



## **Statusrapport 2 - bæredygtigt arktisk byggeri i det 21. århundrede** **Energirigtige ventilationssystemer**

**Kragh, Jesper; Rose, Jørgen; Svendsen, Svend**

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Bæredygtigt arktisk byggeri i det 21. århundrede  
- Energirigtige ventilationssystemer

Statusrapport 2 til  
VILLUM KANN RASMUSSEN FONDEN

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Energirigtige ventilationssystemer  
Statusrapport 2 til  
**VILLUM KANN RASMUSSEN FONDEN**

Jesper Kragh  
Jørgen Rose  
Svend Svendsen



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## Forord

Dette er statusrapport 2 for projektet med titlen *Energirigtige ventilationssystemer* støttet af VILLUM KANN RASMUSSEN FONDEN.

Nedenfor er først givet et kort resumé af forskningsindholdet i den første periode og herefter følger en beskrivelse af hvad der har været arbejdet med siden sidste statusrapport.

## Resumé af forskningsindhold i 1. periode

Udviklingen af energirigtig ventilation til brug i kolde klimaer indebærer en række forskellige problemstillinger, og igennem projektets første del har fokus først og fremmest været på at dokumentere og analysere disse problemstillinger, samt at opstille forslag til nye typer løsninger.

Erfaringer og målinger fra en række danske forsøgshusprojekter blev analyseret med henblik på at vurdere mekaniske ventilationssystemer med høj varmegenvindingseffektivitet under vinterdrift. Som forventet viste målingerne tydeligt at varmevekslerens høje effektivitet medfører tilsningsproblemer i kolde perioder når den fugtige afkastsluft (indeluft) køles ned til under frysepunktet. Samtidigt viser målingerne at indblæsningstemperaturen i kolde perioder falder markant, hvilket må formodes at medføre trækgener for beboerne.

De varmegenvindingsenheder der findes på ventilationsmarkedet i dag, er ikke velegnede til drift i kolde klimaer, hvor udetemperaturen i længere perioder ligger under frysepunktet. Der er derfor behov for at udvikle og afprøve nye varmegenvindingsenheder tilpasset kolde klimaer.

## Projektets arbejdsområder

Siden sidste statusrapport har der været arbejdet med følgende emner:

- Opstilling af beregningsmodel for varmeveksler incl. effekten af kondens og rim
- Konstruktion og afprøvning af ny kasseveksler med afrimningsfunktion
- Laboratorietest af varmegenvindingsenhed til lavenergihuset i Sisimiut

I det følgende beskrives arbejdet med de enkelte delemner nærmere.

### ***Opstilling af beregningsmodel for varmeveksler***

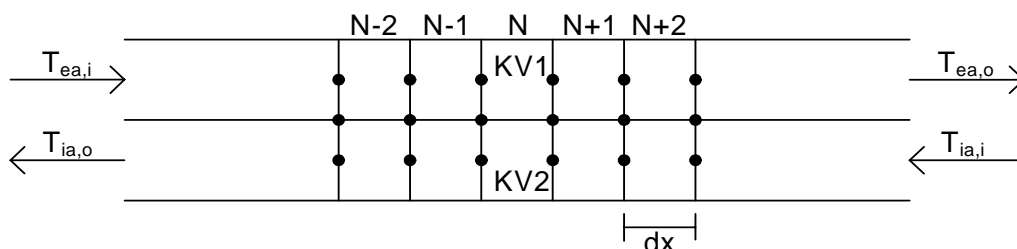
For at kunne foretage detaljerede teoretiske analyser af forskellige varmevekslerudformningers ydeevne under forskellige klimatiske påvirkninger, er der udviklet en matematisk model af en varmeveksler. Med denne model kan funktionen af nye vekslerudformninger beregnes og sammenlignes med test i laboratoriet.

Før arbejdet med modeludviklingen blev påbegyndt blev der gennemført et omfattende litteraturstudie af emnet *matematisk modellering af varmevekslere under hensyntagen til kondens- og isdannelse*, se bilag 1. Dette litteraturstudie viste at der er lavet en del arbejde indenfor området, men at langt størstedelen af litteraturen omhandler køleteknik og at der således kun er ganske lidt der direkte beskæftiger sig med luft-til-luft varmevekslere til bygningsventilation.

Med modellen kan man gennemføre analyser af forslag til veksler-udformninger, og med baggrund i de resultater der opnås, udvikle varmegenvindingskoncepter, således at man sikrer at tilisningsproblemer og trækgener undgås eller minimeres.

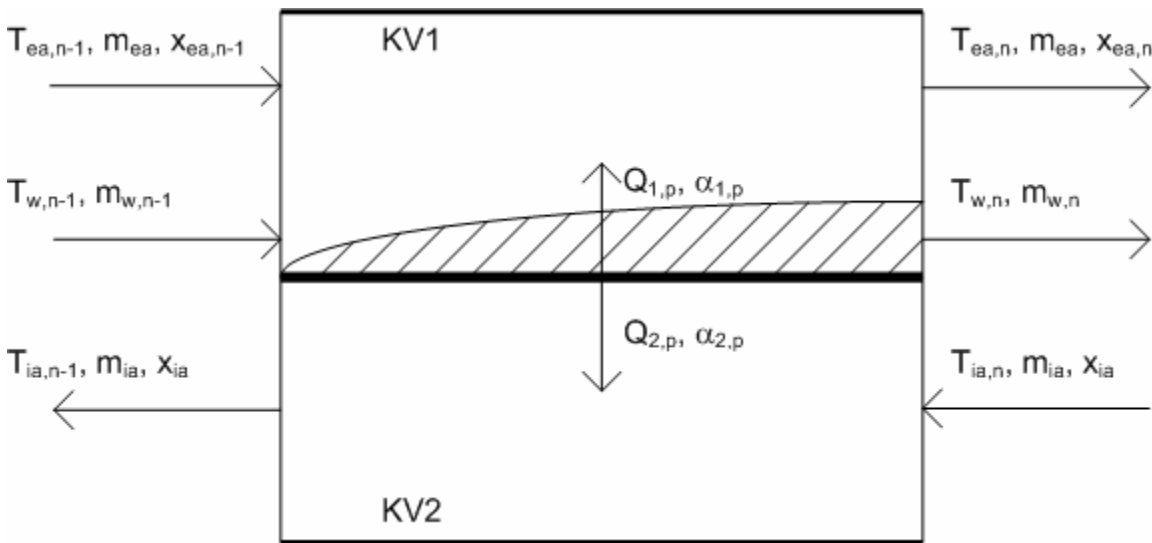
Formålet med dette arbejde er således at udvikle en nøjagtig og anvendelig model/metode til at beregne de komplekse varmeoverføringsmekanismer som forekommer i luft-til-luft pladevarmevekslere når der forekommer faseskift, dvs. tilisning eller kondens, i den ene af de to luftkanaler. Kondens og tilisning vil i høj grad påvirke varmeoverføringskoefficienterne, energibalancen, tryktabet og luftgennemstrømningen i varmeveksleren, og problemstillingen er derfor primært at fastlægge sammenhængen mellem disse forhold i en tilstrækkeligt nøjagtig form. Tryktab og luftgennemstrømning vil først spille en rolle når modellen videreudvikles til dynamiske forhold, hvor ophobningen af is eller kondens vil spille en rolle.

I første omgang er der udviklet en 1-dimensional stationær model, som kan benyttes til fastlæggelse af varmeoverføringskoefficienterne under forskellige forhold, dvs. svarende til tilfældene uden faseændring og når kondens eller tilisning forekommer. Modellen er opbygget som et Excel-regneark. I Figur 1 er vist hvorledes problemstillingen diskretiseres ved at varmeveksleren inddeles i et antal lige store delelementer (i figuren benævnt N-2, N-1, N, N+1 og N+2), med længde  $dx$ . Betragtes elementet N, indeholder dette 3 kontrolvolumener; den varme luftstrøm (KV1), den kolde luftstrøm (KV2) og pladematerialet som adskiller luftstrømmene. I hvert af kontrolvolumenerne haves en ind- og udløbstemperatur (angivet med sorte prikker i figuren). Når der forekommer vand (kondens) i den øverste kanal, antages det at dette vand forlader kanalen med samme temperatur som pladevæggen mellem de to luftstrømme.



Figur 1. Diskretisering af problemstilling.

For hvert delelement opstilles der masse- og energibalancer som vist i Figur 2.



Figur 2. Masse- og energibalancer for element N.

For udsugningen i Figur 2, øverst, er luften karakteriseret ved en indløbstemperatur  $T_{ea, n-1}$ , en massestrøm af tør luft  $m_{ea}$  og et fugtindhold  $x_{ea,n-1}$  og vandet (eventuel kondens fra tidligere delelement) er karakteriseret ved en indløbstemperatur  $T_{w,n-1}$  og en massestrøm  $m_{w,n-1}$ . Massestrømmen af tør luft  $m_{ea}$  regnes konstant, og derfor er de ubekendte for kontrolvolumen 1 udløbstemperaturen for luften,  $T_{ea,n}$ , fugtindholdet i luften som forlader kontrolvolumenet,  $x_{ea,n}$  samt temperatur og massestrøm for vandet,  $T_{w,n}$  og  $m_{w,n}$ .

Pladematerialet som adskiller de to luftstrømme modtager varme fra udsugningsluften dels via den konvektive varmeoverføring men også en eventuel kondensvarme og frysevarme. Pladevæggens temperatur  $T_p$  skal altså fastlægges ved en varmebalance mellem tilføring af varme fra udsugningen og afgivelse af varme til indblæsningen. Der tages herudover også hensyn til en eventuel varmeledning på langs i pladematerialet, mens der ikke tages hensyn til varmeledning på tværs af pladematerialet.

For indblæsningen, nederst, er luften karakteriseret ved en temperatur  $T_{ia, n}$ , en massestrøm af tør luft  $m_{ia}$  og et fugtindhold  $x_{ia}$ . Massestrømmen af indblæsningen samt fugtindholdet i luften antages at være konstant, og derfor er der for indblæsningsluften kun én ubekendt svarende til temperaturen af luften som forlader kontrolvolumen 2,  $T_{ia,n-1}$ .

Denne 1-dimensionale stationære model danner grundlaget for en videreudvikling af den matematiske formulering, således at der kan gennemføres beregninger under dynamiske (tidsvarierende) forhold, hvilket vil muliggøre analyser af ophobning af is i veksleren på baggrund af f.eks. design reference år (vejrdato), f.eks. design vejrdato for Uummannaq og Nuuk. Herved vil man kunne karakterisere en given vekslers ydeevne, forudsige i hvilke situationer der kan forekomme problemer med tilisning, og samtidig vil man kunne optimere vekslerudformninger således at problemer minimeres mens ydeevne maksimeres.

Arbejdet med modeludviklingen vil blive dokumenteret i en artikel som forventes publiceret i et internationalt tidsskrift under titlen "Counter flow air-to-air heat exchanger model with phase change".

Den 1-dimensionale stationære model skal naturligvis verificeres ved sammenligninger med målinger på konkrete varmevekslere under kontrollerede forhold. Der er gennemført en enkelt sammenligning af modellens resultater med resultater fra en forsøgsopstilling i BYG.DTU's forsøgshal. Den varmeveksler som er brugt i den pågældende forsøgsopstilling er en modstrømsvarmeveksler af mærket Recair Sensitive (se evt. følgende Internet-adresse <http://www.recair.nl/GB/recair.sensitive.htm>) med en længde på ca. 0,3 m, svarende til at den skulle have en temperatureffektivitet på ca. 90 % ved en luftstrøm på 50 m<sup>3</sup>/h.

I forsøget sendes afkastluft med temperatur 20,0 °C og relativ fugtighed på 33 % ind fra den ene side og indblæsningsluft med temperatur -2,5 °C ind fra den anden side, og der aflæses hvilken temperatur og relativ fugtighed afkastluften har når den forlader veksleren og hvilken temperatur indblæsningsluften har når den forlader veksleren.

