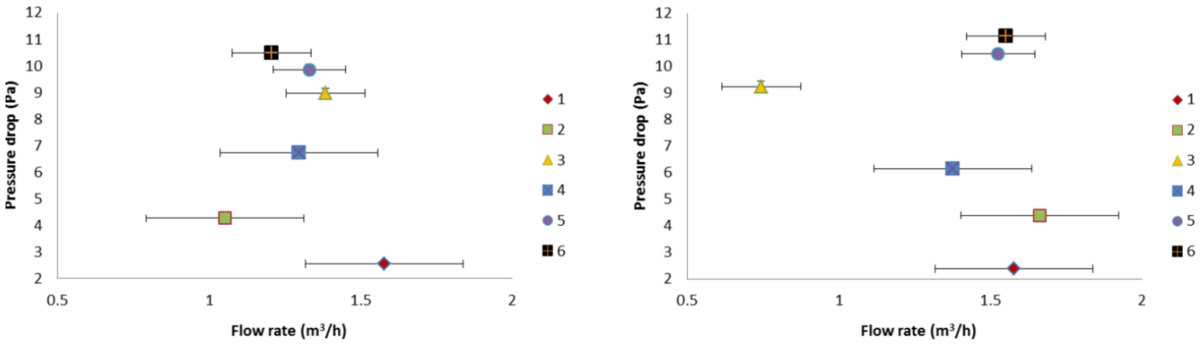

Figure 12 Correlation between the exhaust flow rate and the exhaust side pressure drop (left). Correlation between the supply flow rate and the supply side pressure drop (right). Error bars represent the total calculated error. When only one flow rate was measured the error was set to 0.26 m3/h which was the maximum calculated flow rate error for all tests.

The result may indicate other reasons causing the pressure drop. The pressure drop over the supply air side was not stable in all experiments. In Experiments 2, 3, 7 and 8 the supply air side pressure drop decreased, while in Experiments 4, 5 and 6, it increased. In both Experiments 2 and 4, a great amount of ice formation was observed. The test conditions were quite similar in these two experiments. However, the pressure drop over the supply air side had opposite developments in both tests, as shown in Figure 8. A possible reason for the difference may be the material properties. Experiment 2 utilized wrapping plastic, while Experiment 4 tested PP sheets. While an ice layer was building up in the PP-exchanger, blocking the exhaust air channels and increasing the exhaust air side pressure drop, the channel height of the supply air side channels changed and the pressure drop increased as well. This did not happen in Experiment 2 using the stiff plastic sheets.

When mass unbalance occurred between both sides of the membrane, the membrane curved towards the lower pressure, making the channels with the lower air flow rate narrower. The pressure drop at this side increased, as shown in Figure 8 for Experiments 1, 2, 3 and 4. For Experiments 5 and 6, this effect cannot be seen, suggesting a possible leakage between layers of the exchanger.

Evaluation of the Test Rig

For simplicity when building the experimental set up, an previously installed glycol cooling loop was used to produce cold air. Since the air flow rates were kept at low mass flow to increase the temperature efficiency, the temperature loss to the surroundings through the distance from the cooling coil to the heat exchanger was significant. This restricted the cold side temperature to about -10.5 °C for the coldest experiment. Challenges to get the same inlet conditions for the experiments, especially regarding pressure drop, air flow rates and supply inlet temperatures, made it problematic to compare the results. However, the exhaust air side of the test rig delivered very stable humidity and temperature conditions through the test periods and was found to be almost not affected by changes in the surroundings.

The hypothesis that ice formation would create an increase in the pressure drop over the exhaust air side of the heat exchanger was verified by the results. This means that the test rig was suitable to investigate whether ice formation problems occurred or not.

