

Bæredygtigt arktisk byggeri i det 21. århundrede - Energirigtige ventilationssystemer

Statusrapport 3 til
VILLUM KANN RASMUSSEN FONDEN

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Energirigtige ventilationssystemer

Statusrapport 3

VILLUM KANN RASMUSSEN FONDEN

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Forord

Nærværende rapport er statusrapport 3 for projektet med titlen Energirigtige ventilationssystemer støttet af VILLUM KANN RASMUSSEN FONDEN. Projektet er udført i perioden 2003 til november 2006 på BYG·DTU, Danmarks Tekniske Universitet.

Projektet er tidligere blevet afrapporteret i statusrapport 1 (maj 2004) og statusrapport 2 (maj 2005). I denne rapport gives indledningsvis en beskrivelse af forskningsindholdet og hvilke problematikker der er arbejdet med gennem projektet. Herefter følger en beskrivelse af hvad der har været arbejdet med siden statusrapport 2.

Rapporten er desuden et udkast til den endelige slutrapport der fremsendes november 2006. Slutrapporten vil hovedsagelig være den samme, men dog opdateret mht. artikler der i øjeblikket er under publicering i internationale tidsskrifter.

Jesper Kragh
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Maj 2006
Danmarks Tekniske Universitet
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Bilag 3 Mechanical Ventilation System with Heat Recovery in Arctic Climates

Bilag 4 Mechanical ventilation with heat recovery in cold climates

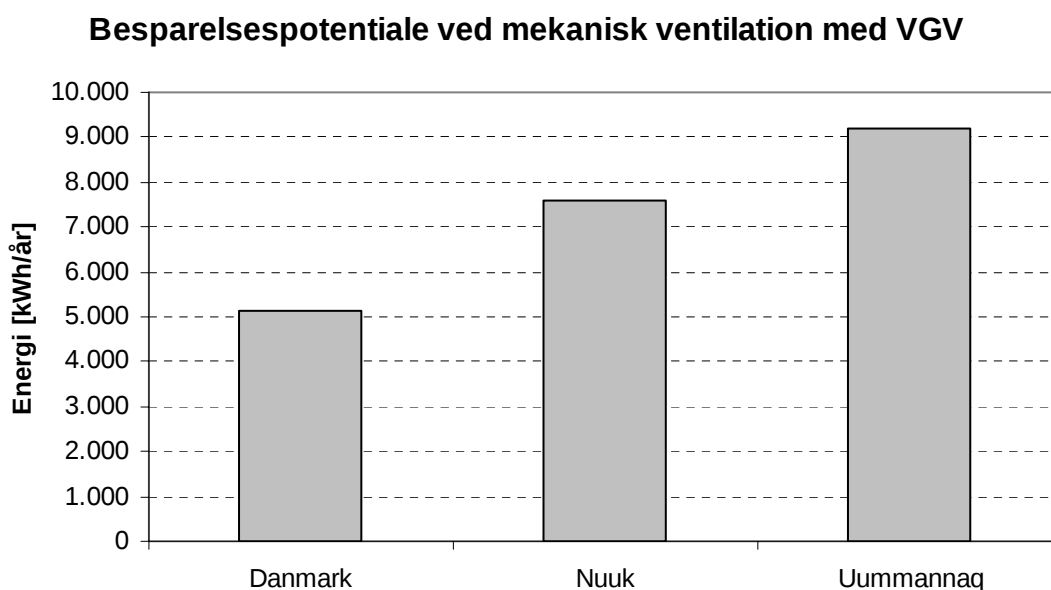
Bilag 5 New counter flow heat exchanger designed for ventilation systems in cold climates

Bilag 6 Presentation at Energy-efficient building, April 12th – 14th 2005 · Symposium in Sisimiut

Bilag 7 Presentation at Nordic Symposium on Building Physics, Reykjavik June 13th – 15th 2005

1 Indledning

Anvendelse af mekanisk ventilation med højeffektiv varmegenvinding i nordeuropæiske eller arktiske klimaer, er en meget effektiv måde at reducere energiforbruget i bygninger på. En analyse af energibesparelspotentialt blev foretaget med danske (nordeuropæiske) og grønlandske (arktiske) referencevejrdataår samt to huse; et grønlandsk selvbyggerhus kaldet Illorput og et dansk typehus fra firmaet Lind & Risør. Figur 1.1 viser en simulering af besparelspotentialt i opvarmningsbehovet ved brug af ventilation med varmegenvinding (effektivitet 80%). Der antages et luftskifte som opfylder bygningsreglementets krav.



Figur 1.1 Simuleret besparelse i opvarmningsbehov ved brug af ventilation med varmegenvinding (VGV).

Som det ses af Figur 1.1 er besparelspotentialt i opvarmningsbehovet relativt stort, men det skal nævnes at elforbruget til ventilatorerne ikke er medtaget og at dette er afgørende for om besparelsen økonomisk er rentabel. Derfor er det altafgørende at der udvikles varmegenvindingssystemer med lavt tryktab og at der bruges energieffektive ventilatorer med lavt elforbrug.

Udfordringen ved anvendelse af mekanisk ventilation med højeffektiv varmegenvinding er imidlertid at konstruere og designe en varmeveksler, der ikke fryser til. Når den fugtige indendørsluft kommer i kontakt med en kold overflade med en temperatur under dugpunktstemperaturen for vanddampen i luften, forekommer kondens, og hvis temperaturen af overfladen er under frysepunktet vil der forekomme udskillelse af iskrystaller. I en varmeveksler vil udskillelse af iskrystaller på den varmevekslende overflade (på afkastsiden i varmeveksleren) medføre en reduktion af varmevekslerens effektivitet, svarende til at den totale varmeoverføringskoefficient reduceres, og samtidig vil der ligeledes forekomme et tryktab fordi isen blokerer luftens strømning gennem kanalerne. Med mindre der på dette tidspunkt igangsættes afrimningsmekanismer, vil varmeveksleren ende med at fryse helt til.

Tilrimningsproblemet er også indledningsvist påvist i projektet ved en analyse af måledata fra en række danske forsøgshusprojekter. Som forventet viste målingerne tydeligt at varmevekslerens høje effektivitet medfører tilisningsproblemer i kolde perioder når den fugtige fraluft (indeluft) køles ned til under frysepunktet. Samtidigt viser målingerne at indblæsningstemperaturen i kolde perioder falder markant, hvilket må formodes at medføre trækgener for beboerne.

Der findes forskellige metoder til at undgå/fjerne is i varmevekslere, men oftest vil disse metoder have en negativ indvirkning på varmevekslerens effektivitet eller medføre et ekstra energiforbrug, f.eks. ved forvarmning af udeluften til en temperatur, som sikrer at den varmevekslende flades temperatur holdes over frysepunktet. Derfor er der et stort behov for at analysere mulighederne for at udvikle mere energieffektive metoder til undgåelse/fjernelse af is i varmevekslere.

I de følgende afsnit benyttes følgende benævnelser jf. DSF 447 [1] for de fire luftstrømme: i et mekanisk ventilationsanlæg

Fraluft	Indeluft fra boligen før passage gennem varmeveksleren
Afkastluft	Indeluften efter passage gennem varmeveksleren
Udeluft	Udeluft fra det fri før passage gennem varmeveksleren
Tilluft	Frisklufttilførelsen til boligen efter passage gennem varmeveksleren

1.1 Resumé af testede afrimningsprincipper

I projektets startfase blev forskellige tekniske løsninger af tilisningsproblemet afprøvet og analyseret under laboratorieforhold. Efter test af de forskellige løsninger er konklusionen for de forskellige løsninger følgende:

Standard frostsikringen

Tidligt i projektet blev der udført en test af den i dag typisk anvendte frostsikringsmetode. Testen viste at denne metode, hvor volumenstrømmen langsomt reguleres ned (på tilluftsiden), i praksis vil medføre at veksleren aldrig kommer i drift når udetemperaturen er under frysepunktet. Metoden er derfor ikke egnet til kolde klimaer, hvor udetemperaturen i længerevarende perioder er konstant under frysepunktet.



Afrimningsprincip 1

Princippet benytter to seriekoblede vekslere med mulighed for skiftende rækkefølge i forhold til hvilken af de to vekslere der først "møder" den kolde udeluft. Efter et passende tidsinterval byttes om på rækkefølgen af de to vekslere, således at den kolde veksler efter et skift modtager den forvarmede udeluft fra den kolde veksler. Princippet resulterer dog i så komplicerede kanal- og spjældløsninger at et stort tryktab er næsten uundgåeligt og dermed også et stort elforbrug. Endvidere er der ekstra komponentomkostninger hvilket gør at løsningen samlet set ikke kan anbefales.

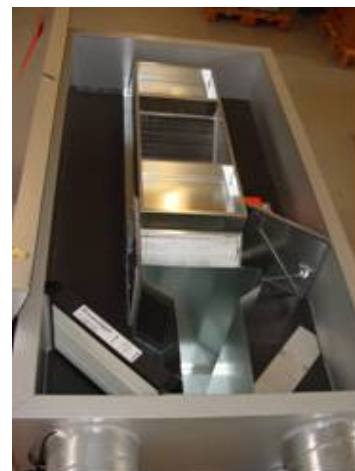


Afrimningsprincip 2

Igen benyttes to seriekoblede vekslere, men i stedet for at skifte om på vekslernes rækkefølge, som ved princip 1, skiftes der her om på flowretning gennem vekslerne.

Metoden er testet med en prototype af løsningen der er installeret i forbindelse med lavenergihus projektet i Sisimiut.

Laboratorieforsøgene viste at løsningen virker efter hensigten mht. afrimningsfunktionen, men forsøget viste også at indblæsningsluften umiddelbart efter spjældskift kan kræve supplerende opvarmning for at hæve indblæsningstemperaturen til et acceptabelt niveau mht. termiske komfort. Effektiviteten forringes også af varmekapaciteten i veksleren.



2 Rapporter og Artikler

Under projektet er følgende rapporter og artikler udfærdiget. Artiklerne er gengivet i bilag 3 til 5:

Rapporter:

Bæredygtigt arktisk byggeri i det 21. århundrede - Energirigtige ventilationssystemer
Statusrapport 1 – maj 2004, Danmarks Tekniske Universitet

Bæredygtigt arktisk byggeri i det 21. århundrede - Energirigtige ventilationssystemer
Statusrapport 2 – maj 2005, Danmarks Tekniske Universitet

Bæredygtigt arktisk byggeri i det 21. århundrede - Energirigtige ventilationssystemer
Statusrapport 3 – maj 2006, Danmarks Tekniske Universitet
(Foreløbig version af slutrapport – nærværende rapport)

Literature study – Heat exchangers, februar 2005, Danmarks Tekniske Universitet
Se bilag 2

Artikler:

Kragh J., Rose J., Svendsen S., 2005, Mechanical Ventilation System with Heat Recovery in Arctic Climates, Energy-efficient building, April 12th – 14th 2005 · Symposium in Sisimiut, Technical University of Denmark.
Se bilag 3

Kragh J., Rose J., Svendsen S., 2005, Mechanical ventilation with heat recovery in cold climates, Nordic Symposium on Building Physics, Reykjavik 13-15 June 2005, Technical University of Denmark.
Se bilag 4

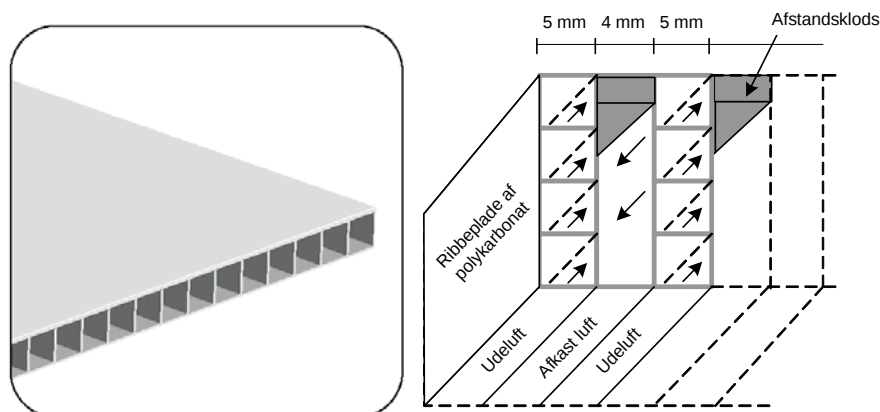
Kragh J., Rose J., Svendsen S., 2006, New counter flow heat exchanger designed for ventilation systems in cold climates, Technical University of Denmark.
Se bilag 5

Rose J., Kragh J., Svendsen S., 2006, Simulation model of a counter flow air-to-air heat exchanger with phase change, 2006, Technical University of Denmark.
(Under udarbejdelse)

3 Nyudviklet kassevarmeveksler

I projektet er udviklet og designet en ny type varmeveksler med indbygget afrimningsfunktion. Veksleren er forholdsvis stor sammenlignet med tilsvarende produkter, men består til gengæld af relativt billige byggematerialer.