I laboratorieforsøget er følgende resultater opnået:

$$T_{\text{afkast}} = 1,5 \text{ °C}$$

$$RF_{\text{afkast}} = 87 \%$$

$$T_{\text{indblæsning}} = 17,5 \text{ °C}$$

I varmevekslermodellen er vekslerens karakteristiske dimensioner og øvrige data indtastet. Herudover fastsættes de kendte temperaturer og fugtindhold for luftstrømmene, og der gennemføres en beregning af ovenstående tre parametre. Resultaterne er angivet nedenfor:

$$T_{\text{afkast}} = 2,4 \text{ °C}$$

$$RF_{\text{afkast}} = 96 \%$$

$$T_{\text{indblæsning}} = 17,7 \text{ °C}$$

Sammenligner man de to sæt resultater er det tydeligt at der er forskel på de i praksis opnåede værdier og de teoretisk bestemte værdier, men forskellene er relativt beskedne og modellen giver altså et rimeligt godt billede af forholdene i veksleren.

Det mest interessante aspekt i varmevekslermodellen er muligheden for at tage højde for den varme som opstår i forbindelse med at der kondenserer vand i den ene side af veksleren. For at kunne vurdere betydningen af at der tages højde for kondensvarmen, er der gennemført endnu en beregning hvor der ikke tages hensyn til kondensvarmen, og resultaterne er som følger:

$$T_{\text{afkast}} = 0,2 \text{ }^{\circ}\text{C}$$

$$RF_{\text{afkast}} = 0 \text{ \% (den relative luftfugtighed bliver 0 \%, da fugten fjernes i beregningerne)}$$

$$T_{\text{indblæsning}} = 17,4 \text{ }^{\circ}\text{C}$$

Sammenlignes disse resultater med resultaterne hvor kondensvarmen blev medtaget, er det tydeligt at det har en stor betydning, og specielt for afkasttemperaturen som falder fra 2,4 °C til 0,2 °C. Dette viser at kondensen har en stor betydning og at det dermed er nødvendigt at medtage denne i modellen.

Der er fortsat en række områder som der skal arbejdes videre med i forbindelse med modellen, og i det efterfølgende er kort opridset nogle af de videre analyser der vil blive gennemført.

1) Modellen er i som udgangspunkt opdelt i 10 del-områder, svarende til at der indenfor hver af de 10 områder foretages en 1-dimensional, stationær beregning af varmeudvekslingen, og en af de ting der bl.a. skal overvejes i forbindelse med verificeringen af modellen er, hvorvidt denne inddeling af veksleren er tilstrækkeligt nøjagtig.

2) I modellen antages det at en eventuel kondens vil forekomme på pladen som adskiller de to luftstrømme, og her vil det skulle vurderes hvorvidt en del af kondensen kan forekomme som tågedannelse i luften i stedet for. I modellen er der allerede indlagt mulighed for at ændre på dette forhold.

3) Når den 1-dimensionale stationære model er endeligt verificeret vil den endeligt fastlagte matematiske formulering og metode blive benyttet i forbindelse med opbygning af en dynamisk model, og denne model kan herefter benyttes i forbindelse med dynamisk simulering af varmevekslere. Kondens- og isdannelse vil påvirke tryktab og luftgennemstrømning i varmevekslere og dette forhold vil også skulle indgå i den dynamiske model, således at der kan foretages vurderinger af forskellige metoder for afisning af varmevekslere.

### ***Konstruktion og design af ny kasseveksler***

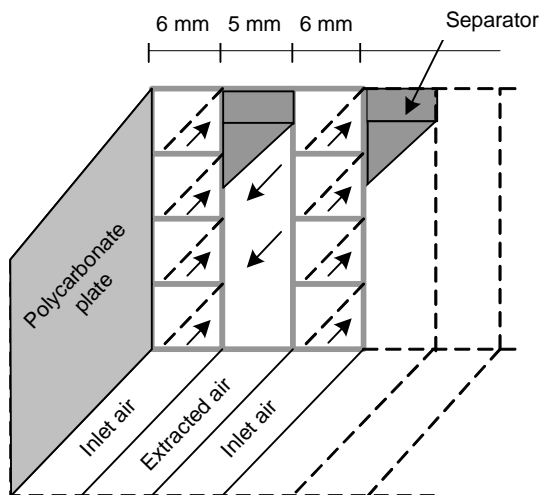
Der er brugt en del ressourcer på at udvikle og designe en ny type varmeveksler, der kontinuerligt kan afrime den is, der uundgåeligt vil dannes i veksleren. Varmeveksleren er opdelt i to vekslere, der skiftevis er aktive. Når den ene vekslerdel er inaktiv benyttes ca. 10 % af den varme afkastluft til afrimning af vekslerens overflader. Da energiindholdet i den fugtige afkastluft er større end i den tørre udeluft, er de resterende 90 % af afkastluften nok til at forvarme den kolde indblæsningsluft tilstrækkeligt. I designet af veksleren er desuden lagt stor vægt på at minimere tryktabet, idet ventilatorernes elforbrug også indgår i det samlede energiregnskab. Endvidere er der lagt vægt på et simpelt og driftsikkert design med mulighed for lokal produktion i mindre arktiske byer. En prototype af veksleren er blevet konstrueret på BYG-DTU og testes under kontrollerede laboratorieforhold. Målingerne skal desuden senere benyttes til at validere beregningsmodellen beskrevet ovenfor.

På Figur 3 ses et billede af veksleren. Den første prototype af veksleren er ikke et fuldska-  
la forsøg, idet veksleren er dimensioneret til kun at skulle kunne dække 1/5 af luft volu-  
menstrømmen fra et almindeligt ét familie hus. I praksis vil kasseveksleren således skulle  
være ca. 5 gange dybere.

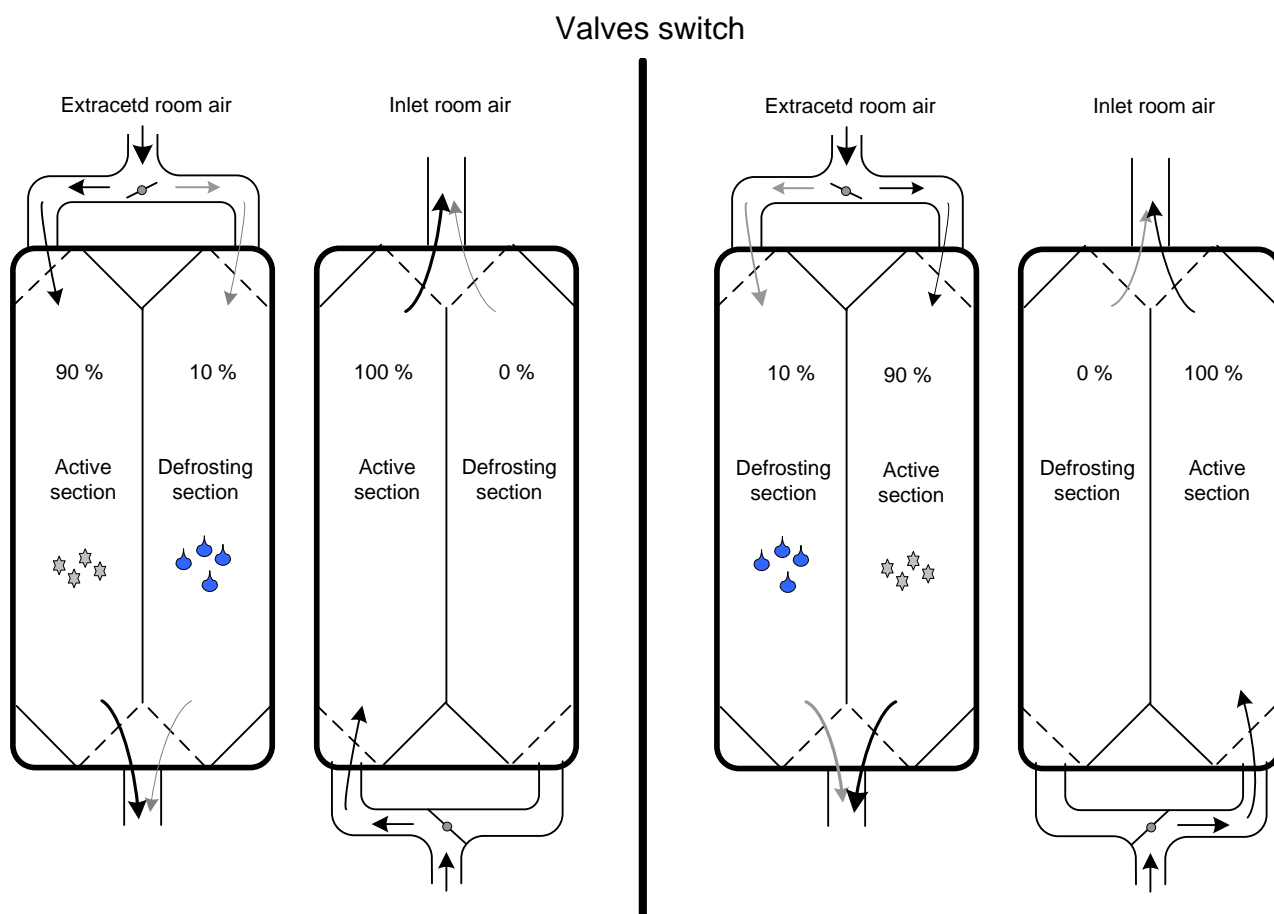


Figur 3 Billede af kasseveksleren.

Veksleren er opbygget af ribbeplader, hvor luften føres modstrøms for at opnå maksimal  
effektivitet. Principskitse af vekslerens opbygning ses på Figur 4 og Figur 5.



Figur 4 Opbygning af veksleren med angivelse af flowretning



*Figur 5 Skitse af kasseveksleren i to forskellige plan for hhv. afkast og indblæsningsluften. Ved spjældskift, angivet ved midterlinien på tegningen, ændres hvilken af de to vekslerhalvdele der er aktive. På afkastssiden fordeles luftstrømmene til de to sektioner med hhv. 90 % og 10 %. De 10 % af luftstrømmen benyttes således til afrimningen.*

Foreløbig test af kasseveksleren viser at den automatiske afrimning fungerer og at indblæsningstemperaturen ikke falder til kritiske niveauer.

### **Test af varmegenvindingsenhed til lavenergihuset i Sisimiut**

Til det nyopførte lavenergihus i Sisimiut er udviklet en ny varmegenvindingsenhed i samarbejde med ventilationsfirmaet EXHAUSTO A/S. Enheden består af to modstrømsvekslere koblet i serie. Når udetemperaturen er under frysepunktet vil der dannes rim i den første veksler. Efter et forudindstillet tidsinterval vendes flowretningen ved hjælp af to spjæld på både afkastssiden og indblæsningssiden, således at den tilrimede veksler optøs. I den anden veksler vil ny rim samtidigt begynde at afsættes.

Enheden blev testet under laboratorieforhold på DTU inden den blev installeret i Lavenergihuset. Testen viste at afrimningsfunktionen virkede efter hensigten, men at det dog var nødvendigt med en eftervarmefflade for i meget kolde perioder at kunne hæve indblæsningstemperaturen et par grader for at undgå risiko for trækgener for beboerne. På Figur 6 ses et billede af varmegenvindingsenheden.