The uncertainty of the experimental results was huge, especially concerning the flow rate measurements and the temperature loss to the surroundings. The cooling coil tended to freeze and the air flow rate on the supply air side decreased with time. The test duration was restricted to the time before ice blocking of the cooling coil. For most experiments this happened after about 800 minutes. Yet, the accuracy was sufficient to see ice formation in the plastic based heat exchangers. Condensation of water was observed to occur after short time in these experiments (1,2 and 4). The differences in pressure drops also started to rise at the beginning of these experiments as seen in Figure 8.

Evaluation of the Membrane Based Heat Exchanger Prototype

The plastic based heat exchanger prototypes were tested to acknowledge water condensation and frost formation in the exhaust air channels. Water droplets were already observed through the transparent acrylic top plate in the top exhaust channel after few hours in these experiments. The frost formation appeared near the supply air inlet (see Figure 10), which correlates very well with the

CFD analysis of [Zhang, 2010]. Zhang's analysis shows that the coldest area in the exhaust air channels will be near the supply air inlet. The pressure drop difference in these experiments rose significantly through the test periods.

In the membrane based Experiments 3, 5, 6 and 7, neither condensation nor ice was formed. The change of pressure drop was not significant (Experiment 7 must be disregarded here, since the test duration was too short).

In the last experiment, a small amount of condensate water was observed near the supply air outlet in the upper exhaust air channel after 8 hours. The tested membrane material expanded in very humid conditions, as shown in Figure 10. At the end of the experiment, ice was found.

The frost appeared in different areas in the plastic based prototypes compared to the last membrane based experiment. In the plastic prototypes, the ice was formed in the coldest area in the exchanger; near the supply air inlet. The ice in the membrane prototype appeared near the supply air outlet, which was the second warmest side of the exchanger. A hypothesis of why ice was formed in this part was made in the Experiment 8. The crumpling of the membrane due to high humidity levels in the exhaust air side made the membrane stuck to the upper and lower heat exchanger frame plates, making the exhaust air stagnant. The supply air side cooled the stagnant exhaust air almost down to the temperature of the supply air. Since the moisture transfer efficiency cannot be 100%, condensation and freezing occurred.

As neither condensate water nor ice was found in the coldest spot (near the supply air inlet), this may indicate that the expansion of the membrane was the problem that caused the ice formation. Expansion and crumpling were observed when the humidity was high, for example in Experiment 8 with an exhaust humidity of 46.6 % and supply air temperature of -9.6 °C, but not in Experiment 7 with 37.3 % and -10.5 °C. Experiment 8 had the highest exhaust inlet absolute humidity at 8 g/kg (gram water per kg air). The supply air temperature in Experiment 7 was colder than in Experiment 8, though the exhaust humidity was lower (6.5 g/kg). It appears that the temperature and humidity conditions that lead to crumpling lay in the range between these two experiments. Since the humidity inside a residential building in winter seldom gets above 40 % RH, the tested membrane based heat exchanger may work in even lower outdoor temperatures than -10 °C.

Experiments show that for the colder supply air inlet, the mean relative humidity close to the membrane was higher. This decreased the mean moisture transfer resistance and increased the moisture transfer efficiency. Since the exhaust air flow rate was much higher than the supply air flow in the two last experiments, the measured efficiencies at the exhaust air side reached 90 %. This is evidently a much higher value than it would have been if the air flow rates were equal. Nevertheless, the correlation shows that an even lower supply inlet temperature may lead to higher moisture transfer efficiency.

The heat exchanger area and the flow rates were too low to compare this prototype to a "real case". However, the experimental results show that the freezing and condensation problems were smaller in the membrane based exchanger. Further research is needed to find out the temperature and humidity levels for which the membrane exchanger will freeze due to the limitation of moisture transfer effectiveness. The tested membrane was very elastic, which made it difficult to get it stretched out when building the prototype. The pressure drop behavior caused by the elasticity may be problematic if the flow rates are unbalanced. The elastic membranes would probably create an even bigger unbalance and the energy needed for transport the air through the heat exchanger would increase.

The optimal membrane should therefore not expand when wetted and should preferably not be elastic, due to problems caused by unbalance and the difficulties in building the exchanger.