Veksleren er opbygget af 10 stk. ribbeplader stillet lodret med ca. 4 mm lufthulrum mellem pladerne. Internt i veksleren strømmer den kolde luft i selve ribbepladens kanaler og den varme luft strømmer mellem pladerne, som vist på Figur 3.1.



Figur 3.1 Til venstre en ribbeplade af polykarbonat og til højre opbygningen af modstrømsveksleren

Udvendigt er selve vekslerens sider og top isoleret med hhv. 50 og 100 mm polystyren og beklædt med en 12 mm krydsfinerplade.

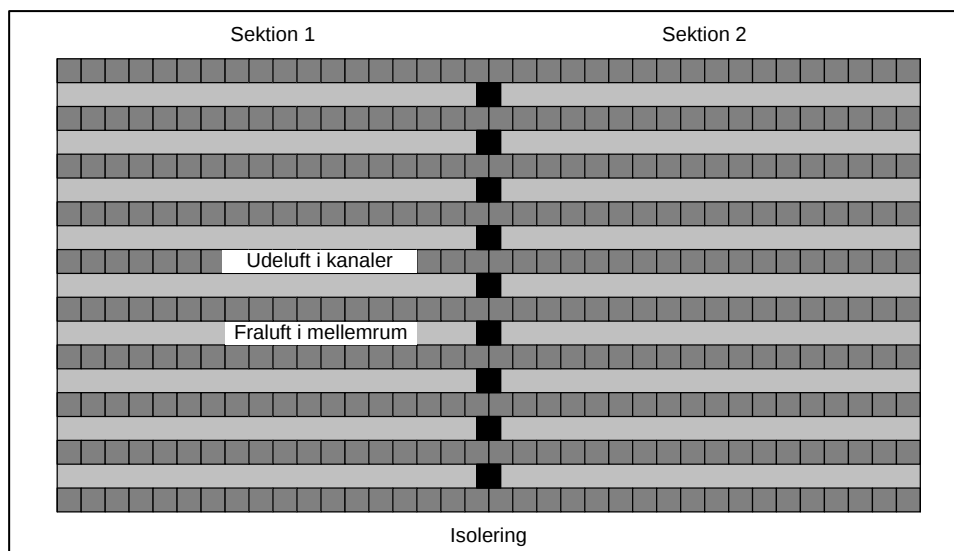
Vekslerens udvendige dimensioner er:

Højde	2260 mm	
Bredde	1300 mm	(inkl. 2 x 100 mm isolering)
Dybde	335 mm	(inkl. 2 x 50 mm isolering)

Varmevekslerpladernes dimension er:

Højde	2000 mm
Bredde	1000 mm
Tykkelse	5 mm

Parallelt med kanalerne er de 10 ribbeplader opdelt/adskilt i to lige store sektioner, idet der i mellemrummet mellem pladerne er lagt en butyl fuge. Se Figur 3.2.

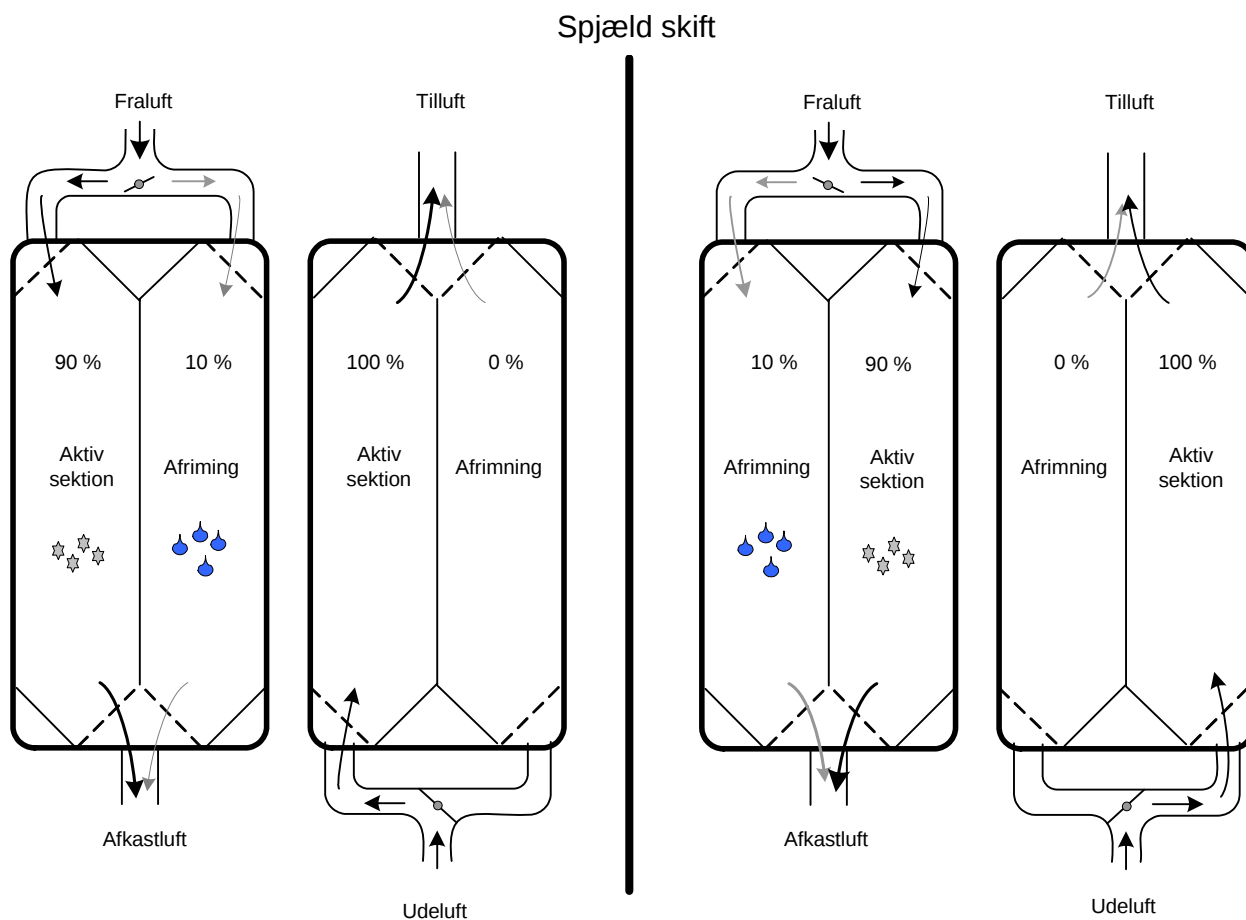


Figur 3.2 Tværsnit af kasseveksleren med angivelse af de to sektioner. Tegningen er ikke i mål.

Figur 3.3 viser en principskitse af funktionen af de to sektioner. I kolde perioder er det hensigten at den ene sektion er aktiv mht. varmeveksling, mens den anden har mulighed for afrimning. Når der er risiko for tilrimning styres luftstrømmene til de to sektioner af to motorspjæld og en simpel timer. Timeren skifter med et passende tidsinterval om på hvilken af de to sektioner, der skal være aktiv mht. varmeveksling og afrimning.

Når afrimning er nødvendig ledes ca. 10 % af afkastluften fortsat gennem den inaktive sektion for på den måde at tø rim fra vekslerens overflader.

På tilluftsiden indstilles spjældet til at 100 % af volumenstrømmen altid føres til den aktive varmevekslende sektion og 0 % til afrimningssektionen.



Figur 3.3 Skitse af modstrømsvarmeveksler bestående af to parallel sektioner der benyttes skiftevis. Afrimningen foretages ved at føre ca. 10 % af afkastluften gennem den passive sektion.

Som det ses af Figur 3.3 føres luftstrømmene vertikalt med fraluften strømmende lodret nedad, hvorved der opnås en optimal bortledning af kondensvandet. I bunden har veksleren en åbning på 50 x 10 cm, hvor afkastluften og kondensvandet kan dryppe uhindret ud af veksleren.

Når der ikke haves behov for afrimning stilles de to spjæld i midterposition således af luftstrømmene fordeles ligeligt til begge sektioner. Derved udnyttes det varmeoverførende areal optimalt.

Ønskes en veksler med en højere eller lavere effektivitet kan antallet af ribbeplader hhv. øges eller reduceres. Figur 3.4 viser et billede af den færdige kasseveksler.

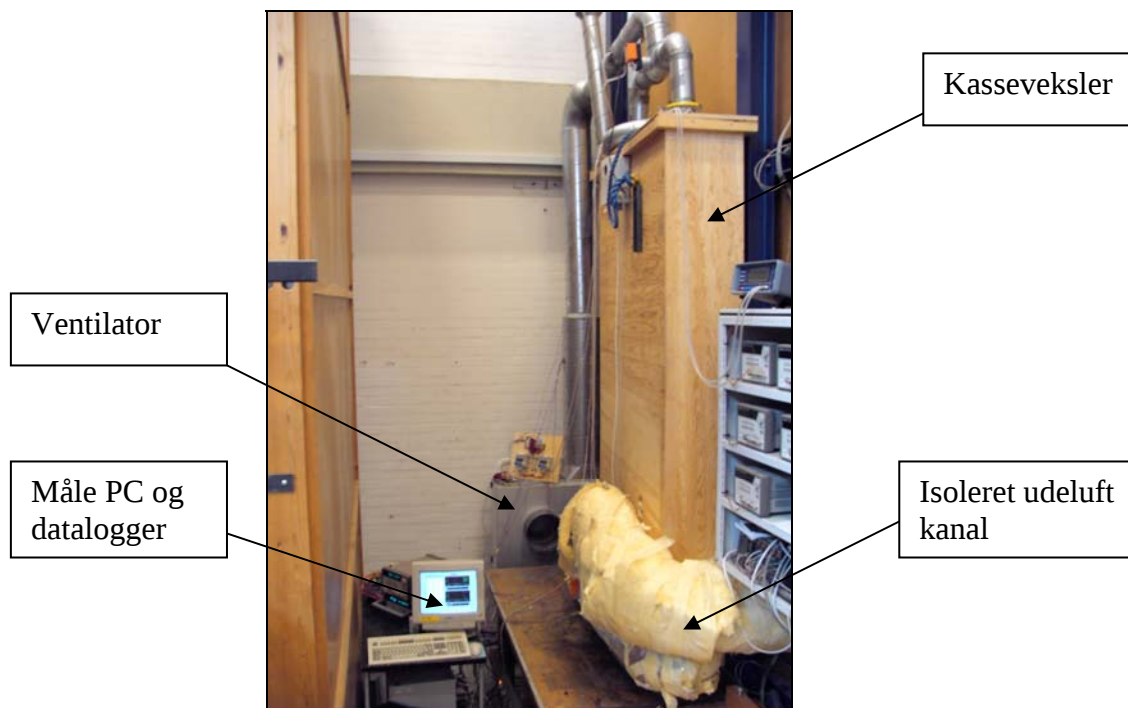


Figur 3.4 Billede af veksleren

3.2 Forsøgsopstillingen

Formålet med forsøgsopstillingen er at teste om den ny udviklede kasseveksler er i stand til at afri-me sig selv kontinuerligt ved lave udetemperaturer.

I BYG·DTU's forsøgshal er opbygget en forsøgsopstilling ved siden af et eksisterende kølerum. Kølerummet kan levere tilluft med temperatur ned til ca. -8°C afhængigt af den ønskede volumenstrøm. Fraluftens tages fra forsøgshallen. Volumenstrømmen af til- og fraluft leveres af to boksventilatorer med manuel trinløs regulering. På Figur 3.5 ses et billede af forsøgsopstillingen.



Figur 3.5 Billede af forsøgsopstilling på Danmarks Tekniske Universitet.

Den kolde udeluft tilføres nederst gennem et rør der af hensyn til udvendig kondens er isoleret. I praksis vil dette skulle laves med et simpelt luftindtag direkte til det fri.

Volumenstrømmen af til- og fraluft måles med $\text{Ø}125$ målekryds forbundet til to differensstryktrans-mittere. Flowfordelingen mellem de sektioner måles med et mircomanometer. Herudover måles relativ fugtighed i fraluftens før veksleren. Figur 3.6 viser rørforbindelsen øverst på veksleren.