*Figur 6 Billede af varmegenvindingsenheden til lavenergihuset i Sisimiut*

## Projektstatus og det videre arbejde

I projektet er forskellige tekniske løsninger af tilsningsproblemet afprøvet og analyseret under laboratorieforhold. Den foreløbige vurdering af de forskellige principløsninger er følgende:

- 1. princip med at bytte om på rækkefølgen af to vekslere giver kompliceret kanal- og spjældløsninger og seriekoblingen af vekslere giver problemer med tryk- tab/elforbrug og driftssikkerhed.
- 2. princip med at vende flowretningen igennem to serieforbundne vekslere giver problemer med lave temperaturer efter vendinger af flowretningen (Viser behov for at tage hensyn til varmekapacitet i beregningsmodellen af varmeveksleren).
- Det foreslåede 3 princip med delstrømme vil have en "forvarmet" veksler når der skiftes. Princippet virker kun hvis der er overskud i energiindholdet i indeluften pga. højere fugtindhold, men der er også kun tilsningsproblemer, hvis fugtforholdet i indeluften er højere end i udeluften.

Den tidligere beskrevne kasseveksler blev derfor designet og konstrueret på BYG-DTU's værksted. De første forsøg er så lovende, at der i projektets afsluttende fase vil der blive fokuseret på at optimere denne kasseveksler. Laboratiormålingerne vil blive sammenlignet med tilsvarende simuleringer foretaget med den udviklede simuleringsmodel af en varmeveksler. Når modellen er verificeret kan forskellige parametervariationer vise, hvor kasseveksleren kan optimeres yderligere.

## Regnskab

Se bilag 2

## Publikationer

*Mechanical ventilation with heat recovery in cold climates*, Kragh J., Rose J., Svendsen S., Department of Civil Engineering, Technical University of Denmark, marts 2005, Nordic Symposium on Building Physics, Reykjavik 13-15 June 2005

Se bilag 3

*Mechanical ventilation systems with heat recovery in arctic climate*, Kragh J., Rose J., Svendsen S., Department of Civil Engineering, Technical University of Denmark, April 2005, Symposium on Energy Efficient Building in Sisimiut, April 2005.

### Artikler og papers under udarbejdelse:

*Simulation of ventilation systems for single-family houses in cold climates*, Rose J., Kragh J., Svendsen S., Department of Civil Engineering, Technical University of Denmark.

*Counter flow air-to-air heat exchanger model with phase change*, Rose, J., Nielsen, T. R., Kragh, J. and Svendsen, S., Department of Civil Engineering, Technical University of Denmark.

*New designed counter flow heat exchanger for cold climate*. Kragh J., Rose J., Svendsen S., Department of Civil Engineering, Technical University of Denmark

*Measurements of heat exchangers in Danish testhouses*, Kragh J., Rose J., Svendsen S., Department of Civil Engineering, Technical University of Denmark (Artikel som forventes publiceret i Acta Physica Aedificiorum (<http://www.byv.kth.se/avd/byte/bphys/>))

## Præsentationer

*Mechanical ventilation systems with heat recovery in arctic climate*, Kragh J. Department of Civil Engineering, Technical University of Denmark, April 2005, Symposium on Energy Efficient Building in Sisimiut, April 2005.

Se bilag 4

## **Bilag 1: Literature study – Heat exchangers.**

This document represents a summation of the information gathered during a literature study on the subject of “heat exchangers”. In the literature study, focus has been on counter flow air-to-air plate heat exchangers, and especially the mathematical formulation and modeling of the heat transfer mechanisms that occur when condensation or frost (ice) formation occurs, i.e. the changes in heat transfer mechanisms that occur when water or ice is present on one side of the heat exchanger.

### **Background**

Using mechanical ventilation with highly efficient heat recovery in northern European or arctic climates, is a very efficient way of reducing the energy use for heating in buildings, however it also presents a series of problems concerning condensation and frost formation in the heat exchanger. When moist air comes in contact with a cold surface that has a temperature that is below the dew-point temperature of the water vapor in the air, condensation will occur. If the cold surface has a temperature that is below the freezing point, frost formation will occur. The deposition of frost will typically reduce the heat exchanger efficiency, i.e. the heat transfer rate is reduced, and the exhaust air side of the heat exchanger will experience pressure drops, as the frost growth blocks the air flow passage. Unless defrosting mechanisms are initialized at this point, the heat exchanger will eventually freeze up.

There are different ways of avoiding/removing frost formation in heat exchangers, but typically these will have a negative effect on the heat exchanger efficiency or imply the use of extra energy. Therefore there is a need to further analyze the possibilities of more energy efficient methods of avoiding/removing frost formation in heat exchangers. In order to perform this type of analysis it is necessary to perform both experimental and theoretical studies on the subject, and the theoretical approach is the logical first step.

### **Purpose**

The purpose of this literature study is to establish the knowledge for developing an accurate and useful model/method for calculating the complex heat transfer mechanisms that occur in a counter flow air-to-air plate heat exchanger when phase changes occur in one of the air ducts, i.e. condensation or frost formation. Condensation and frost formation will influence the heat transfer coefficients in the heat exchanger, and it is basically a question of determining these relationships in an adequately accurate form that we seek to do.

The first objective is to develop a 1-dimensional stationary model that is valid for determining the heat transfer coefficients under different circumstances, i.e. when condensation or frost formation occurs on one side of the heat exchanger. This model could, due to its relatively simple nature, be developed in a spreadsheet. The second objective is to expand the formulation to take into account the transient development of the problem, i.e. in order to analyze how and when condensation or frost formation occurs and what effect it has on air flows, pressures and especially heat transfer coefficients. This second and relatively more complex model

could be developed in a mathematical environment as Matlab<sup>®</sup>, where non-linear integrals can be solved by built-in routines.

### **Literature – Reviews of past research in the field**

The study of frost formation and frost growth in heat exchangers has gone on for more than 50 years, and a huge effort has been put into better understanding and especially modeling of this phenomenon. The primary focus through these 50 years of research in the field have not been on air-to-air heat exchangers for building ventilation, but on air-to-refrigerant heat exchangers used in the refrigerating industry, however the basic problems concerning the heat transfer mechanisms are the same.

The primary objective of the research performed in this field is to develop correlations for describing the frost in a way that makes it possible to accurately predict how, and under which circumstances it will occur, so that it is possible to use these correlations for heat exchanger design and the development of energy efficient defrosting methods. The most important properties of frost growth, affecting the heat exchanger performance, are the thickness of the frost layer, the thermal conductivity of the frost and the frost density. However, these properties are all functions of the type of surface, temperature of the cold surface, temperature of the frost, temperature of the air, air velocity and air humidity and therefore the generalization of frost properties is extremely difficult and most of the correlations that have been developed over the years have either been established empirically or theoretically by neglecting terms of lower significance, e.g. by assuming that the surrounding air was saturated ideal gas at room temperature.

In 1985 O'Neal and Tree (1985) published a comprehensive review of frost research in simple geometries (flat plate, cylinders, tubes, parallel plates etc.) with special focus on the available correlations for the determination of frost thickness, frost thermal conductivity and heat transfer coefficient on frosting surfaces. This work would sum up approximately three decades of research in the field of frost formation and frost growth. Padki, Sherif and Nelson (1989) followed up on this, including the new research that had dawned since O'Neal and Tree did their review. However, during the last 15-20 years the advances in computer modeling and computational methods have provided a basis for much more advanced analytical and numerical studies in the field. In 2004 Tao, Jia and Iragorri (2004) published a review and comparative analysis of the different methods and approaches put forth during the last 20 years of research in the field. These comparisons covered all the correlations described by O'Neal and Tree, but also added a review on the different frost growth models that had been developed during the period, including their respective limitations and ranges of operation.

Basically, the research in this field can be divided into four groups, depending on which correlations or models the researcher is trying to establish:

- 1) Correlations for determining frost thickness
- 2) Correlations for determining frost thermal conductivity
- 3) Correlations for determining the heat transfer coefficient on frosting surfaces, and
- 4) Models for frost growth

Often the first 3 groups are intertwined in some way or other, or the researcher uses correlations developed by other researchers for one or more of the correlations in order to establish correlations for the others.

We are trying to establish a method or model for determining the heat transfer mechanisms that occur in an air-to-air plate heat exchanger under condensation or frosting circumstances, and therefore it is necessary to take a look at the methods that have been used by others in the past. In the following section, a brief summation of some of these methods is detailed.

### **Basic calculation principles**

Fundamentally, the mathematical description of heat transfer mechanisms that occur in a heat exchanger, or energy systems in general, can be described by the three general laws of conservation; Conservation of energy (1<sup>st</sup> law of thermodynamics), conservation of mass (continuity), conservation of momentum (the pressure-drop equation). In a system where no phase change occurs, the equations that can be derived from these three laws can be solved analytically, however, when condensation or frost formation (phase changes in general) occur, the solution can no longer be found analytically and has to be found by other means, e.g. numerically with simplification of the system or by using some simplifying correlations for describing the very complex nature of the heat exchanging that occurs.

Formation of frost on subfreezing surfaces is quite complicated, especially because the rate at which heat is transferred from the moist air to the frost layer influences the rate at which the water vapor is diffused into the layer of frost. The temporal dependency of the frost properties and the temporal and spatial dependency of the frost-air interface temperature also complicate the matter. Many investigations have shown that at the initial stages of the deposition process, the heat transfer coefficient will experience an increase, and this effect has been attributed to the fact that the rough frost surface at the initial stage will act as a finned surface, hereby being able to transfer more heat.

The heat transfer mechanisms in a heat exchanger can be described mathematically in a number of different simplified ways; e.g. based on different assumptions concerning the overall heat-transfer coefficient  $U$ , the state of the system, i.e. by assuming an adiabatic system, the uniformity of the temperature distribution over a given cross section and the properties of the heat-exchanging fluids, i.e. assuming that the specific heats of the fluids are constant. There are two types of methods in particular, that has been used extensively in the past to perform theoretical studies on heat exchangers. These include the Log-Mean Temperature Difference approach, or LMTD-approach and the Effectiveness Number of Transfer Units approach, or  $\epsilon$ -NTU-approach.

### **LMTD – Log Mean Temperature Difference**

The LMTD-method has been used in several different studies of heat exchangers. The method is restricted by the following assumptions; 1) Constant flowrates, i.e. the method does not allow for pressure drops/rises due to changes in duct geometry, 2) Constant heat capacities and constant heat transfer coefficient between the medias, i.e. the method does not allow fluid heat capacities to change and no change in phase, 3) Constant heat transfer area in each pass, 4) Shell fluid temperature is uniform over the cross section, 5) No fluid or heat leaks between

shell passes, and finally 6) Heat losses are negligible, i.e. the system is adiabatic. One of the main disadvantages of the LMTD-approach is, that all temperatures at the heat exchanger inlet and outlet need to be known, and therefore models will typically need to solve the set of equations iteratively, until a solution that satisfies the whole system is found. Typically, the method is utilized for investigations that involve experimental validation of some sort.

Sherif, Sengupta and Wong (1998) performed an experimental investigation of frost deposition on a cylinder in a cross-flow heat exchanger in order to obtain empirical correlations for the frost thickness and heat transfer coefficient as functions of time. They used the LMTD-method for the mathematical description of the heat transfer coefficient. The correlations that they derived for frost-thickness and overall heat transfer coefficient were found to represent experimental data well, especially for the heat transfer coefficient. The correlation for frost thickness was most accurate towards the end of the 2-hour experiments, where the first 20 minutes of the experiments resulted in deviations of up to 25%.

Deng, Xu and Xu (2003) evaluated heat transfer performance of an experimental industrial size air cooler under frosting conditions. The overall heat transfer coefficients were based on the LMTD-approach and the energy transfer coefficients based on a Logarithmic Mean Enthalpy Difference, LMED, i.e. basing the heat transfer coefficients on mean temperature and the energy transfer coefficients on mean enthalpy. Their experiments show, what others have shown before, that the overall heat transfer coefficient initially increases when frost formation occurs but rapidly starts to decrease afterwards. Furthermore, they draw conclusions as to geometry, size and spacing of fins in order for optimum performance under frost conditions.