Conclusion

The experimental tests showed frost formation on the plastic prototypes in the exhaust air channels near the outdoor air inlet side of the exchanger. This was not the case for the test with the hydrophilic membrane, which showed no condensation or ice near the outdoor air inlet. However, in the experiment with the highest exhaust air inlet humidity (46.6 % RH), the membrane had expanded and was crumpled near the supply air outlet. Condensate water and ice were found in the exhaust air channels near the supply air outlet in this experiment. The hydrophilic membrane was therefore found to be superior to the two plastic materials regarding water condensation and frost formation in the heat exchanger prototype when the exhaust air relative humidity was below 37% and the temperature above -10.5 °C. The pressure drops over the heat exchanger were found to be strongly influenced by the membrane materials elasticity and were not proportional to the flow rate as expected. The elasticity and the membrane's tendency to expand at high humidity made the tested membrane difficult to use in a membrane based heat exchanger. Methods to decrease the elasticity or stiffen it must be developed. Lamination of the membrane to a supporting fabric might be a possibility. Other types of membranes should also be tested. The experimental investigation was restricted to a supply air temperature about -10 °C. The membrane based heat exchangers performance at even lower temperatures should be investigated to see if the membrane based heat exchanger could work in extreme winter conditions without extra defrosting system. These tests should be repeated in a full-scale prototype to avoid scaling effects.

Membranes should also be tested for durability and pollution transfer to decide wether the technology is suitable for use in residential buildings with several living units.

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References

EERE, Ed. (2009). 2008 Building Energy Data Book. The Buildings Technologies Program, Energy Efficiency and Renewable Energy.

ASHRAE (1997.). *Handbook-Fundamentals*. Atlanta, GA, USA, .

Steimle, F., J. Roben (1992). Ventilation requirements in modern buildings. *Proceedings of the 13th AIVC conference Nice, France*: pp. 414-422.

Aristov, Y. I., I. V, Mezentsev, V. A.,Mukhin (2008). A new approach to regenerating heat and moisture in ventilation systems. *Energy and Buildings* 40(3): 204-208.

Phillips, E. G., R.E. Chant, B.C. Bradley, and D.R. Fisher.   (1989). A model to compare freezing control strategies for residential air-to-air heat recovery ventilators. *ASHRAE Transactions, Vol. 93, Pt. 2*.

Holmberg, R. B. (1989). Prediction of Condensation and Frosting Limits in Rotary wheels for Heat Recovery in Buildings. *ASHRAE Transactions,* Volume 95: pages 64-69, .

Incropera, F. P., D.P., Dewitt (1996). *Introduction to heat transfer* New York

Bilodeau, S. B., P. Lacroix, M. ,Mercadier, Y. (1999). Frost formation in rotary heat and moisture exchangers. *International Journal of Heat and Mass Transfer* 42(14): 2605-2619.

Pfeiffer, S. H. (1987). Untersuchungen zum Einfrieren von regenerativ wärmeübertragern. *ki klima - kälte*.

Drivsholm, C. O., H, Larsen, C. Jensen, J. S., Nielsen T. R. , Kragh J. , Svendsen S. . Udvikling af energiøkonomisk ventilasjonløsning med varmegenvinding til boliger. BYG- DTU. (2005).

Aarnes, S. (2012). Membrane based heat exchanger. . EPT. Trondheim, NTNU. . Master thesis.

Patel, H. N., C.J.Simonson, R.W, Besant (2010). Contaminant transfer between the supply and exhaust air streams of run-around membrane energy exchanger. *Proceedings of the 7th International Conference on Indoor Air Quality, Ventilation and Energy Conservation in Buildings .IAQVEC 2010*, Syracuse, New York, USA, August 15-18.

Zhang, L.-Z. (2010). Heat and mass transfer in a quasi-counter flow membrane-based total heat exchanger. *International Journal of Heat and Mass Transfer* 53(23-24): 5478-5486.