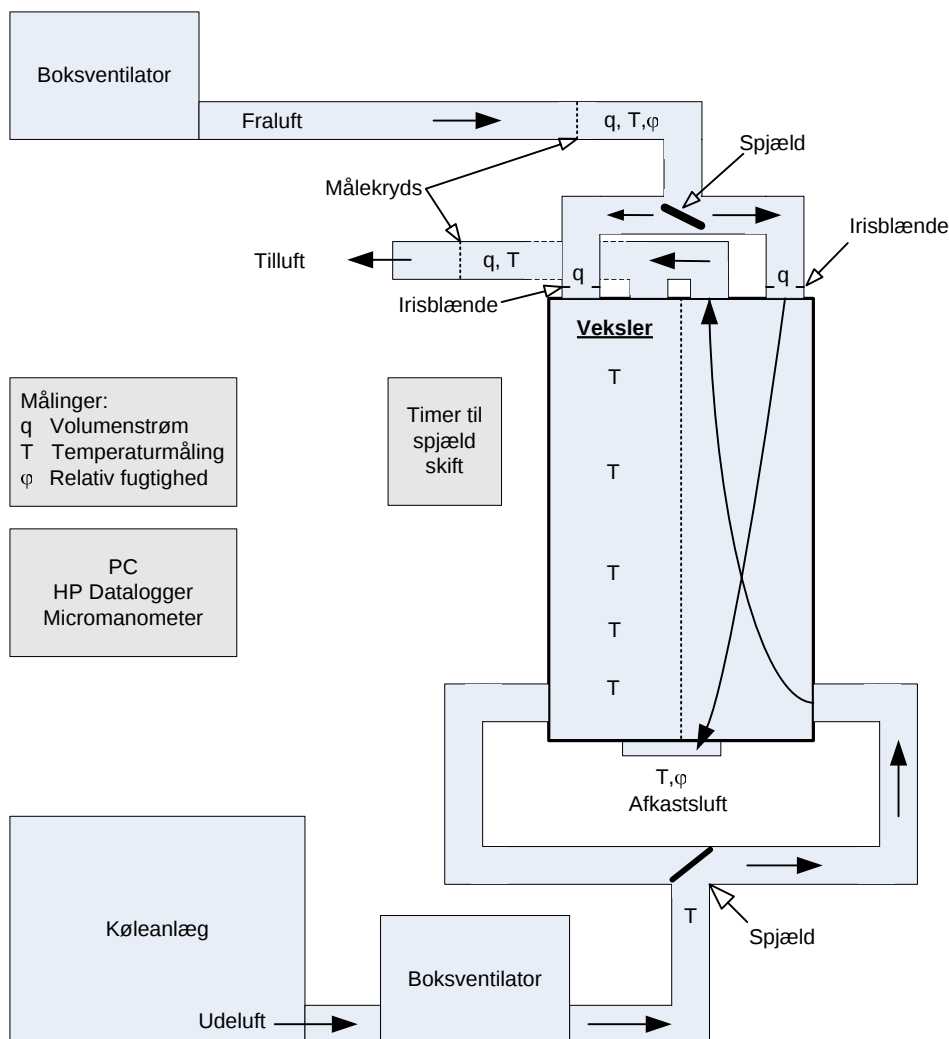


Figur 3.6 Nærbillede af timeren og rørforbindelserne øverst på varmeveksleren

I hver kanal umiddelbart før og efter varmeveksleren måles temperatur ved hjælp af kobber/konstantan termoelementer (type TT). Temperaturerne måles i et symmetrisk tværsnit. Antallet af målepunkter i hver kanal ses nedenfor:

Fraluft	2 målepunkter	Afkastluft	3 målepunkter
Udeluft	2 målepunkter	Tilluft	4 målepunkter

I den efterfølgende præsentation af måleresultaterne er benyttet en middeltemperatur af alle målepunkterne for hver luftstrøm. Desuden måles temperaturprofilen på fraluftsiden gennem den ene sektion ved brug af 5 målepunkter. Figur 3.7 viser en skitse af forsøgsopstillingen med angivelse af de forskellige målepunkter.



Figur 3.7 Skitse af forsøgsopstilling til kondens- og tilsningsforsøg

Til dataopsamlingen benyttes en stationær PC med programmet Benchlink installeret. Til PC'en er en HP datalogger (Agilent 34970A) forbundet, hvor alle data logges hvert minut. I bilag 1 ses en oversigt over det benyttede måleudstyr.

3.3 Forsøgsresultater

I det følgende præsenteres resultaterne for en række forsøg foretaget med den ovenfor beskrevne forsøgsopstilling. Forsøgene er foretaget ved forskellige temperaturniveauer af udeluften. Grundet usikkerheden ved måling af meget små volumenstrømme er forsøgene foretaget ved volumenstrømme mellem 25 og 50 l/s (ca. 90 til 180 m³/h). Dette svarer dog til det nødvendige niveau i typiske enfamiliehuse der opfylder bygningsreglementet krav til et luftskifte på 0.5 h⁻¹.

Temperaturvirkningsgraden er i det følgende defineret som:

$$\eta = \frac{T_{\text{tilluft}} - T_{\text{udeluft}}}{T_{\text{fraluft}} - T_{\text{udeluft}}}$$

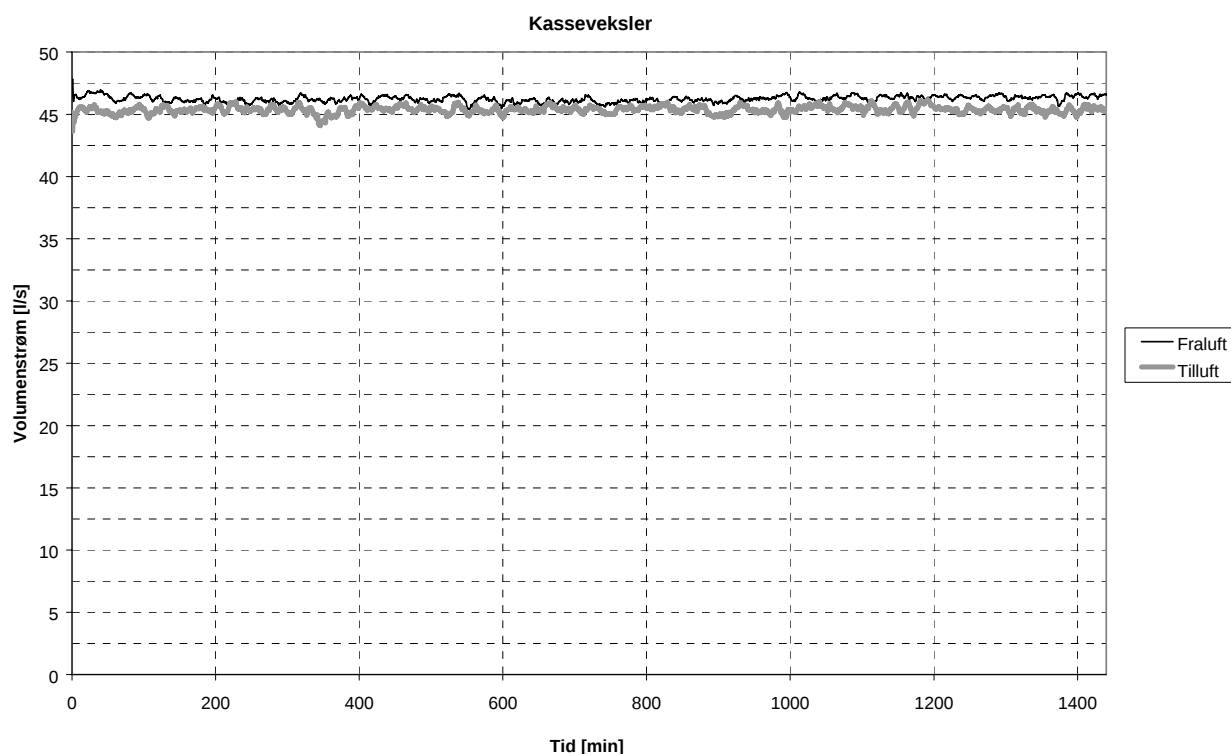
Ligning 1

Hvor T_{tilluft} er temperaturen af indblæsningsluften, T_{udeluft} er temperaturen af den simulerede udeluft og T_{fraluft} er temperaturen af luften i hallen.

3.3.1 Måling under normale driftsforhold uden tilrimning

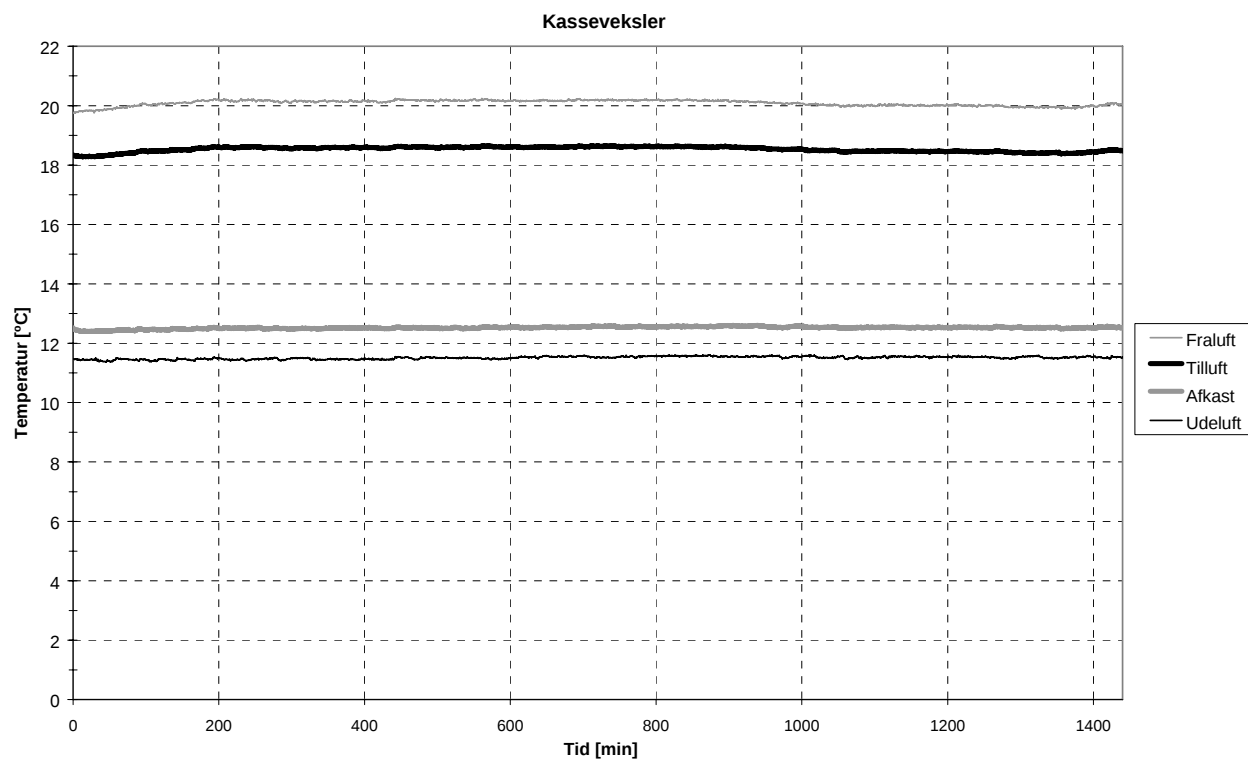
Veksleren kan som tidligere beskrevet benytte begge sektioner til varmeveksling, hvis der ikke er risiko for tilrimning. Dette gøres ved at stille de to spjæld i midterstilling, hvorved luftstrømmene fordeles ligeligt mellem de to sektioner og derved fordobler det varmevekslende areal.

På Figur 3.8 ses den målte temperaturvirkningsgrad for denne situation. Forsøget er udført som en tør måling dvs. uden kondens på fraluftsiden. Temperaturen af udeluften under forsøget lå konstant på ca. 11,5 °C og fraluftens relative fugtighed lå på ca. 55 %.