### **$\epsilon$ -NTU – Effectiveness – Number of Transfer Units**

The  $\epsilon$ -NTU-approach (effectiveness number of transfer units) is also a method that has been used quite extensively for solving heat exchanger problems. The method is typically used where only the inlet temperatures of the hot and cold fluids are known, i.e. the outlet temperatures of the hot and cold fluids are unknown and therefore the LMTD-approach cannot be used directly.

The main problem with this method, in respect to the investigations that we are trying to undertake, is that the method has difficulty with handling situations where the heat transfer coefficient changes significantly over the heat exchange surface. The heat transfer coefficient will be dependent on phenomena as condensation and frost formation, and therefore this particular method is not very applicable for investigations including frost formation. However, the investigations that have previously been performed using this approach can still be interesting with respect to the methods that are applied for describing the heat transfer mechanisms that occur in the heat exchanger. In the following some of the investigations using the  $\epsilon$ -NTU approach are briefly summarized.

Söylemez (2000) developed a method for thermo-economic optimization of the heat exchanger area for energy recovery applications. The method was based on simple algebraic formulas and using the  $\epsilon$ -NTU approach, and it covers both parallel flow, counter flow, single fluid and phase change heat exchange. The validity of the method is tested on a sample problem taken from

the literature, Stoecker (1989), and it is concluded that the method is helpful, especially for industrial applications.

Wetter (1999) developed a static simulation model for air-to-air heat exchangers (counter flow, parallel flow and cross flow), taking into account the dependence of the convective heat transfer coefficient on the air mass flow and temperature. The model prescribes that no condensation occurs (i.e. condensation and frost formation is not covered in the model). The primary purpose of the model is to be able to calculate the energy consumption of a heat exchanger at an early stage, i.e. for design purposes primarily. The heat exchanger effectiveness calculations are based on  $\varepsilon$ -NTU calculations.

Gvozdenac and Sad (1990) developed an analytical method for calculating the transient response of a parallel flow heat exchanger with finite wall capacitance. The model is developed on the base of three local energy balance equations, which are solved by the Laplace transform method for step change of the primary fluid inlet temperature. The model was verified by comparing results for equal fluid velocities and infinite fluid velocities and proven to be correct. The solution was based on the NTU approach, i.e. defining the number of transfer units as a function of the heat transfer coefficients and the thermal capacity.

Gvozdenac and Sad (1993) developed an analytical solution for the transient response of a counter flow heat exchanger with finite wall capacitance. As above, they applying the energy conservation equation to both fluids and the wall, obtaining three simultaneous partial differential equations that can be solved by the Laplace transform. Again the solution is based on NTU and therefore it is not directly applicable for situations where phase change occurs, i.e. where the variation of the heat transfer coefficient of the heat exchanger cannot be regarded as uniform throughout the exchanger.

Brouwers and Van Der Geld (1996) were looking for a method for optimizing heat exchanging surface area, in order to minimize heat exchanger cost, by developing an accurate model of a heat exchanger taking into account the influences of condensation and fog formation in the heat exchanger. First they developed a model for heat transfer without condensation and fog formation based on energy balance equations and using the NTU-approach. Then they moved on and developed a numerical method for solving the problem when condensation/fog formation occurs, i.e. based on energy balances taking into account mass fluxes, liberation of latent heat etc. The numerical model was devised to work with two different film models, i.e. a compound film model and an asymptotic film model, in order to evaluate their usability. This showed that that the two methods produced identical results, but also that the asymptotic model would require double the computational time. The model results were compared to experiments, and they found that the fog film models did not always correspond to actual condensation, as sometimes, especially for high values of the vapor mass fraction, the condensation would be drop wise, and this would result in a slight overestimation of heat exchanger performance. Otherwise their model was proven to be quite accurate.

In addition to these, there have been studies where the LMTD and  $\varepsilon$ -NTU approaches have been combined. Below are a few examples of some of these studies.

Wang and Sundén (2003) developed a model for designing/optimizing a plate heat exchanger by the use of both the LMTD- and NTU-approach, but their method was developed specifically for avoiding the many trials that are often necessary when using these methods, because of the necessity of meeting the pressure drop constraints. By using the allowable pressure drops as a design objective, they avoided the many trial iterations typically needed by other methods. The thermal-hydraulic model linking pressure drop and heat transfer for a shell-and-tube heat exchanger existed in the literature, and the authors extended it to plate heat exchangers. The model proved useful for optimal design of plate heat exchangers, basing the design on either fixed allowable pressure drops or complete optimal design without pressure drop specifications. In the latter, pressure drops are economically optimized and it is guaranteed that pressure drops are fully utilized simultaneously.

Eirola et al (2002) developed a mathematical model for a single-pass cross flow heat exchanger under the restriction of dry surface heat transfer, and the NTU approach is used as a reference point for the developed method. They developed their model based directly on the differential equations governing the heat flows in the system, and using a discretization of the problem to obtain a numerical formulation of the problem. They compared the results of the model with results obtained using the NTU approach and found that there was a good agreement between results within the specified operating conditions.

### **Other approaches – analytical, numerical...**

Hrnjak et al (2002) developed a quasi-steady finite-volume model for frosting of a plain-fin-round-tube heat exchanger. The model was based on different assumptions and correlations taken from the literature, e.g. using Yonko and Sepsy's (1967) correlation for frost thermal conductivity. The purpose of their work was to develop and validate a model for frost growth on full-scale heat exchangers, covering a wide range of conditions, i.e. air supply temperature, inlet relative humidity, face velocity and refrigerant inlet temperature. The model was developed on the basis of experimental setup, and calculation results from the model was compared to experimental data for verification and a good agreement was found.

Galovic, Virag and Zivic (2003) did an analytical analysis of recuperative parallel flow, counter flow and cross flow heat exchangers, based on the relative entropy generation which is directly related to the heat changer effectiveness. The analytical solutions they presented cover situations where evaporation or condensation occurs in one or both streams, and they present analytical solutions for parallel flow and counter flow whereas an analytical-numerical solution is presented for cross flow.

Al-Nimr (1998) investigated the transient response of counter-flow and parallel-flow flat-plate and shell-and-tube heat exchangers with phase change, and derived expressions that can be used for evaluating different design parameters for heat exchangers. In order to simplify the mathematical description it is taken into consideration, that the heat transfer coefficient in the two-phase section is much higher than in the one-phase section, and therefore the mathematical formulation can be simplified significantly. In this case the phase change that was considered was either condensation or evaporation.

Willatzen et al (1998) developed a mathematical model describing the transient phenomena of two-phase flow heat exchangers based on the one-dimensional partial-differential equations representing mass and energy conservation, i.e. leaving out the momentum equations by assuming pressure drops to be negligible. In part one of this two-part paper, the focus is on moving-boundary formulation of two-phase flows with heat exchange. In Pettit et al (1998), the second part of the paper, the set of equations developed in part one are used for an evaporator and different case studies of transient behavior are examined.

Ribeiro and Andrade (2002) developed an algorithm for steady-state simulation of plate heat exchangers. The algorithm is developed on the base of a system of linear, first-order, ordinary differential equations with constant coefficients considering an overall heat-transfer coefficient, and the solutions are approximated by a linear combination of exponential functions. They validate the algorithm by comparing results with existing exact analytical solutions for simple cases and experimental data, and the validation proves successful with an approximate error of  $\pm 3\%$  when simulating a plate heat exchanger used for milk pasteurization.

Goryainov and Chernyshov (2003) developed a 2-dimensional model of a recuperative heat exchanger, and showed that the model produced results that are in satisfactory agreement with experimental data. The model covers parallel-flow heat exchangers, and it can be used to determine heat fluxes in different directions and the temperature at any point inside the heat exchanger.

### **Frost growth modeling**

Bensafi, Borg and Parent (1997) developed a computational model for detailed design of finned coils in plate-fin-and-tube heat exchangers. The program can treat single-phase, condenser and evaporator cases. The pressure drops are calculated using different correlations depending on type of flow, i.e. single-phase or two-phase, and the heat transfer coefficients are determined using correlations depending on type of flow, and whether condensation or evaporation occurs etc.

Chen, Thomas and Besant (2003) modify an existing validated numerical model for frost growth on heat exchanger fins in order to simulate a fan-supplied finned heat exchanger under refrigerating frosting conditions. They find that frost growth in refrigeration heat exchangers causes a reduction in the fin heat rate, the fin efficiency and that pressure drop increases, and they conclude that design selections for fan, fin spacing and fin thickness will alter the frost growth and cycle time between defrosts of heat exchangers.

Lee, Kim and Lee (1997) developed an analytical model for the formulation of frost growth on a cold flat surface by considering the molecular diffusion of water, and the heat generation due to the sublimation of water-vapor in the frost layer. Results obtained using the model was compared to experimental data, and these comparisons show that there is an average error of approximately 10 % in the determined frost thickness. At low inlet temperatures errors rise to approximately 20 %.

Kim, Yun and Min (2002) developed a model for frost growth and frost properties with airflow over a flat plate at subfreezing temperature. Based on measurements they developed an empirical correlation for average frost roughness, and used the modified Prandtl mixing-length scheme to calculate heat and mass transfer coefficients. The frost growth model is based on assumption of one-dimensional heat transfer in frost layer, perfect gas law and thermodynamic equilibrium conditions prevail at frost surface, frost density is uniform, frost roughness is evenly distributed and convection and radiation effects are negligible. The model showed good agreement with test data taken from the literature.

Tao, Mao and Besant (1994) presented numerical results of frost formation under freezer temperature conditions along with measurements of frost characteristics during the early growth period. Their focus is freezer applications where the air that flows across the heat exchanger is below the freezing point, and especially early stage frost growth on different materials. They conclude that frost deposition on non-metallic surfaces tend to have more uniform ice-particle distributions than metallic surfaces. The ambient humidity plays a significant role for the early stage frost deposition, whereas surface temperature and ambient velocity plays minor roles which could be indicating that the mass transfer coefficient is relatively constant for the Reynolds number range considered (2840–5680).

Seker, Karatas and Egrican (2004a, 2004b) published a two-part contribution concerned with frost formation on fin-and-tube heat exchangers. In the first part they perform numerical analysis on heat and mass transfer characteristics of heat exchangers during frost formation, and develop a numerical model. In the second part they validate their model by comparing results to experimental investigations. The model is formulated under certain simplifying assumptions: all local heat transfer surface temperatures are below the frost point, frost deposition is homogenous, quasi-steady-state, frost layer is characterized by average properties, frost thermal conductivity varies only with frost density, radiation between moist air and frost layer is negligible and one-dimensional heat and mass transfer is considered. The comparisons made in part two of the paper show a reasonably good agreement between the calculated UA-value (total conductivity) and the measured experimental UA-value, and also comparisons of the experimental pressure drops are in good agreement, especially when the heat exchanger has lower fin pitches.

## **Conclusion**

This literary study has focused on heat exchangers and especially the mathematical formulation of models for the theoretical study of condensation and frost formation in heat exchangers. From the literary study it can be concluded that huge amounts of research has already been performed in this area, and that there are a lot of different approaches for developing mathematical models to represent the heat transfer mechanism that occur in heat exchangers where condensation or frost formation occurs.

The primary focus of research in this field has been on heat exchangers used in the refrigeration industry, and only a very few investigations has focused on air-to-air heat exchangers for building ventilation. Highly efficient heat exchangers used for building ventilation will experience problems with condensation and frost formation in northern European and arctic climates, i.e. in the areas where the ventilation heat loss will typically be extremely large. Therefore

there is a need for developing mathematical models that can help analyze these phenomena in detail for specific heat exchangers, so that heat exchangers can be developed to either entirely avoid frost formation or have integrated energy-efficient methods for defrosting.

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## Bilag 2: Regnskab

### Projekt: Ventilationssystemer

BYG-DTU

Brovej 118, 2800 Kgs. Lyngby

Projektnummer: 25498

Projektleder: Svend Svendsen

Finansiering: Villum Kann Rasmussen Fonden

Bevillingsgivers journalnr.: 2000-743/48-0001

**Regnskabsperiode: 1. maj 2003 - 30. april 2005**

	Realiseret total	Rev. budget
<b>Indtægter:</b>		
VKR. fonden	660.000,00	990.000,00
Indtægter i alt	660.000,00	990.000,00
<b>Udgifter:</b>		
Forskertimer	760.253,55	905.517,44
Teknisk/administrativ bistand	34.671,17	56.482,56
Rejser	11.540,00	26.000,00
Drift og materialer	691,90	2.000,00
Øvrige		
Udgifter i alt	807.156,62	990.000,00
Projektsaldo i alt	-147.156,62	0,00

Kgs. Lyngby d. 13. maj 2005



Michael Berthelsen  
Administrationschef

## **Bilag 3: Artikel til Nordic Symposium on Building Physics, Reykjavik 13-15 June 2005**

### *Mechanical ventilation with heat recovery in cold climates*

*Jesper Kragh, Assistant Professor*  
*Department of Civil Engineering, Technical University of Denmark*  
[jek@byg.dtu.dk](mailto:jek@byg.dtu.dk)

*Jørgen Rose, Assistant Professor*  
*Department of Civil Engineering, Technical University of Denmark*  
[jro@byg.dtu.dk](mailto:jro@byg.dtu.dk)

*Svend Svendsen, Professor*  
*Department of Civil Engineering, Technical University of Denmark*  
[ss@byg.dtu.dk](mailto:ss@byg.dtu.dk)

**KEYWORDS:** *Mechanical ventilation, heat recovery, energy consumption, heat exchanger, defrosting.*

#### **SUMMARY:**

*Building ventilation is necessary to achieve a healthy and comfortable indoor environment, but as energy prices continue to rise it is necessary to reduce the energy consumption. Using mechanical ventilation with heat recovery reduces the ventilation heat loss significantly, but in cold climates like the Northern Europe or in arctic climate like in Greenland or Alaska these ventilation systems will typically face problems with ice formation in the heat exchanger. When the warm humid room air comes in contact with the cold surfaces inside the exchanger (cooled by the outside air), the moisture freezes to ice. The analysis of measurements from existing ventilation systems with heat recovery used in single-family houses in Denmark and a test of a standard heat recovery unit in the laboratory have clearly shown that this problem occurs when the outdoor temperature gets below approximately  $-5^{\circ}\text{C}$ . Due to the ice problem mechanical ventilation systems with heat recovery are often installed with an extra preheating system reducing the energy saving potential significantly. New designs of high efficient heat recovery units capable of continuously defrosting the ice without using extra energy consumption are therefore suggested in this paper for future work.*

#### **Introduction**

There are basically two different methods of ventilating buildings, mechanical ventilation and natural ventilation. The energy performance of these two methods of ventilating, are to be improved with respect to both the use of heat and electricity. In single-family houses, the mechanical ventilation system has become more and more common because of its ability to fulfill the increasing demands for a healthy indoor climate.

A life cycle analysis of mechanical ventilation system with air-to-air heat recovery has been carried out in (Nyman M and Simonson C. J, 2005) and was found to be an environmentally friendly solution in cold climates (Helsinki, Finland) and that the greater the temperature efficiency, the more environmentally friendly the systems become. In (Palin S. L., McIntyre D. A. and Edwards R. E., 1996), mechanical ventilation with heat recovery is compared to natural ventilation and extracts fans, and is found to be the most effective

system for maintaining a low humidity level. However, mechanical ventilation in cold climates with highly efficient heat recovery also presents some problems; ice formation in the heat exchanger and electricity consumption for the fans. In (Ninomura P. T. and Bhargava R, 1995) the problem with ice formation in the heat exchanger for ventilation systems in arctic climates is recognized, but only the preheating of the supply air is discussed as a possible solution, and rejected due to the findings of (Phillips E. G., Chant R. E., Fisher D. R. and Bradley B. C, 1989), that suggests that this solution significantly reduces the recovered energy. Highly efficient fans have already been developed and in (Berry J, 2000) it is recommended that fan power input is less than  $1 \text{ W} \cdot \text{l}^{-1} \cdot \text{s}^{-1}$  for highly efficient mechanical ventilation systems. Investigations on natural ventilation with heat recovery (Skåret E., Blom P. and Hestad T, 1997) have shown that these types of system require assisting fans to work properly, hereby significantly reducing the energy saving potential. Natural ventilation without heat recovery is not suitable for use in arctic climates, due to the cold supply air creating drafts and severe increases in ventilation heat loss.

### **Building ventilation**

When designing ventilation systems for buildings, it is necessary to consider several different aspects and take into account the different demands concerning the overall functionality of the building. The primary focus should be kept on securing the necessary air change rate in order to both achieve a healthy indoor environment for the inhabitants (avoiding the so-called sick-building-syndrome, SBS) and at the same time securing that the building constructions are not exposed to destructive levels of moisture in the air.

When dealing with mechanical ventilation systems with heat recovery, it is also important to choose a system that is suitable for the climate in which it should function. For cold climates like northern Europe or Greenland (arctic climate) ice formation in the heat exchanger can stop the exhaust airflow. This will severely influence the ventilation of the building. Furthermore, focus should be on the extra energy used for the fans in the ventilation system, and minimizing this is desirable.

### **Mechanical ventilation with efficient heat recovery**

Mechanical ventilation with efficient heat recovery consists of two fans, a heat exchanger, filters, ducts, inlet and outlet diffusers and a controlling system. Using a heat exchanger with high efficiency will typically reduce the ventilation heat loss by 80-90% and the total heat loss by 30-60%, depending on the insulation level of the house etc. Outlets are normally placed in the rooms where moisture, odour and other pollutants of the indoor environment are produced, i.e. the kitchen, bathrooms and scullery, and the inlet diffusers are placed in rooms where the people are present over periods of time, i.e. the living room and bedrooms. In this way the moisture and odour from cooking and bathing etc. is removed effectively without polluting the surrounding rooms, and fresh air is blown into the building, providing a good indoor climate. The disadvantages of using mechanical ventilation systems are higher installation costs, necessary space for the components and ducts and very importantly, the electricity consumption by the fans. In the design phase for a ventilation system the attention must always be on minimizing the pressure loss in the system, as this is directly proportional to the electricity used by the fans.

## **Natural ventilation**

Natural ventilation systems are normally driven by the buoyancy force, where the temperature difference between inside and outside, or in special designs the wind pressure, drives the ventilation. The advantage of using natural ventilation is that there is no electricity consumption in the system (no fans) and lower installation costs compared with traditional mechanical ventilation systems. The disadvantages are that it is often difficult to control natural ventilation, i.e. the air change rate, and attention must be given to check if the required ventilation rate is fulfilled at all times.

Natural ventilation with heat recovery is rarely seen and very difficult to construct because of the conflict between the use of the temperature difference as a driving force and the equalization of temperature in the heat exchanger. Because of the large temperature difference between the inside and outside air in cold climates, natural ventilation will result in very large ventilation heat losses, and preheating the inlet air will be necessary if draught is to be avoided. Traditional preheating systems, i.e. using heating coils, are assumed to be unacceptable in this project due to the extra energy consumption that this implies. Other methods of preheating the air could be achieved by solar heating, but solar radiation is not always available.

The challenge in using a natural ventilation system in cold climates is therefore to develop and design a system that allows for preheating the inlet air without using extra energy. Natural ventilation systems will not be examined further in this context, as this work focuses on systems with efficient heat recovery.

## **Experiences with mechanical ventilation with heat recovery**

In general, there are three major problems that should be addressed when using ventilation systems with highly efficient heat recovery (energy efficiency of approximately 90%) in cold climates, i.e. northern Europe or arctic climates: freezing in the exchanger, use of electrical energy for the fans and draught due to low inlet temperatures.

At the Technical University of Denmark, a research project is presently being carried out, where mechanical ventilation systems with heat recovery are being analyzed. Measurements have been carried out on both existing systems used in different single-family houses in Denmark and in the laboratory. In-situ measurements have been carried out during the winter in order to evaluate problems with condensation/ice and risks of draught due to low inlet temperatures.

### **Ice formation in the heat exchanger**

Mechanical ventilation with heat recovery in cold climates can present problems with ice formation in the heat exchanger. When the warm humid room air is brought in contact with the cold surfaces of the exchanger (cooled by the outside air), the moisture at the exhaust air condensates in the heat exchanger. If the outside air temperature is below zero, the water vapour can then freeze to ice resulting in a pressure rise on the exhaust side of the heat exchanger, which in turn decreases the air flow through the exhaust side. The decrease in the amount of warm air through the exchanger will result in the exchanger being cooled further, and the system will eventually have to stop.

Temperature measurements shown in Fig. 1 were performed on a typical Danish single-family house during the winter of 2003-2004. The heat exchanger unit used in the house has a built-in feature to avoid ice in the system. If the cooled exhaust air temperature is

below 3°C, the units control system will reduce the inlet airflow without reducing the outlet airflow, until the cooled exhaust air temperature is above 5°C.

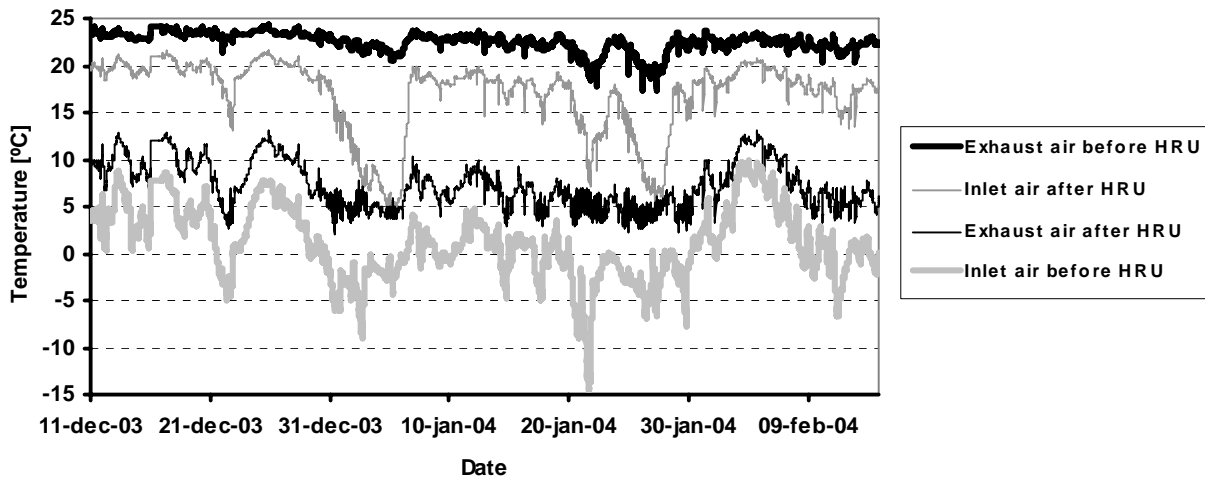


FIG. 1: Measured temperatures in heat recovery unit (HRU). Typical Danish winter.

From Fig. 1 it is evident that from 02-01-2004 – 07-01-2004 and again from 21-01-2004 – 29-01-2004, the heated inlet air temperature makes a sudden drop and at the end of the periods reaches a level below 5°C. This drop in the inlet temperature is caused by ice in the heat exchanger. It can also be seen from Fig. 1 that the cold inlet air temperature, at the time when the heat exchanger freezes, has reached a minimum of approximately –9°C and –15°C.

It is important to notice that the temperature of the cooled exhaust air never goes below 3°C, because the built-in feature to avoid ice in the heat exchanger is initiated. This indicates that this type of system is functional in cold climates but would have serious problems in arctic climates where the temperature is continuously below 0 °C for long periods of time (months). The reason why the system starts working again is that the outside temperature rises significantly. In Fig. 2, the temperature efficiency of the heat exchanger is shown during the same period.