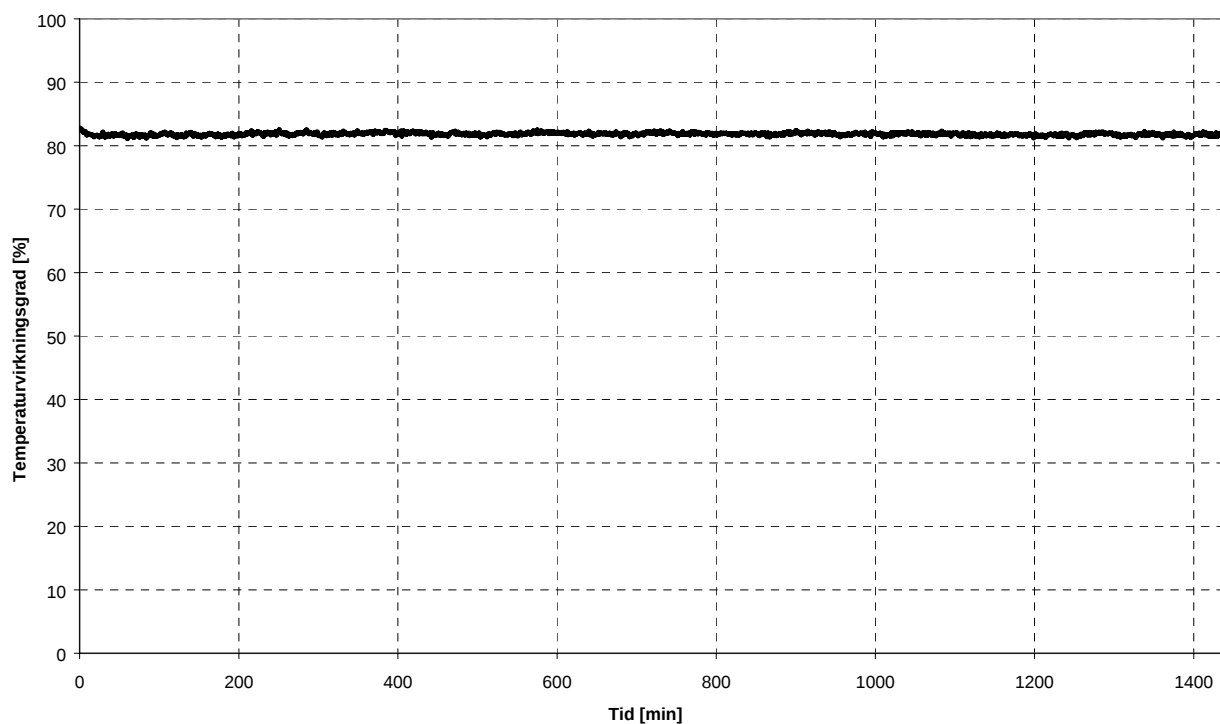


Figur 3.8 Målingen af den totale volumenstrøm gennem begge vekslerens sektioner på både fraluft- og tilluftsiden.

Figur 3.9 og Figur 3.10 hhv. de målte temperaturer før og efter veksleren og temperaturvirkningsgraden. Det ses at denne ligger på ca. 82 % målt ved en volumenstrøm på ca. 46-47 l/s.



Figur 3.9 De målte temperaturer før og efter veksleren. Det ses at udeluften har ligget på ca. 11,5°C

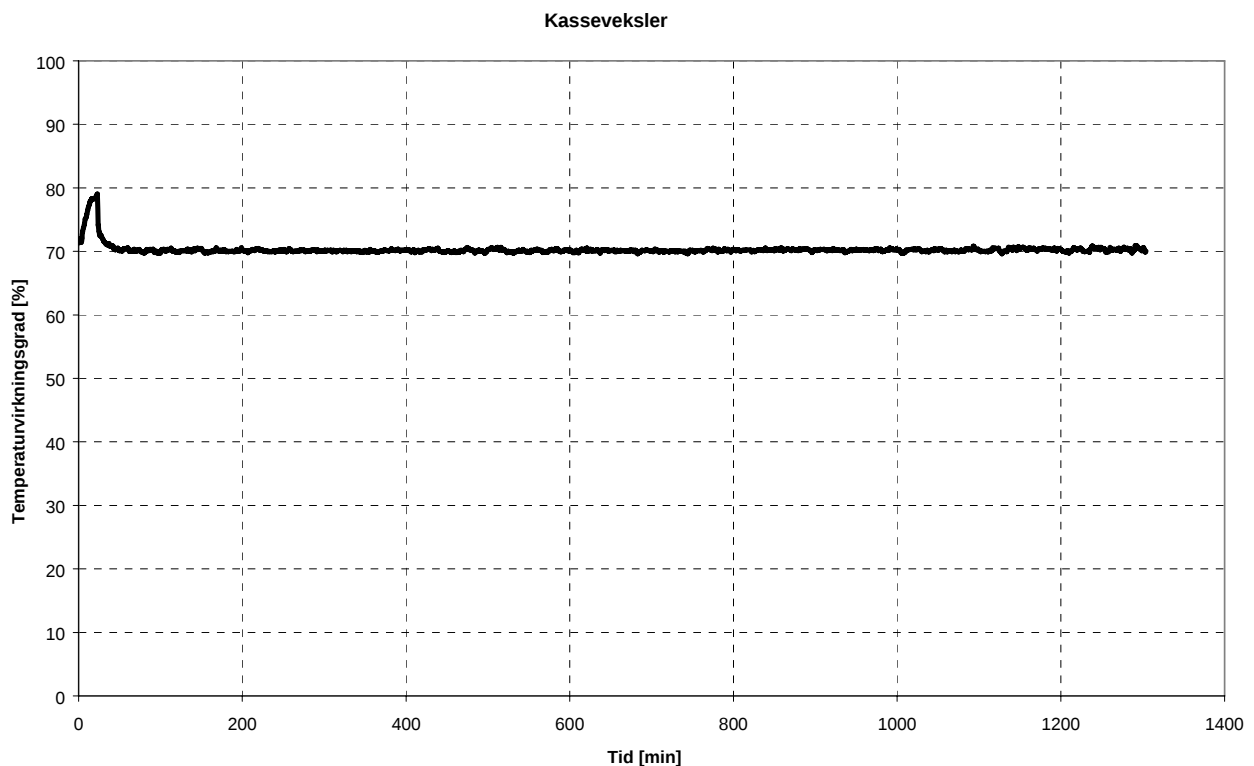


Figur 3.10 Temperaturvirkningsgraden for hele veksleren under forsøget

3.3.2 Tør måling af temperaturvirkningsgrad for én sektion

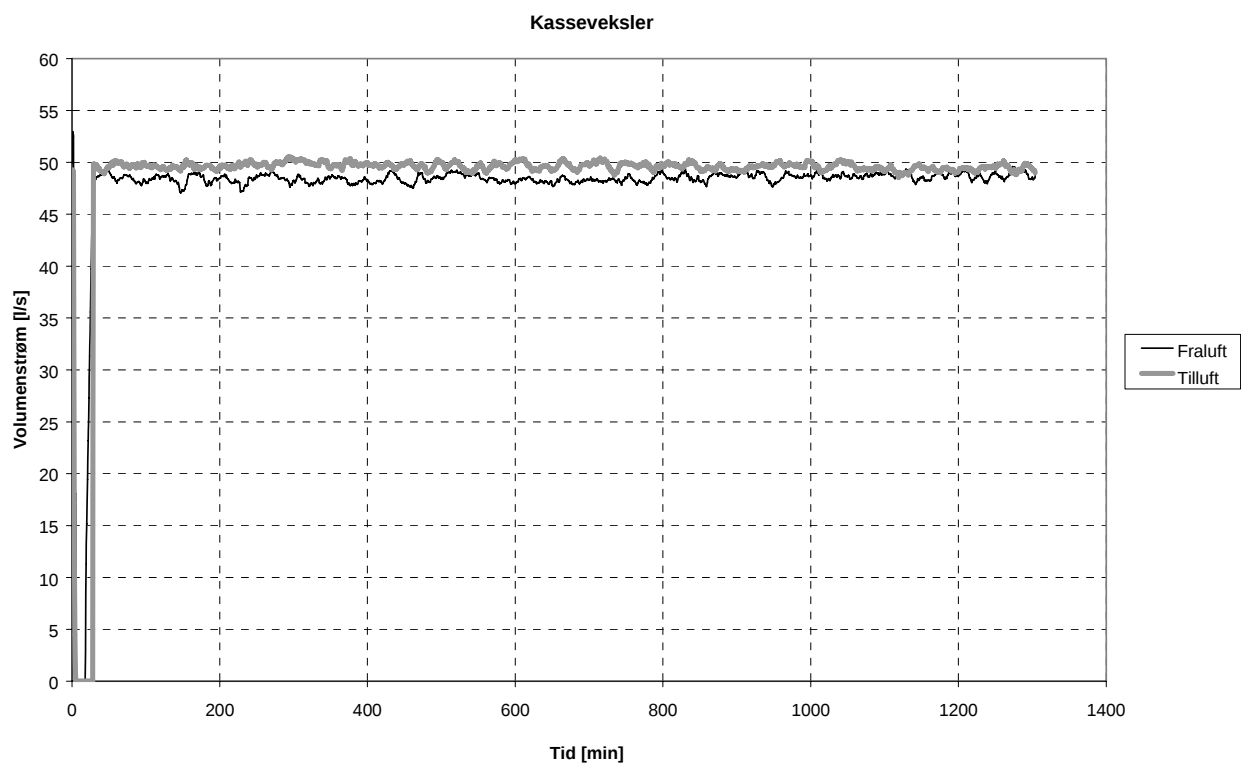
I kolde perioder vil det være nødvendigt at benytte afrimningsfunktionen, hvilket vil sige at kun en sektion af gangen benyttes. Da det varmevekslende areal derved halveres reduceres også temperaturvirkningsgraden, hvilket er undersøgt i det følgende. En måling af varmevekslerens tørre temperaturvirkningsgrad er foretaget ved samme volumenstrøm som ovenfor (ca. 48 l/s) og ved en temperatur af udeluften på ca. 12°C. Herved skulle hverken kondensering eller tilrimning forekomme internt i veksleren.

Som det ses på Figur 3.11 reduceres temperaturvirkningsgraden fra de 82 % til ca. 70 % når kun den ene sektion er aktiv målt ved samme temperaturniveauer og volumenstrømme.

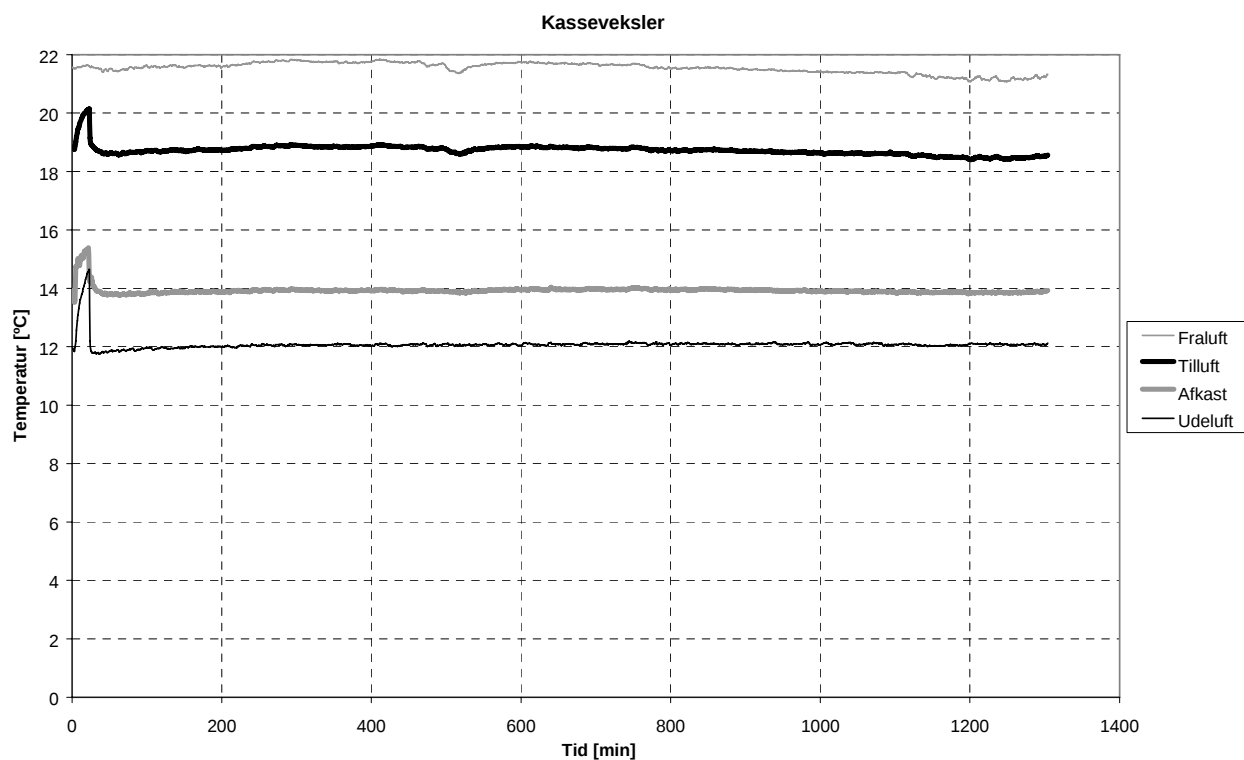


Figur 3.11 Den målte temperaturvirkningsgrad for halvdelen af veksleren foretaget ved en temperatur af udeluften over dugpunktstemperaturen. Der er således ikke kondensation internt i veksleren

På Figur 3.12 og Figur 3.13 ses måledataene for hhv. volumenstrømmene og til- og afgangstemperaturerne til varmeveksleren målt under forsøget.