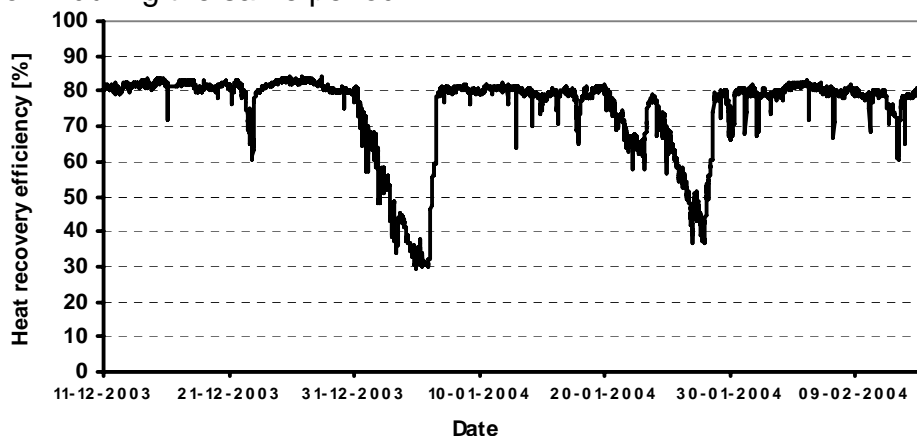


FIG. 2: Temperature efficiency of heat exchanger. Typical Danish winter.

From Fig. 2 it is evident that the freezing in the heat exchanger has a dramatic effect on the temperature efficiency of the heat exchanger. In this case the heat exchanger efficiency is reduced significantly, to a level of 30% and 40% respectively, for periods of 5 and 8 days respectively, and during this time the inlet air temperature is extremely low.

This example shows that the Danish climate during winter can sometimes give rise to problems with typical heat exchangers, but in arctic climates, where the temperature is significantly lower, these problems can be more severe, and it can be impossible to use mechanical ventilation with heat recovery during the coldest periods.

At Building Energy and Services Engineering a laboratory test has been performed on a typical heat recovery unit in order to investigate how the airflow is influenced by the condensation and ice in the heat exchanger. The unit consists of two ventilators and a counter flow heat exchanger made of plastic with an expected temperature efficiency of 90%. The built-in feature to avoid ice in the system was deactivated during the test. The test was performed using cold inlet air at  $-5^{\circ}\text{C}$  and warm exhaust room air at  $21^{\circ}\text{C}$  (relative humidity of air: 42%). As shown in Fig. 3 the exhausted airflow after only 2 hours starts to fall due to the ice inside the exchanger. This means that in cold climates the built-in feature to avoid ice in the heat exchanger would be initiated continuously influencing the airflow rates significantly and thereby the recovered heat.

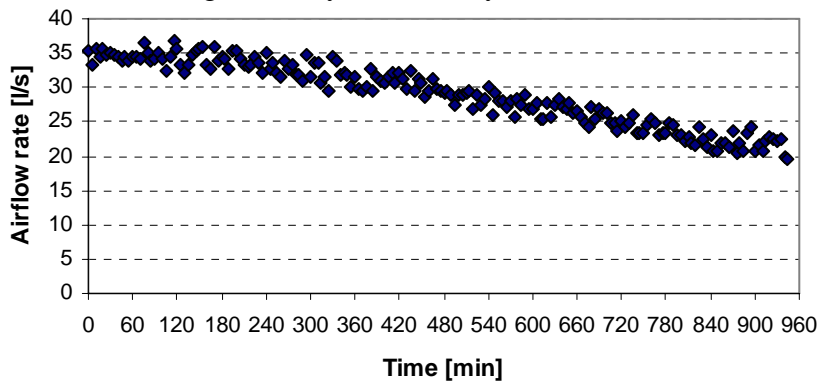


FIG. 3: Measurements of the exhaust air flow rate of a typical heat recovery unit at low inlet temperature ( $-5^{\circ}\text{C}$ ).

The analysis of measurements from existing ventilation system with heat recovery used in single-family houses in Denmark and a test of a standard heat recovery unit in the laboratory have clearly shown that problems occur when the outdoor temperature gets below approximately  $-5^{\circ}\text{C}$ . This corresponds well with findings in (Veltkamp B, 1996) where the theoretical freezing temperature for a counter flow heat exchanger has been calculated as  $-8^{\circ}\text{C}$ . Therefore development of new designs of heat recovery units able to function all year round in cold or very cold climates is desirable.

### Indoor climate, low inlet temperature, draught

Another problem that may occur as a function of a low outside temperature is draught due to a low inlet temperature from the system. Whether or not a given inlet temperature will be felt as draught in the room depends on a lot of factors such as the person, the inlet air speed, the design of the diffuser, where the diffuser is mounted etc., and therefore it can be hard to set the rules for what the acceptable minimum inlet temperature should be. A general rule of thumb says that the inlet temperature should be higher than  $15^{\circ}\text{C}$  to avoid draught, and this temperature is used in the following examples as a measure for the ventilation systems performance.

With a minimum inlet temperature of  $15^{\circ}\text{C}$ , it is possible to calculate what circumstances will produce inlet temperatures that may cause problems. If, for instance, we have an inside temperature of  $20^{\circ}\text{C}$  and a heat recovery unit with a 90% temperature efficiency and

balanced airflows, the minimum outside temperature can be determined as  $-30^{\circ}\text{C}$ . In Denmark the outside temperature will almost never go below  $-30^{\circ}\text{C}$ , and therefore a heat recovery unit with 90% temperature efficiency will not present problems with draught. If, however, the efficiency of the heat recovery is only 85% or 80%, the minimum outside temperature will be approximately  $-13^{\circ}\text{C}$  and  $-5^{\circ}\text{C}$  respectively, and this would mean that there might be times when the inlet temperature is lower than the accepted  $15^{\circ}\text{C}$  minimum. This shows that the temperature efficiency of the heat exchanger is extremely important, and if the outside temperature is below the minimum, it is necessary to have some kind of feature in the system to avoid low inlet temperatures that can produce draught, e.g. a heating coil that can preheat the inlet air.

In one of the single-family houses that have been analyzed during the project mentioned above, the heat recovery was expected to have a temperature efficiency of 85-90%. However, measurements on the system have shown that the efficiency was a bit lower, i.e. 81%, and therefore it is interesting to see how this affects the inlet temperature in the house. In Fig. 4 the temperatures for the system are shown (the data covers the winter of 2003-2004).

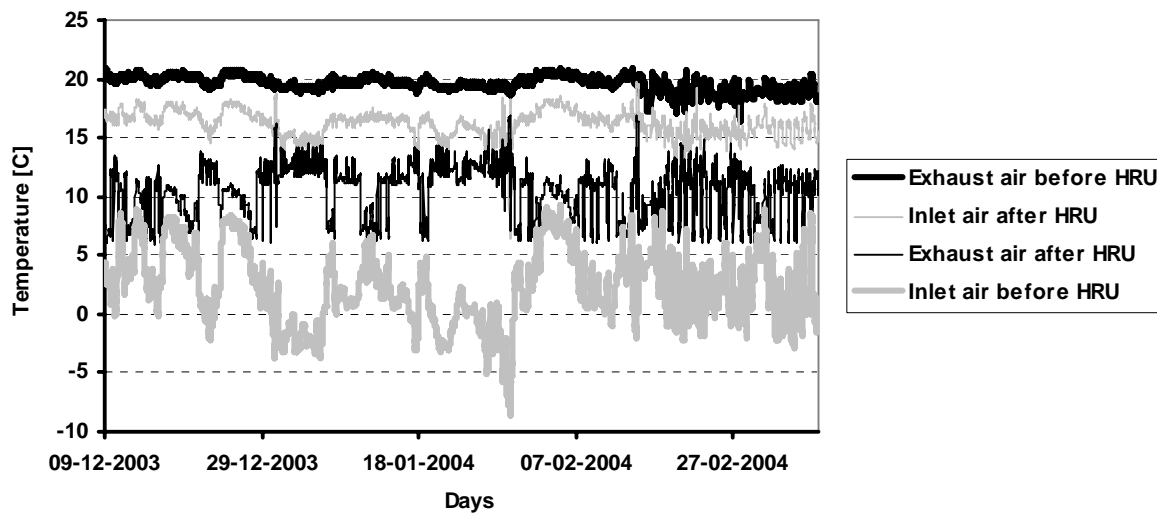


FIG. 4: Temperature measurements for heat recovery ventilation system.

From Fig. 4 it is evident that the inlet air temperature is typically between  $15^{\circ}\text{C}$  and  $20^{\circ}\text{C}$ . However, when the outside temperature reaches the freezing point, the inlet air goes below  $15^{\circ}\text{C}$ . The fluctuations at the end of the period occur as a result of the inlet air bypassing the heat recovery unit. The system does this whenever the outside temperature is higher than  $16^{\circ}\text{C}$ .

In another house, the efficiency of the heat recovery unit was also expected to be between 85% and 90%. However measurements showed that the efficiency was as low as 77%. In this house, during the winter of 2001-2002, the inlet temperature would sometimes go below  $10^{\circ}\text{C}$ , which would normally suggest problems with draught for the inhabitants.

From the investigations performed on typical Danish single-family houses it is evident, that the risk of draught from mechanical ventilation systems with heat recovery is present when the temperature efficiency of the heat recovery unit is significantly lower than 90%. It is also evident that typically the existing heat recovery units do not live up to such expectations. Therefore it is necessary to develop heat recovery units that have higher temperature efficiencies and to document their performance.

If a unit has a temperature efficiency of 80% or less, there is a need for preheating the inlet air. Using either an electrical heating coil or water heated coil after the heat recovery unit is one way of dealing with the problem, however this method is energy consuming, and especially in arctic climates the energy consumption for pre-heating the ventilation air would be quite significant on a yearly basis (approximately 2000 kWh for a 100 m<sup>2</sup> house in Nuuk), and therefore new methods of avoiding low inlet temperatures should be investigated.

## **New design of heat recovery units without freezing problem**

Preheating the inlet air could be a way of dealing with problems concerning both draught and freezing in the exchanger. Several methods of preheating the inlet air have been tried and tested, and in the following sections some of these are described.

The primary focus when designing a highly efficient heat recovery unit should be on reducing the energy consumption of the fans and at the same time securing that freezing in the exchanger is avoided. Reducing the energy consumption of the fans in the system can generally be achieved in two ways - by designing the duct system so that the total pressure loss in the system is minimal, and by using/developing highly energy efficient ventilators, i.e. as good as or better than existing solutions.

Avoiding freezing in the heat exchanger is, as can be seen by the previously described simulations and measurements, essential when dealing with cold climates. If freezing occurs every time the outside temperature is below zero, then a heat recovery unit will not be cost-effective.

In the following several solutions of freeze protection is considered for the future work.

### **1) A low efficiency exchanger**

Low efficiency exchangers don't freeze if the extracted air isn't cooled below 0°C. This could be the case for exchangers with low temperature efficiency. As the primary focus in this work is on energy savings, it's considered unsuitable for future development of a freeze protected heat recovery unit.

### **2) Control strategies**

-Bypassing or stopping the airflows until the exchanger is defrosted again could work in semi cold climates like the northern European. Tests have shown that defrosting an exchanger filled with ice can take several hours and is therefore not serviceable in arctic climates.

-Preheating the inlet air is often used in today's mechanical ventilations system with heat recovery. As previously described the extra energy consumption makes this solution uninteresting.

### **3) Munter's solution**

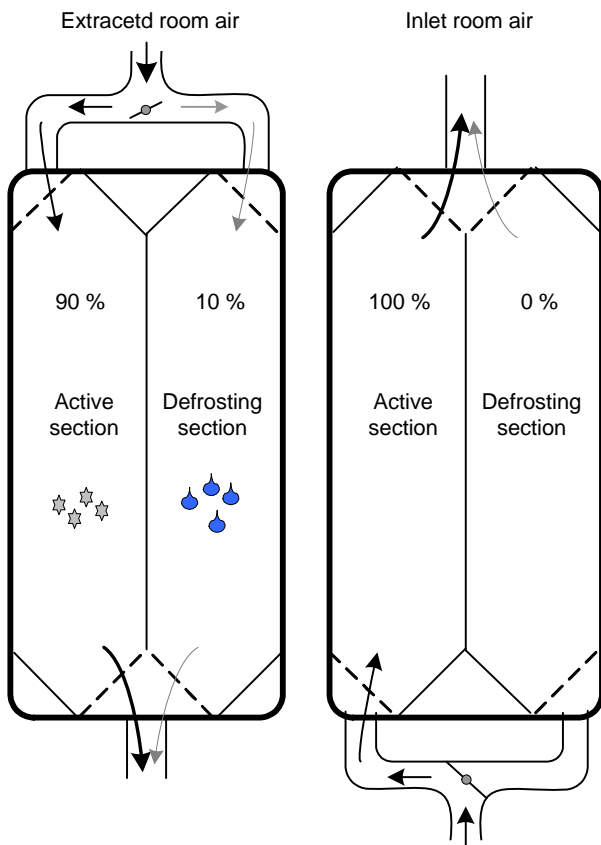
The well known Munter's solution (Munters AB Carl, Haellgren, K, 1981) where a small section of the exchanger is inactive due to a plate continuously moving across the inlet airside blocking a small part of the opening to the exchanger. In this way the exchanger has time to defrost the inactive section.

### **4) Serial connection of two exchangers**

Two exchangers coupled in serial connection. When freezing occurs in one of the exchangers, the flow direction is switched and the frozen exchanger will start defrosting. The extra cost of having two exchangers, extra valves and a control system is of course a serious limitation of this design and therefore it may not be a cost-effective way of dealing with the problem.

### 5) Alternating flow in two parallel coupled heat exchangers

A large counter flow box exchanger made of polycarbonate plate could also be an option. Recognizing that condensing water cannot be avoided, the exchanger has to be designed in a way that allows for easy removal of the water without extra pumps or valves. By having an exchanger with vertical airflows, the extracted room air is cooled down from the top to the bottom of the exchanger. In this way the condensing water will simply fall to the bottom of the exchanger due to both the gravitation and the airflow pressure. From the bottom of the exchanger the water simply drops to the ground or, if possible, through ducts coupled to the drain. To gain the high efficiency of a counter flow heat exchanger, the inlet air then has to flow from the bottom to the top of the exchanger. The exchanger consists of two equal sections, which by turn is active or inactive. Two electrical valves control the airflows to the two sections. The airflow rate of the extracted room air is adjusted so that the flow through sections 1 and 2 is 90% and 10% respectively. After an adjusted time interval the airflows switch so that the flow in section 1 is 10% and in section 2 90%. The inlet airflow also switches between the two sections but is always either 100% or 0%. The idea is then that the 10% of airflow (warm extracted air) is able to defrost the inactive part of the exchanger. A sketch of the exchanger design is shown in Fig. 5.



*FIG. 5: Drawing of the exchanger at two sectional views. To the left is shown the exhaust airflow and to the right the inlet airflow.*

The serial connection of two counter flow exchangers and the large box exchanger has been selected for further analysis and testing under laboratories facilities. Each exchanger design and test measurement will be described in coming articles.

## Conclusion

Based on measurements and calculations of existing building ventilation systems with heat recovery, it can be concluded that there is a need for developing new heat recovery systems with both high temperature efficiency and protection against ice formation, if the systems are to be used effectively in cold climates.

Measurements from typical single-family houses and on a heat exchanger in the laboratory have shown that ice formation occurs when the outside temperature reaches a level below  $-5^{\circ}\text{C}$  for a short period of time. Measurements also show that the inlet temperature at times reaches levels below  $15^{\circ}\text{C}$ , which means that draught can occur.

The findings in measurements have led to a number of suggestions concerning the development of new heat exchangers with high temperature efficiency, low electricity use for fans and protection against ice formation.

## Acknowledgement

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## Bilag 4: Præsentation ved symposium i Sisimiut

# Mechanical Ventilation System with Heat Recovery in Arctic Climates

Jesper Kragh  
Assistent research professor  
Technical University of Denmark

Energy-efficient building  
April 12<sup>th</sup> – 14<sup>th</sup> 2005 · Symposium in Sisimiut

## Agenda

- Ventilation in normal houses
- Ventilation heat loss and energy saving potential
- Heat recovery in cold climate – freezing problem
- The ventilation system in the low-energy house
- Test of the heat recovery unit at DTU
- Other solutions of the freezing problem
- Conclusion

## Why ventilate ?

- Water vapour (10-15 l pr. family pr. Day)
- Scent from people
- Gasses from furniture and building components
- High indoor temperature

### Insufficient ventilation

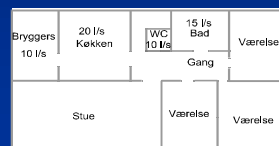
- Poor indoor climate
- Higher risk of mould and rot in building constructions

## Ventilation demands in the coming building code

Total air change rate:  $0.5 \text{ h}^{-1}$

Extracted air flow demands:

Kitchen	20 l/s
Bathroom	15 l/s
Toilet	10 l/s
Scullery	10 l/s



Example: House of  $100 \text{ m}^2$ .

Outside temperature  $-10^\circ\text{C}$  and inside  $+20^\circ\text{C}$ .

Ventilation heat loss:  $1.0 - 1.2 \text{ kW}$

## The heat recovery unit

The energy in the extracted air is used to heat the cold inlet air (from the outside).

The energy saved depends on the heat exchanger's efficiency.

Cross flow heat exchangers	? = 60 – 70 %
Rotating heat exchangers	? = 60 – 80 %
Counter flow heat exchangers	? = 85 – 95 %

**Example of small compact heat recovery unit with ventilators, heat exchanger, filters and control system (? = 90 %)**



## Test of small compact standard heat recovery unit

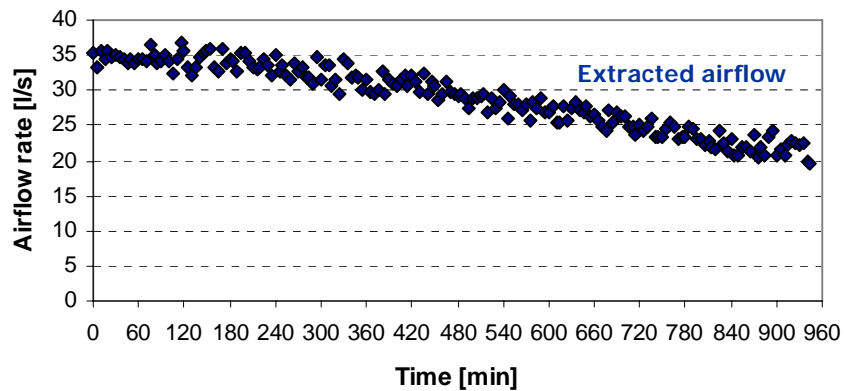
Efficiency of the heat exchanger:  $\eta = 90\%$

Room temperature:  $21^{\circ}\text{C}$

Relative humidity:  $42\%$

Outside temperature:  $-5^{\circ}\text{C}$

Measured exhaust airflow  
(16 hours)

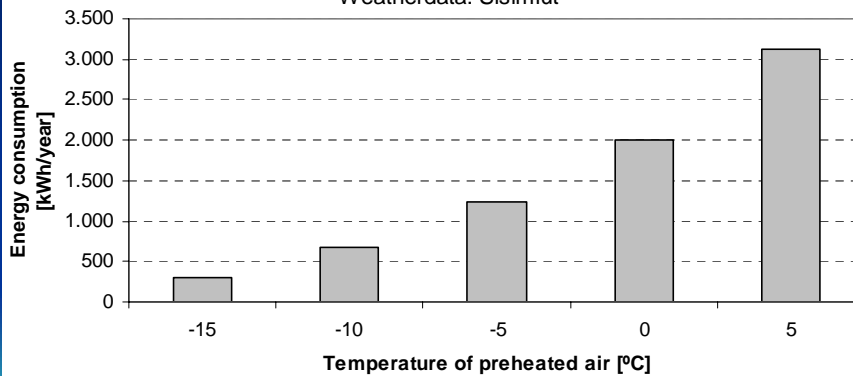


## Preheating the inlet air

### Energy consumption for preheating the inlet air

(House of  $100\text{ m}^2$  with an air change rate of  $0.5\text{ h}^{-1}$ )

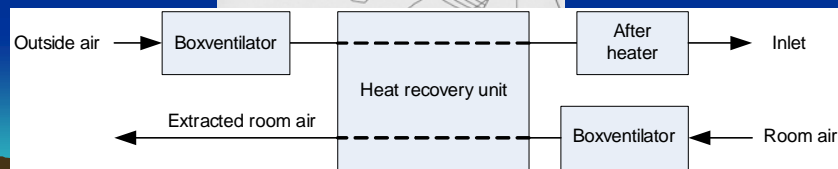
Weatherdata: Sisimiut



## Annual energy saving potential (100 m<sup>2</sup> house in Sisimiut)

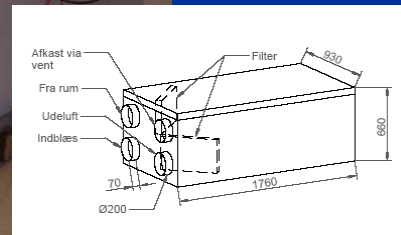
	Ventilation heat loss	Preheating	Fan energy	Savings
	kWh	kWh	kWh	kWh
Ventilation with no heat recovery	9,000	0	0	0
Today's solution: Heat recovery (60%) Preheating (-10°C)	3,300	700	600	4,400
Improved system: Heat recovery (90%) No preheating	900	0	600	7,500

## The ventilation system in the low-energy house

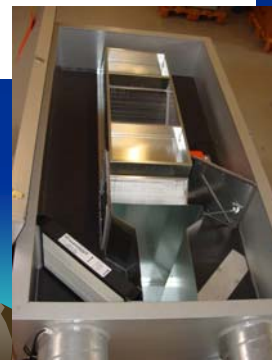
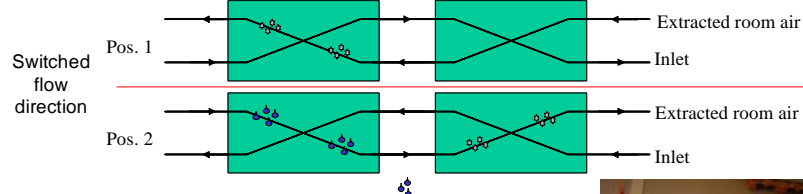


## The ventilation system in the low-energy house

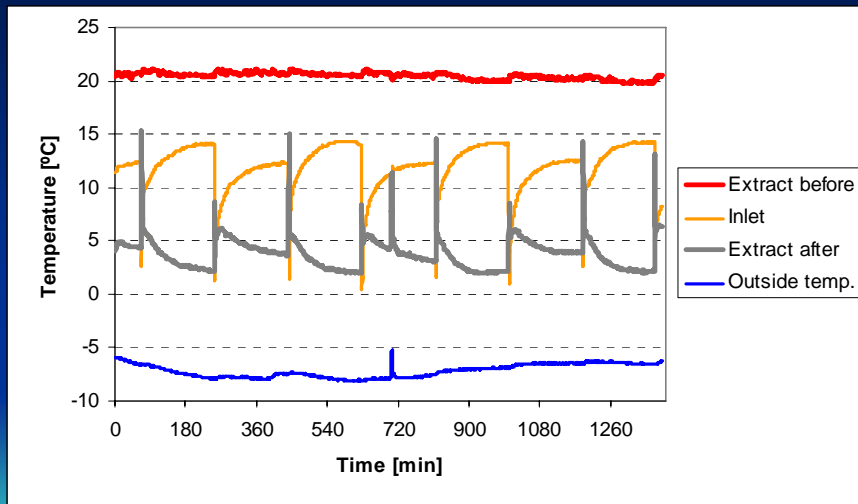
Heat recovery unit developed for the low-energy house in Sisimiut in cooperation between EXHAUSTO A/S and the Technical University of Denmark