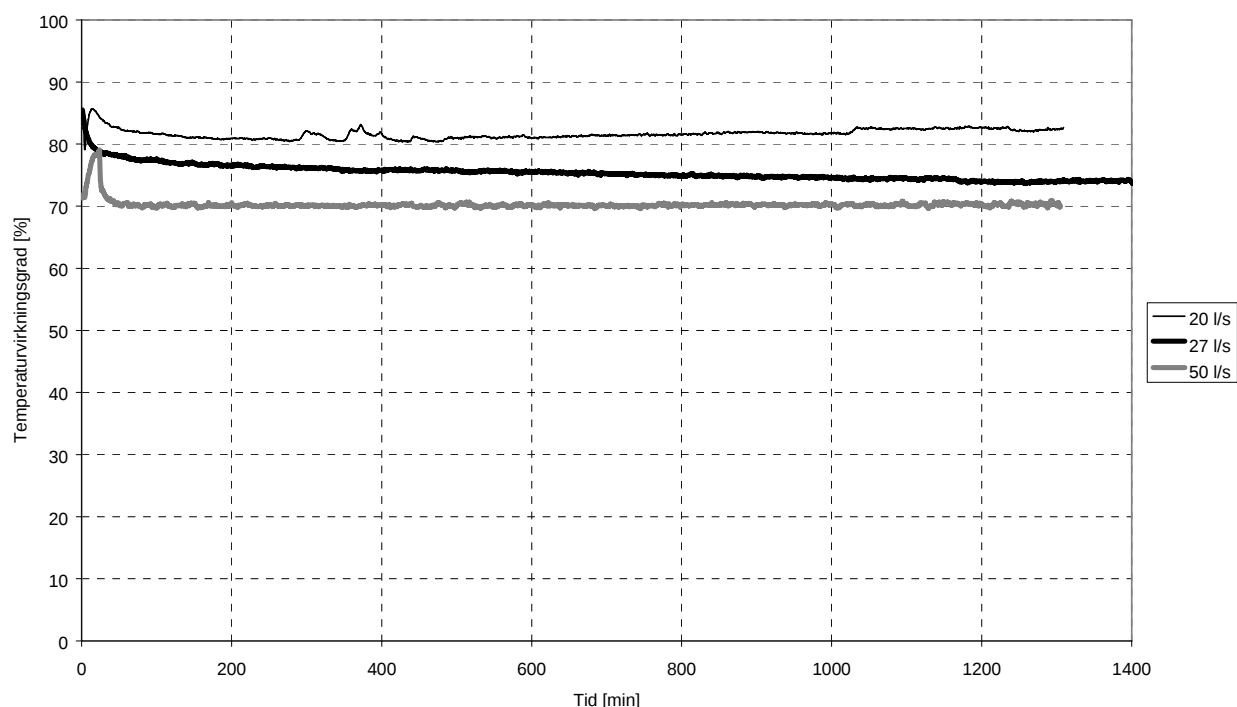
Figur 3.12 De målte volumenstrømme på fraluft og tilluftsiden



Figur 3.13 De målte temperaturer før og efter veksleren. Det ses at udeluften har ligget på ca. 12°C

På Figur 3.14 er foretaget to tilsvarende målinger ved samme temperaturniveau, men med en volumenstrøm på hhv. 20 l/s og 27 l/s på både fra- og tilluftsiden, idet det evt. kan forsvares at reducere

luftskiftet i meget kolde perioder, idet fugtindholdet i udeluften er ekstremt lavt og derved øger udtørringen boligen.

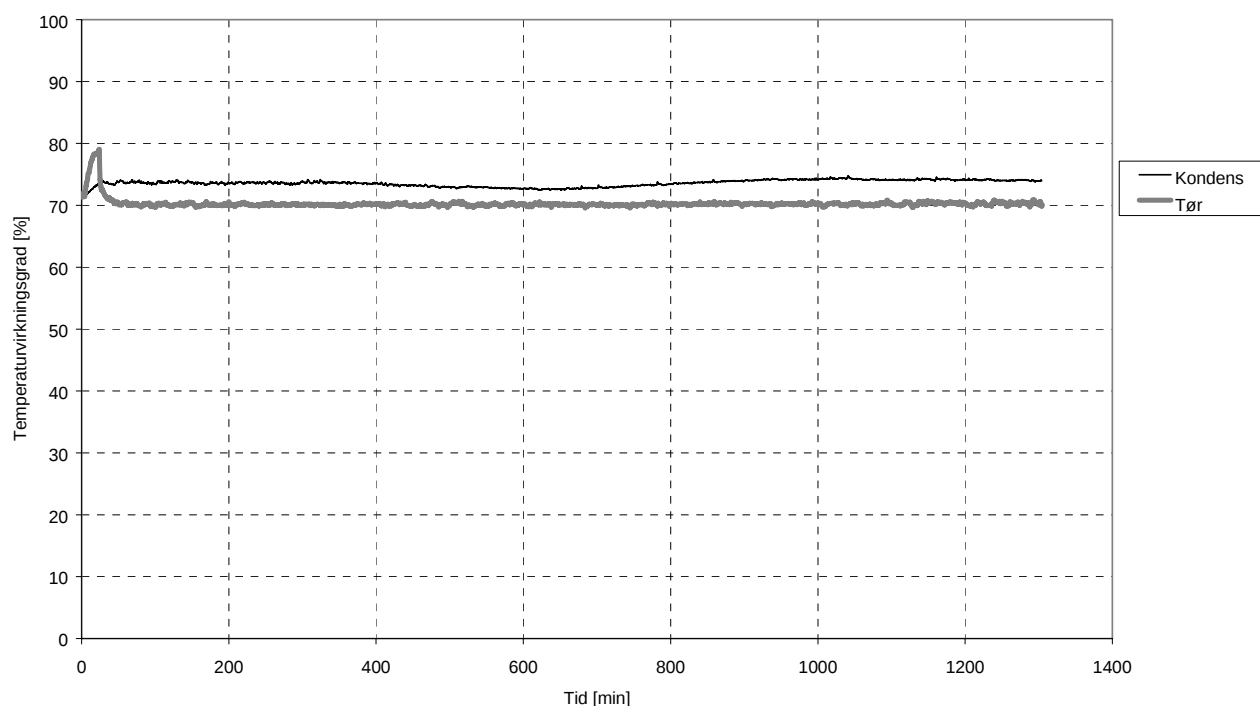


Figur 3.14 Temperaturvirkningsgraden for halvdelen af veksleren ved forskellige niveauer for volumenstrømmen. Volumenstrømmen har under forsøgene været indstillet til samme niveau på fraluft og tilluftsiden.

Som det ses på Figur 3.14 stiger temperaturvirkningsgraden som forventet med faldende volumenstrøm og omvendt falder den ved stigende volumenstrømme.

3.3.3 Kondenseringsforsøg

Den næste serie af forsøg blev foretaget ved temperaturer af udeluften under luftens dugpunktstemperatur men over frysepunktet. Under forsøget havde indeluften en temperatur på ca. 21,5 °C og en relativ fugtighed på 56 %, hvilket resulterer i en dugpunktstemperatur på ca. 12 - 13 °C. Udeluften havde en temperatur på ca. 3,7 °C, hvilket således medfører kondens, men dog ikke rim på afkastsiden. På Figur 3.15 ses varmevekslerens målte temperaturvirkningsgrad under kondenseringsforsøget. Målingen er desuden sammenlignet med den tilsvarende tørre måling uden kondens.

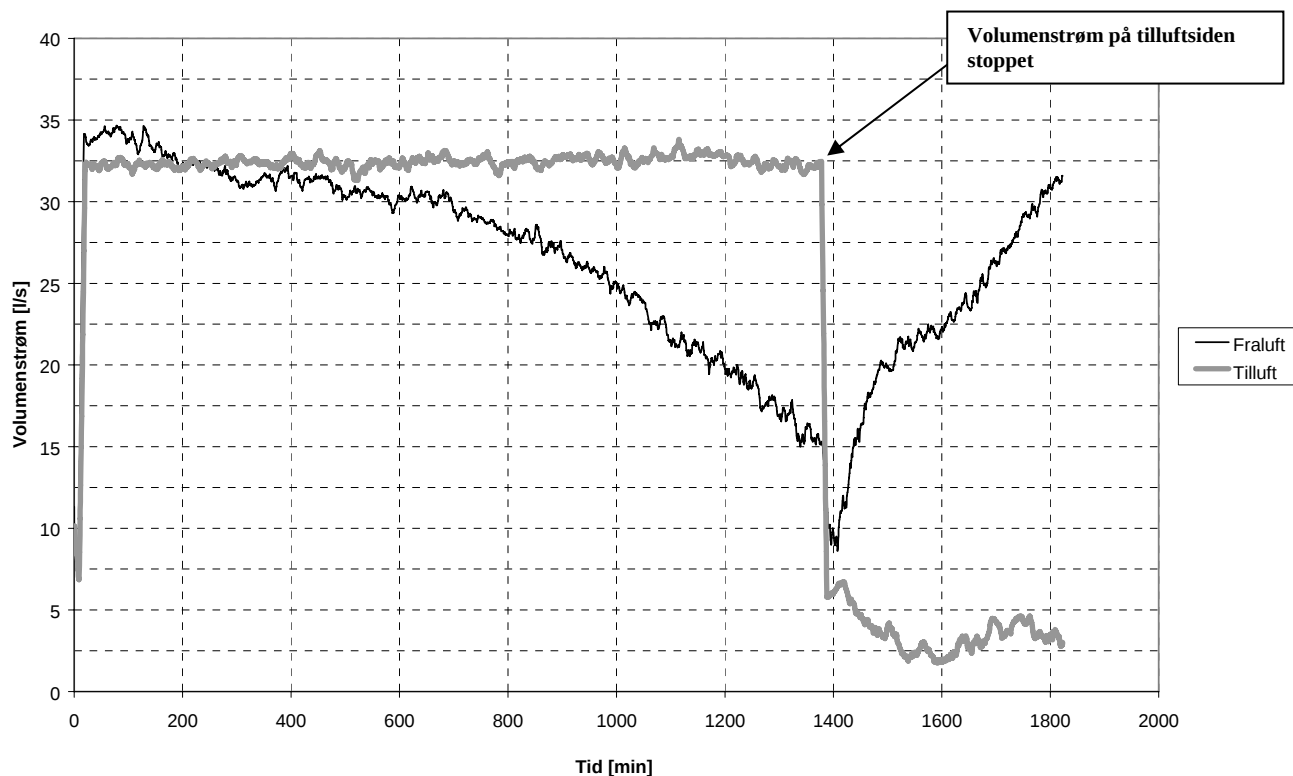


Figur 3.15 Måling af temperaturvirkningsgraden foretaget ved en volumenstrøm på 50 l/s for en tør måling og for en måling med kondensation på fraluftsiden.

Som det ses på Figur 3.15 øges temperaturvirkningsgraden med 3 til 4 % når fraluften kondenserer i varmeveksleren sammenlignet med den tilsvarende tørre måling. Dette skyldes kondensvarmen fra fugtige indeluft, der frigives på vekslerens overflader under afkølingen.