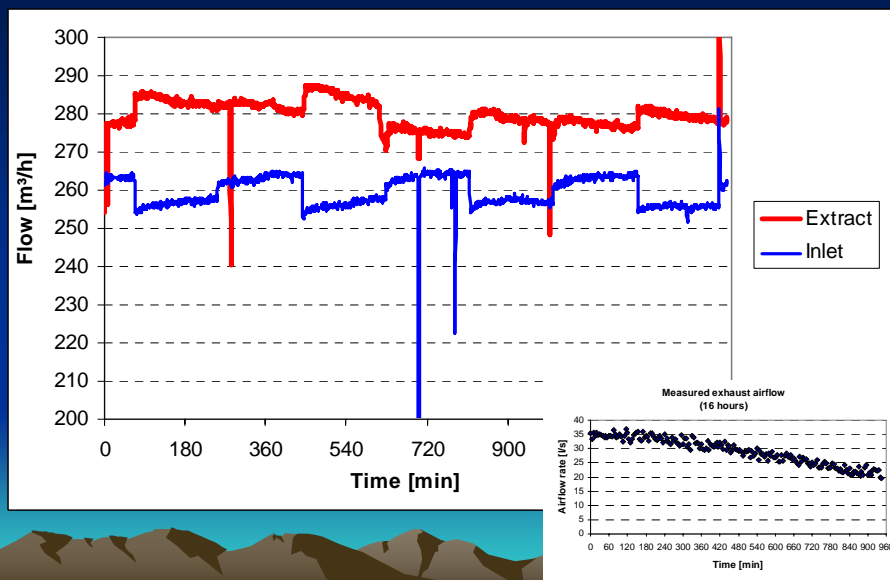
## The ventilation system in the low-energy house



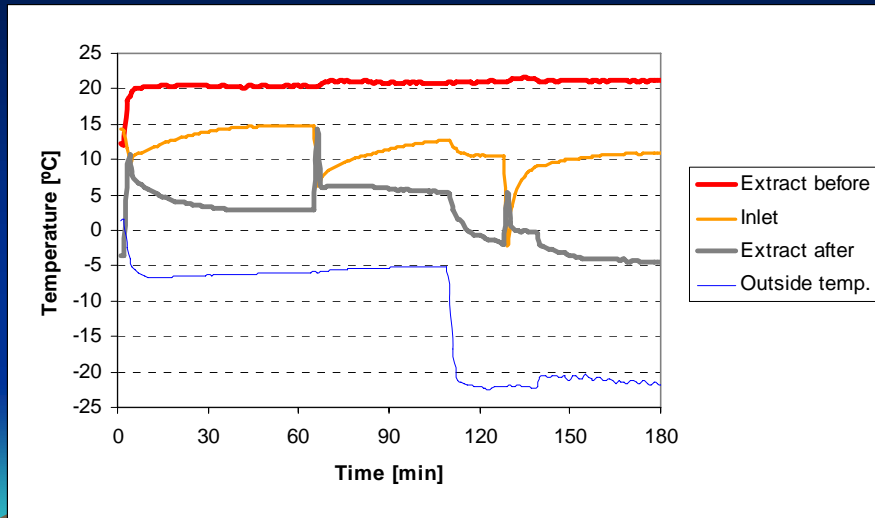
## Laboratory test at DTU



## Laboratory test at DTU



## Laboratory test at DTU



## Other solutions of the frost protection

- Munter's sledge solution
- Serial connection of two exchangers
- Alternating flow in two parallel coupled heat exchangers

## Other solutions of the frost protection

### Munter's sledge solution

A small section of the exchanger is inactive due to a plate continuously moving across the inlet airside blocking a small part of the opening to the exchanger.

In this way the exchanger has time to defrost the inactive section.



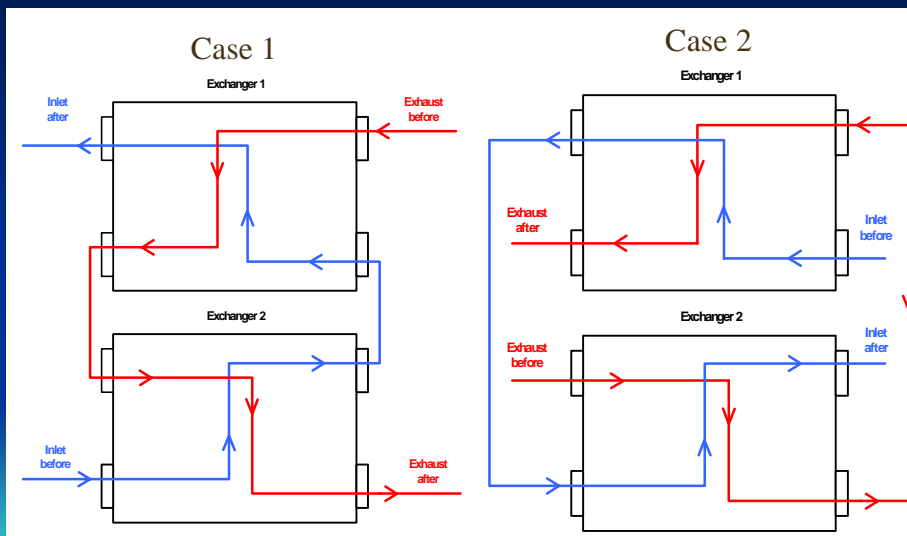
## Other solutions of the frost protection

### Serial connection of two exchangers

Two exchangers coupled in serial connection. When freezing occurs in one of the exchangers, the order of the exchangers is switched around. The frozen exchanger will start defrosting.

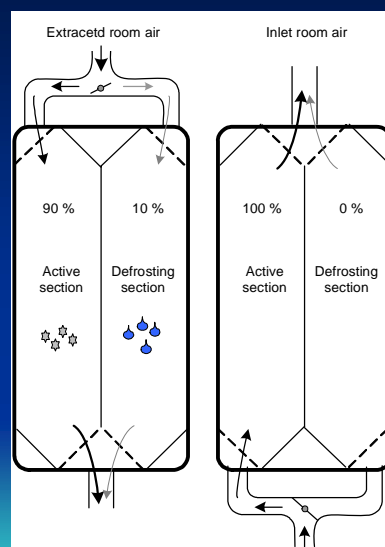


## Serial connection of two exchangers

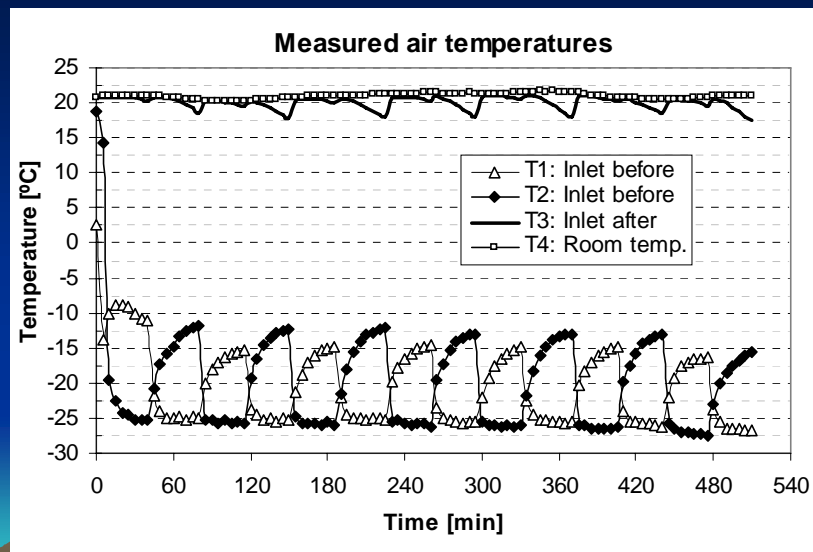


## Alternating flow in two parallel coupled heat exchangers

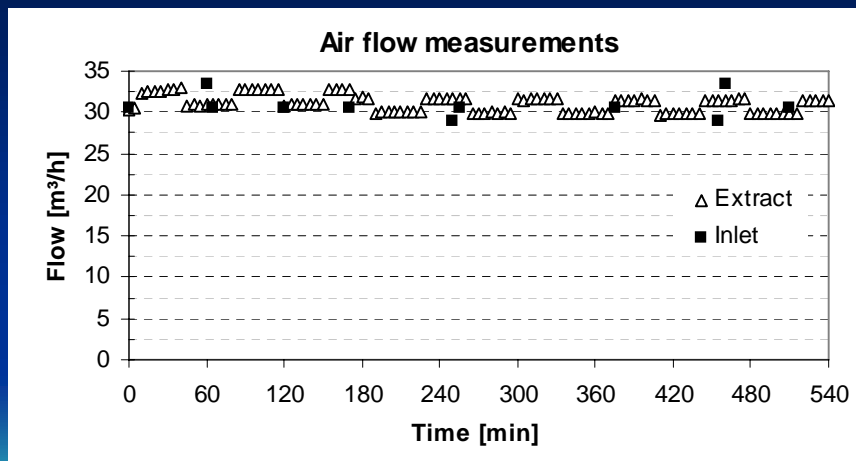
A large counter flow box exchanger made of polycarbonate plate. Since condensing water cannot be avoided, the exchanger has to be designed in a way that allows for easy removal of the water without extra pumps or valves.



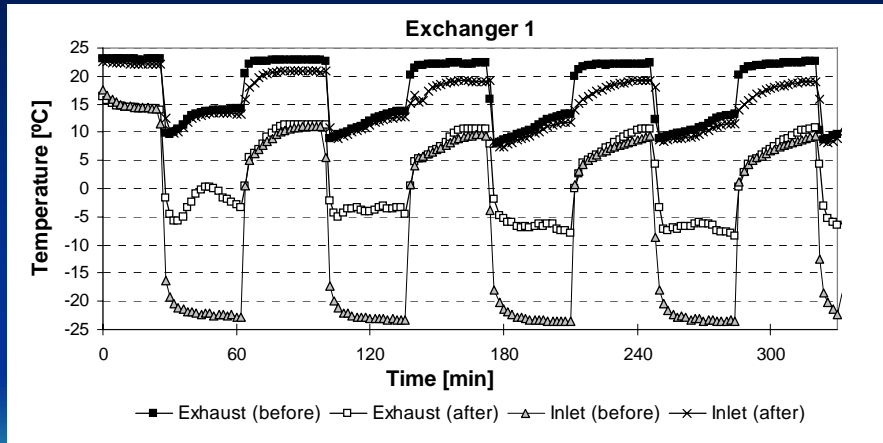
## Other solutions of the frost protection



## Test of the alternating heat recovery unit



## Test of Serial connection of two exchangers



## Conclusion

In cold climates the potential of reducing the ventilation heat loss is huge.

A standard heat recovery unit is not suitable for the cold arctic climate where the low outside temperature easily causes ice formation in the heat exchanger.

A new heat recovery unit was designed and developed for the low-energy house in Sisimiut in cooperation between EXHAUSTO A/S and the Technical University of Denmark. The developed heat recovery unit is capable of continuously defrosting itself.

The defrosting system was tested under laboratory facilities showing that the system worked as intended but still there was a need for extra after heating to minimize the risk of draft for the occupants