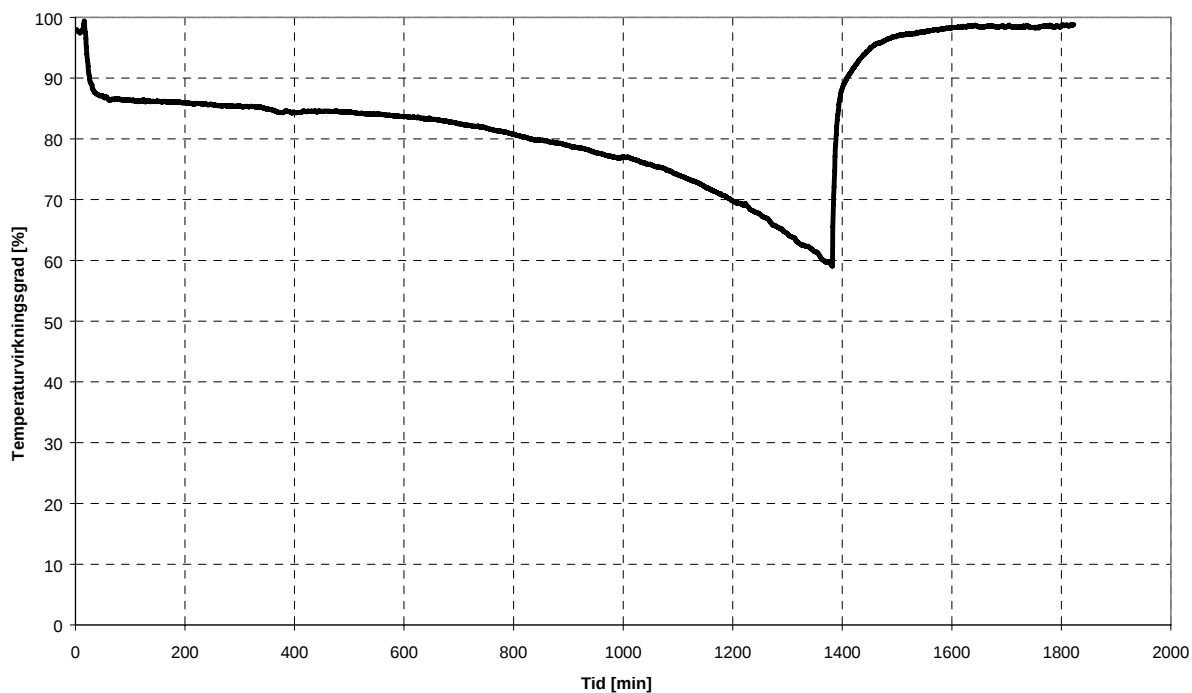
3.3.4 Tilisning og optøning af veksler

I de efterfølgende forsøgsresultater er temperaturen af udeluften sænket til under frysepunktet. For at eftervise at også denne veksleren tilrimes blev afrimningsfunktionen ikke aktiveret under det første forsøg. Herved kunne tilrimningsproblemet bekræftes også for denne prototype af en ventilationsvarmeveksler.



Figur 3.16 Målt volumenstrøm gennem veksleren på fra- og tilluftsiden. Volumenstrømmen på tilluftsiden stoppes efter ca. 1480 minutter, idet volumenstrømmen på afkastssiden er fladet med mere 50 % grundet rimdannelsen på varmevekslerens overflader. Efter ca. 200 minutter er volumenstrømmen steget til start niveauet.

På Figur 3.17 ses hvorledes temperaturvirkningsgraden falder jævnt under forsøget grundet tilisning i veksleren. Efter de 1480 minutter giver temperaturvirkningsgraden ikke mening da volumenstrømmen på tilluftsiden er stoppet. På Figur 3.17 ses den tilsvarende temperaturvirkningsgrad.



Figur 3.17 Måling af temperaturvirkningsgraden under tilrimningsforsøget.

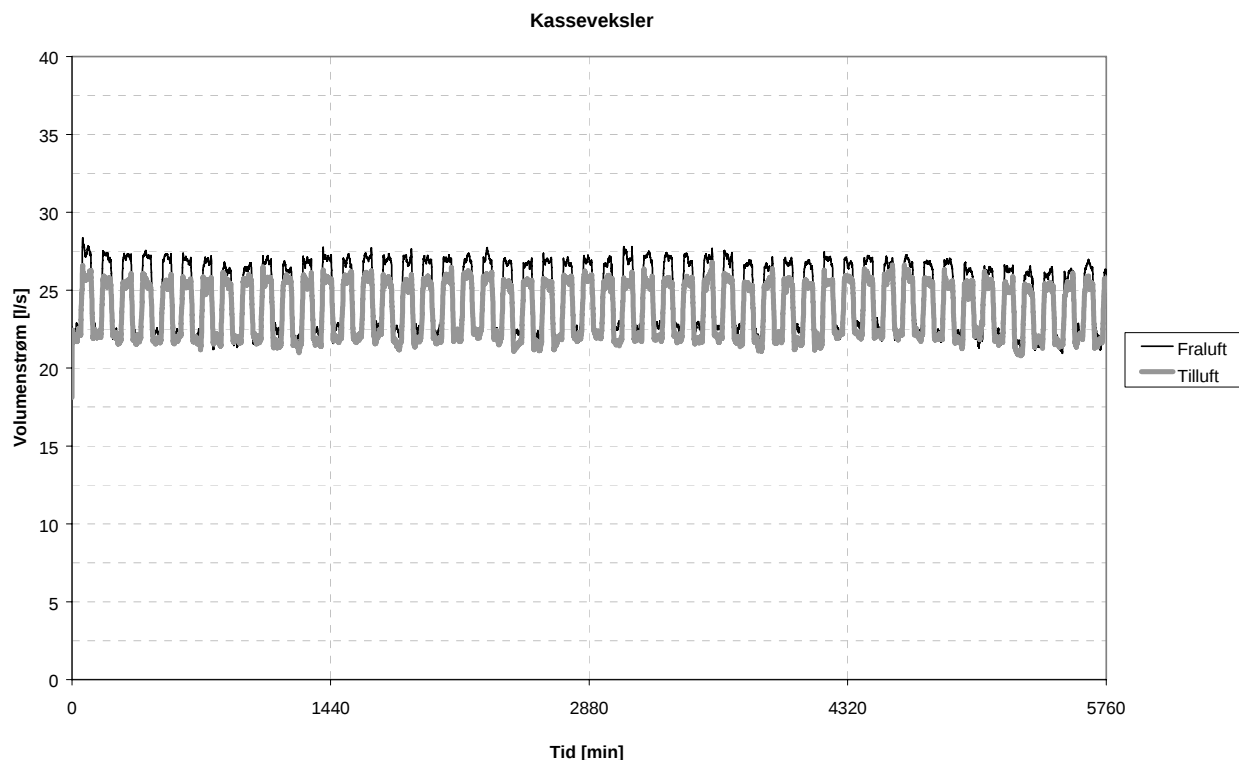
3.3.5 Afrinningsforsøg

I det følgende præsenteres de forsøg der blev foretaget med henblik på at teste varmevekslerens afrinningsfunktion.

Forsøget blev udført ved en temperatur af udeluften på ca. -6°C og en volumenstrøm på ca. 25 l/s. På fraluftsiden blev spjældet indreguleret til at give ca. 90 % af volumenstrømmen til den aktive sektion og 10 % til den passive sektion.

Forsøget viste desuden at tryktabet gennem de to sektioner ikke var helt ens, hvilket påvirker størrelsen af volumenstrømmene. Derfor ses at volumenstrømmen ændre sig noget ved hvert spjældskift. Da volumenstrømmen af begge luftstrømme (fra- og tilluft) ændres ens er dette dog af mindre betydning for forsøget.

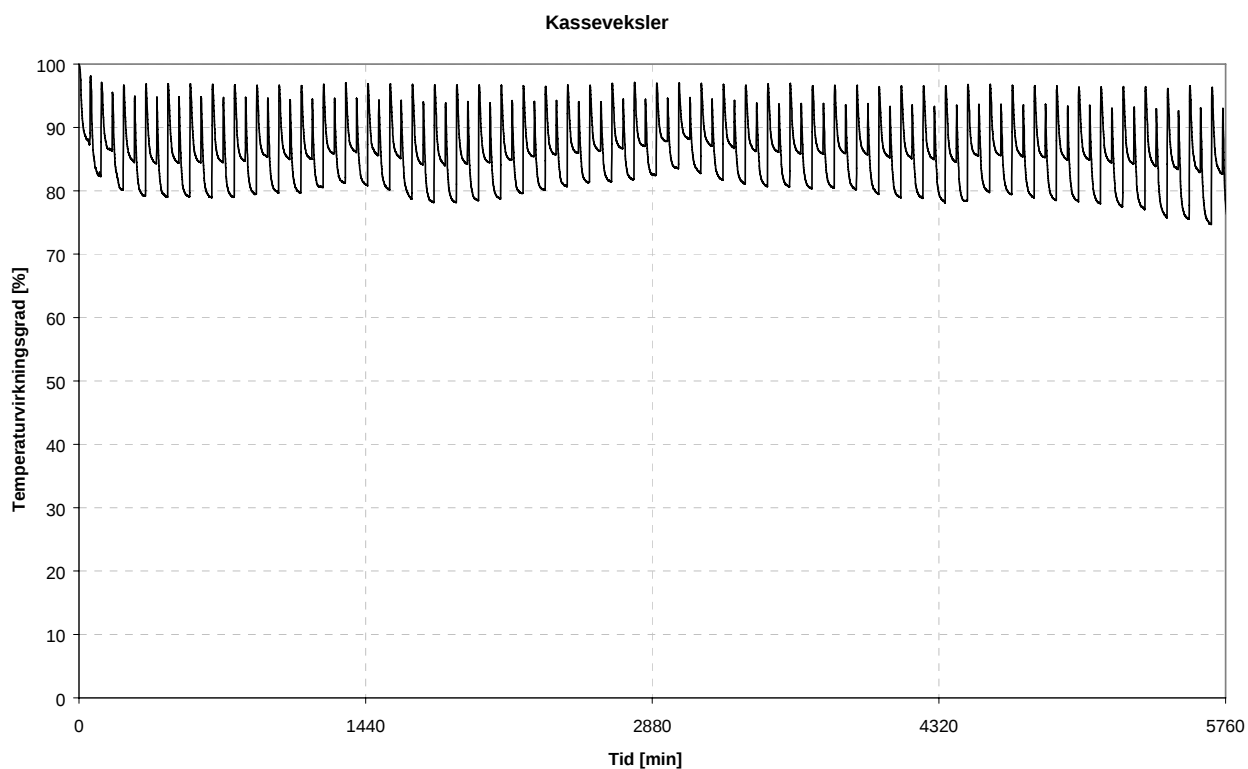
Timeren der aktiverer de to spjæld blev indstillet til en skiftetid/sektionsskift på ca. 1 time. Da forsøget blev udført ved en forholdsvis lille volumenstrøm (25 l/s), medfører dette at det tager relativt lang tid før en evt. tilrimning af vekslerens overflade påvirker volumenstrømmen nok til at det kan registreres. Forsøget blev derfor udført uden afbrydelser over en måleperiode på 4 døgn. På Figur 3.18 ses den målte totale volumenstrøm på fraluft og tilluftsiden.



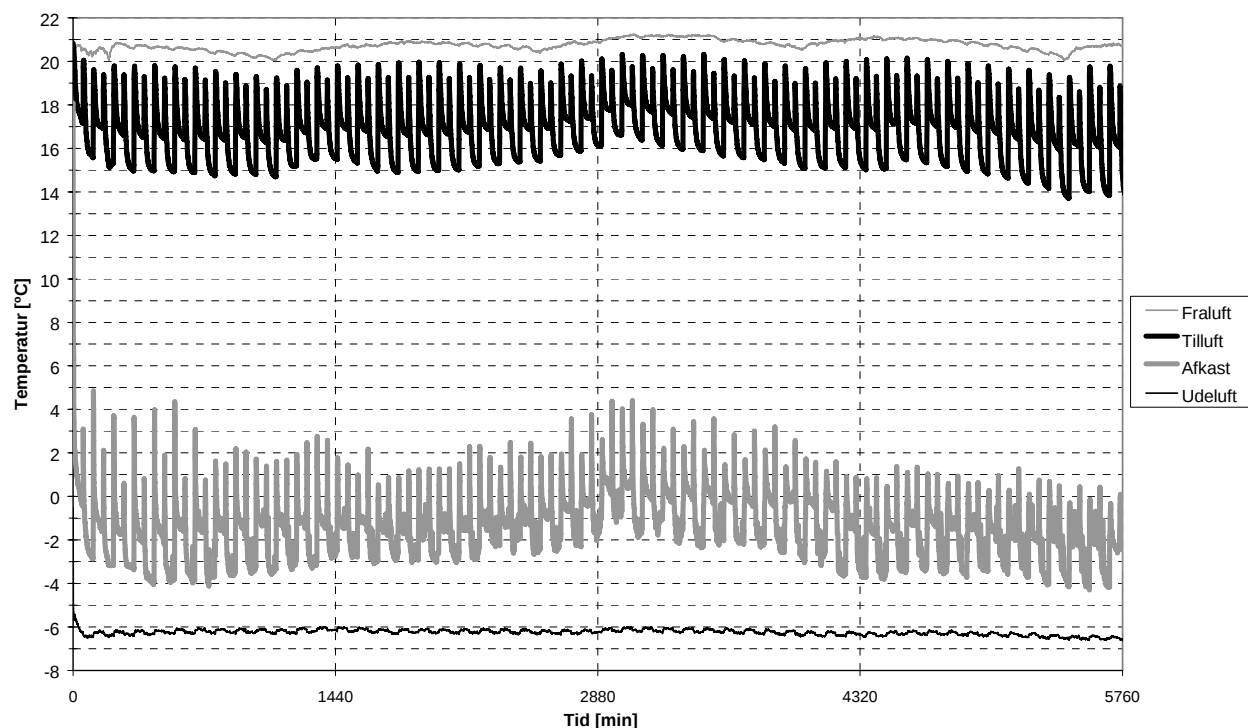
Figur 3.18 De totale volumenstrømme under afrinningsforsøget. Variationen skyldes forskelle i tryktab over de to sektioner.

Som det ses er volumenstrømmenes niveau konstant gennem hele perioden, hvilket viser at afrinningsfunktionen virker.

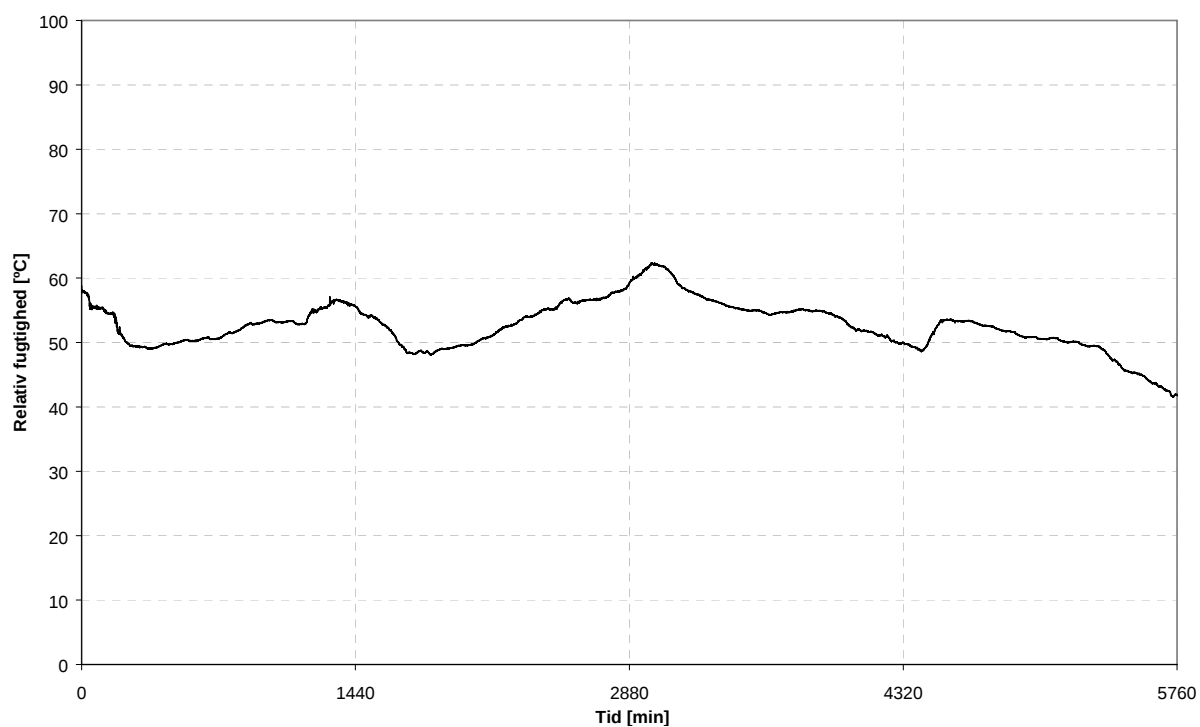
På Figur 3.19 ses temperaturvirkningsgraden under afrimningsforsøget. Det ses at denne er meget ustabil grundet de mange spjæld skift. Som middel var temperaturvirkningsgraden under forsøget ca. 85 %.



Figur 3.19 Temperaturvirkningsgraden under afrimningsforsøget.



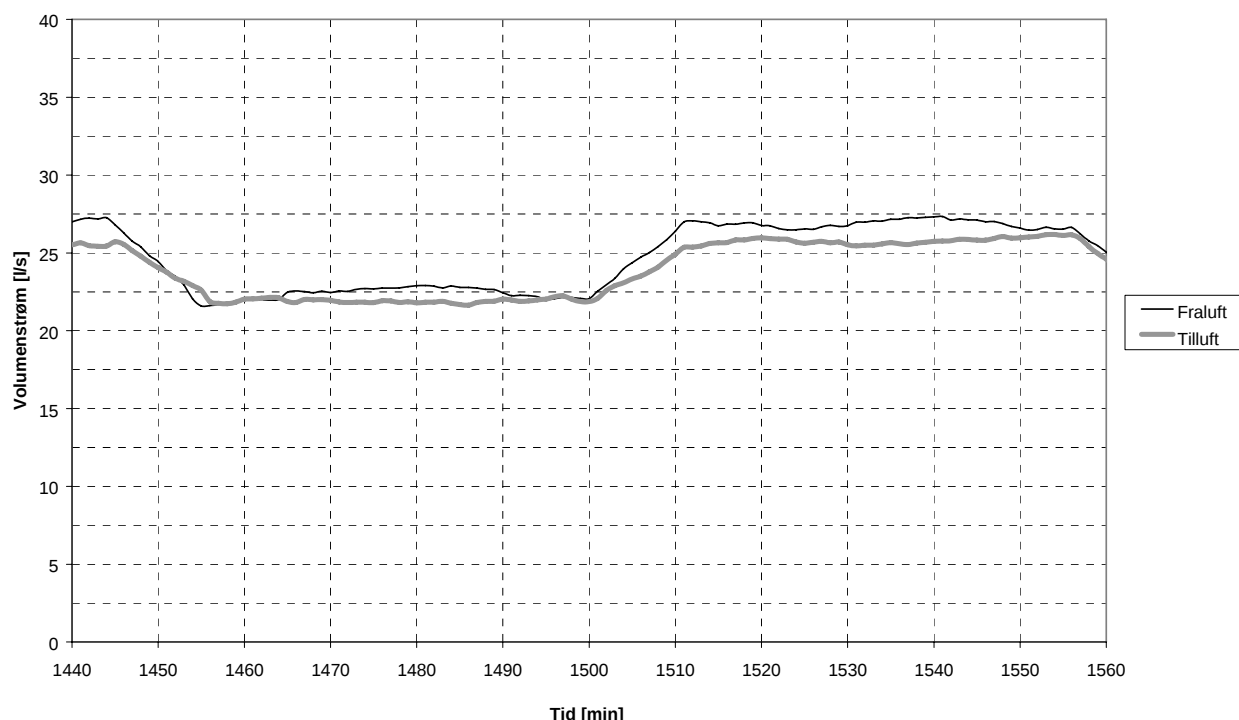
Figur 3.20 Temperaturerne målt i de fire til- og afgangskanaler.



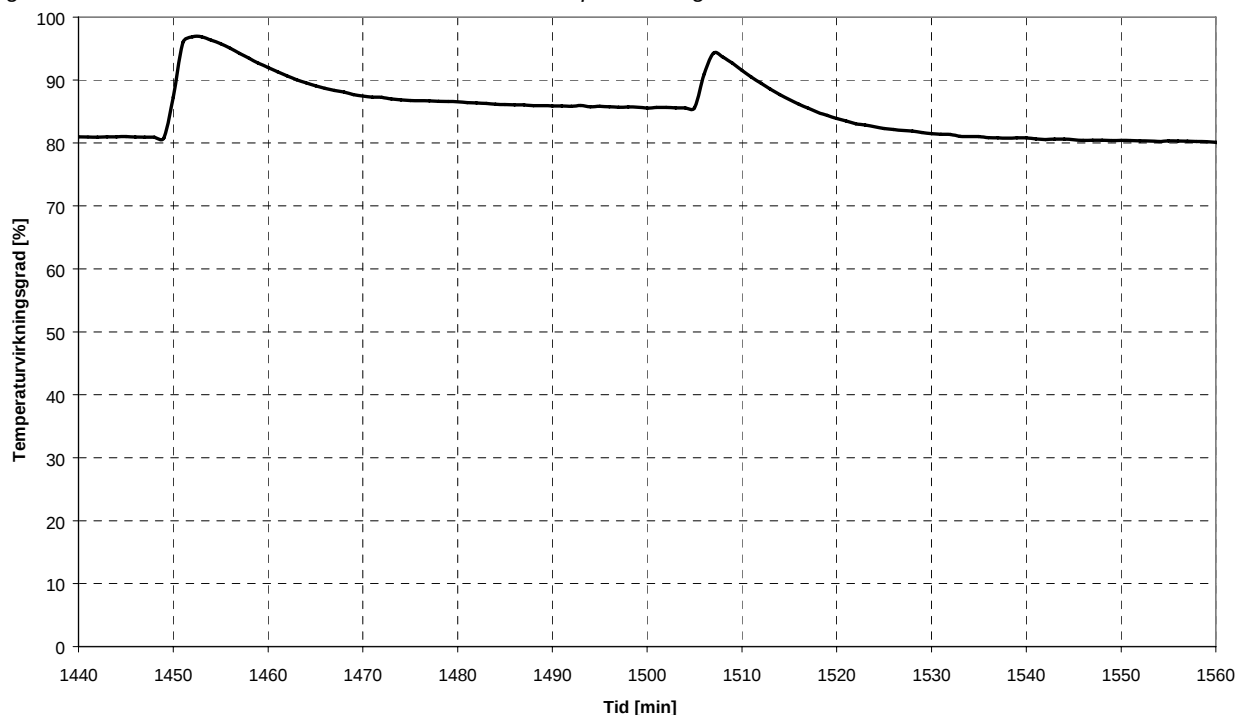
Figur 3.21 Den målte relative fugtighed i fraluften umiddelbart før veksleren.

For at se nærmere på temperaturdynamikken internt i varmeveksleren er der i det følgende vist et zoom af måledataene over en periode på 2 timer svarende til to spjældskift af ca. 1 times varighed.

Figur 3.22 viser de to volumenstrømme, der ses at variere mellem ca. 22 og 26 liter l/s, hvilket som tidligere nævnt kan forklares ved, at der er en mindre forskel i tryktabet over de to sektioner.



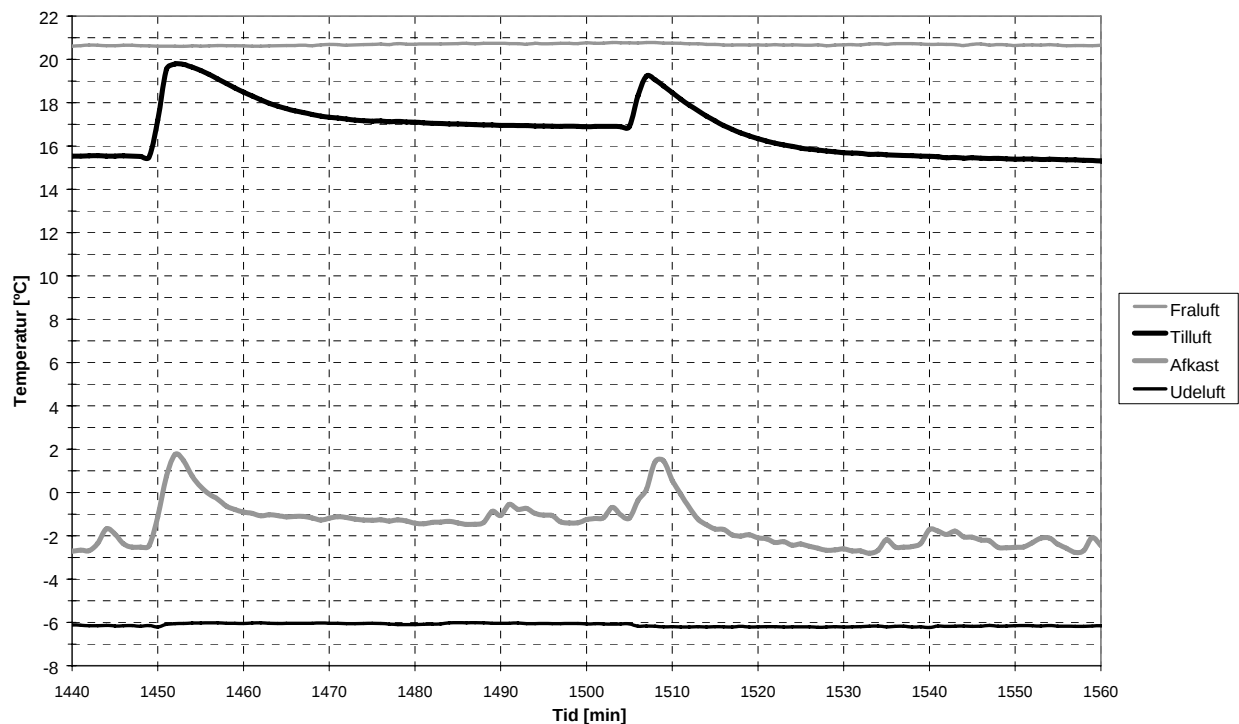
Figur 3.22 Zoom af måledataene for volumenstrømmen på tilluft- og fraluftsiden i løbet af to timer



Figur 3.23 Zoom af temperaturvirkningsgraden gennem to timer

De to stigninger i temperaturvirkningsgraden umiddelbart efter sektionsskift kan forklares ved at noget af den energi der benyttes under afrimningen af den inaktive sektion, lagres som termisk energi i ribbepladen trods det at dennes varmekapacitet er meget lille sammenlignet med andre ma-

terialers. Umiddelbart efter et sektionsskift vil den energi, der er oplagret som termisk energi i ribbepladen bevirke at tillufttemperaturen stiger et par grader, hvilket også ses på Figur 3.24. Jf. ligning 1 vil temperaturvirkningsgraden hæves når tillufttemperaturen stiger.



Figur 3.24 Til- og afgangstemperaturerne i løbet af to timer

3.3.6 Kondensudfældningen og tilrimningen

I det følgende er forsøgt beskrevet de visuelle observationer, der blev foretaget af kondensudfældningen under forsøget. Som tidligere nævnt har veksleren en åbning på 50 x 10 cm i bunden, hvor afkastluften og kondensvandet kan dryppe uhindret ud af veksleren.

Under forsøgene blev det observeret at kondensvandet dryppede fra begge sektioner samtidigt. Umiddelbart efter et sektionsskift øgedes kondensmængden dog markant fra den sektion der var gået fra inaktiv til aktiv. Det kan forklares ved at rimen i den inaktive sektion i løbet af afrimningsperioden ændre sig til dråber der hænger fast på overfladen. Noget drypper ud og noget bliver hængende på siderne. Ved sektionsskift vil den forøgede volumenstrøm hjælpe til med at presse vandet ned mod bunden af veksleren. At der samtidigt drypper vand fra den aktive varmevekslende sektion tyder på at noget af luftens fugtindhold udkondenseres på vekslerens overflader og direkte drypper forbi frostzonen.

3.4 Delkonklusion på kondens og tilisningsforsøg med nyudviklet veksler

Den i projektet udviklede veksler til kolde klimaer blev testet under kontrollerede laboratorieforhold. Her blev vekslerens temperaturvirkningsgrad målt under temperaturforhold svarende til en

typisk forårs/efterårsdrift uden risiko for frost. Ved en volumenstrøm på ca. 45 l/s blev temperaturvirkningsgraden målt til ca. 82 % (tør måling). I perioder med frost benyttes kun den ene vekslersektion, hvilket reducerede temperaturvirkningsgraden til ca. 70 % (tør måling) og 74 % (med kondensudfældning).

Afrimningsfunktionen blev testet gennem et længerevarende forsøg over 4 døgn. Her var volumenstrømmen på fraluftsiden konstant hvilket således efterviste at afrimningsfunktionen virker. Forsøget blev udført ved en temperatur af udeluften på ca. -6°C og en hal temperatur på 20-21°C.

Forsøgene med den nyudviklede veksler er desuden beskrevet i artiklen i bilag 5.

4 Beregningsmodel for varmevekslere

For at kunne undersøge forholdene omkring kondens og tilisning i varmevekslere allerede i designfasen, er det nødvendigt at have beregningsværktøjer, der kan analysere hvordan kondens og tilisning påvirker varme- og luftstrømme afhængig af varmevekslerens design og styringsstrategier. Derfor er der i projektet udviklet en beregningsmodel til simulering af varmevekslere. Beregningsmodellen er indtil videre begrænset til at vurdere indflydelsen af kondens på varmestrømmene i veksleren og kan give et indtryk af hvor i veksleren kondens vil opstå og under hvilke betingelser, der er risiko for tilisning.

Beregningsmodellen er udviklet i trin, svarende til at der først er udviklet en simpel stationær model som efterfølgende er videreudviklet til en dynamisk formulering. Modelleringen af varmevekslere og herunder specielt forholdene som gør sig gældende i forbindelse med faseskift (kondens og isdannelse) er meget kompliceret, og derfor er der indledningsvist arbejdet med en række simplificeringer som siden hen bearbejdes, således at der i sidste ende haves en detaljeret dynamisk model. Modellen vil således efter nærværende projekts afslutning fortsat blive videreudviklet.

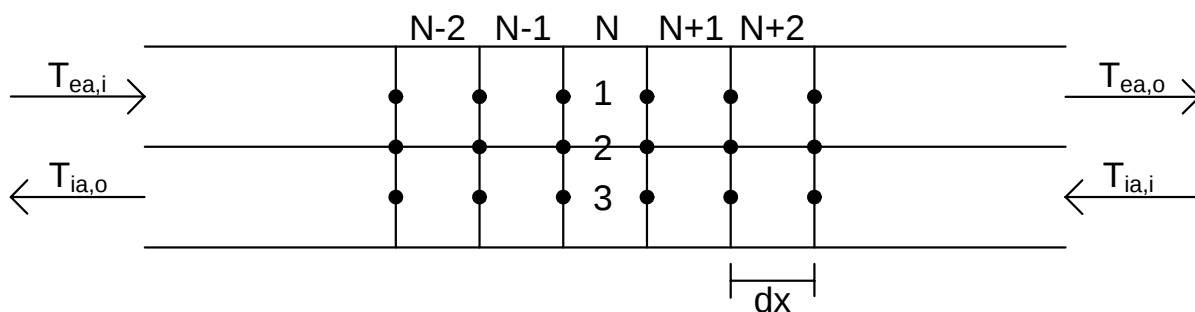
4.2 Formål

Formålet med dette arbejde er 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.

Først gennemgås teorien for modelleringen af en luft-til-luft modstrømsvarmeveksler.

4.3 Teori

Varmevekslermodellen er som udgangspunkt udviklet som en 1-dimensional stationær formulering, og som udgangspunkt diskretiseres problemstillingen som vist i Figur 4.1.



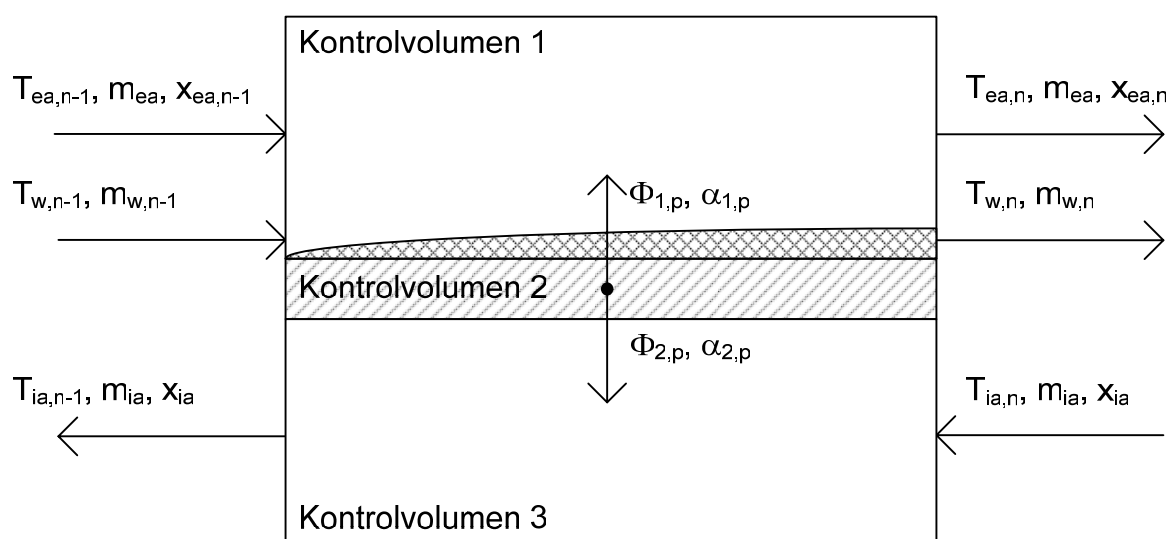
Figur 4.1 Diskretiseringen af problemstillingen.

Varmeveksleren opdeles således i et endeligt antal segmenter hvori varmeudvekslingen antages at forekomme som 1-dimensional og stationær. Antallet af segmenter nødvendigt for at opnå en god nøjagtighed i beregningen vurderes på baggrund af ændringen i varmeoverføringskoefficienten på langs med den varmevekslende skilleflade mellem luftstrømmene. Når modellen efterfølgende udvides til dynamiske (tidsvarierende) forhold vil den nødvendige beregningstid for f.eks. simulering af ét år være meget afhængig af antallet af segmenter, og derfor er det vigtigt at optimere forholdet mellem den nødvendige simuleringstid med behovet for detaljeringsgrad i modellen.

I hvert enkelt segment fastlægges hvilken situation der forekommer ud fra 4 muligheder; 1) ingen faseændring; 2) kondensation opstår; 3) kondensation fortsætter eller 4) isdannelse forekommer. Hvis det for et givet segment fastslås at der forekommer isdannelse, vil beregningen i første omgang gennemføres under antagelse af at kondens og isdannelsen ikke påvirker luftstrømmen, dvs. at der i denne situation udelukkende tages hensyn til ændringerne i varmeoverføringskoefficienterne. I praksis vil varmeveksleren fryse til meget hurtigt efter at isdannelse begynder, og afrimning eller lignende vil skulle igangsættes med det samme for at undgå dette. Den påvirkning som isdannelse i veksleren vil have på luftgennemstrømningen mv., vil ikke blive behandlet i nærværende sammenhæng, men som tidligere omtalt vil dette blive implementeret i forbindelse med det fortsatte arbejde med udviklingen af modellen.

I Figur 4.2 er vist varme- og massetransporten ind og ud af segment N.

For fraluften i Figur 4.2, øverst, er luften karakteriseret ved en indløbstemperatur $T_{ea, n-1}$, en massestrøm af tør luft $\dot{m}_{ea}$ 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 $\dot{m}_{w, n-1}$. Massestrømmen af tør luft $\dot{m}_{ea}$ regnes konstant, og derfor er de ubekendte for kontrol-volumen 1; udløbstemperaturen for luften, $T_{ea, n}$, fugtindholdet i luften som forlader kontrol-volumenet, $x_{ea, n}$ samt temperatur og massestrøm for vandet, $T_{w, n}$ og $\dot{m}_{w, n}$.



Figur 4.2 Varme- og massetransport i segment N.

Pladematerialet som adskiller de to luftstrømme modtager varme fra udsugningsluften dels via den konvektive varmeoverføring men også fra en eventuel kondens- eller frysevarme. Pladevæggens temperatur T_p skal altså fastlægges ved en varmebalance mellem tilføring af varme fra fraluften og afgivelse af varme til tilluften. Der tages ikke hensyn til hverken aksial varmeledning i pladematerialet, eller varmeledning på tværs af pladematerialet, denne simplificering diskuteres senere i rapporten.

For tilluften, nederst, er luften karakteriseret ved en temperatur $T_{ia, n}$, en masse-strøm af tør luft $\dot{m}_{ia}$ og et fugtindhold x_{ia} . Massestrømmen af tilluften samt fugtindholdet i luften antages at være konstant, og derfor er der for tilluften 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 [8]. 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 risiko for problemer med isdannelse minimeres samtidig med at ydeevnen maksimeres.

4.3.1 Simplifikationer

For at kunne opbygge en beregningsmodel er det nødvendigt at foretage nogle simplificerende antagelser vedrørende systemets opførsel. Følgende overordnede antagelser gøres i forbindelse med opbygningen af modellen:

- Varmeoverføringskarakteristika og fysiske egenskaber for materialerne er ikke afhængige af temperatur, position og tid.
- Lufthastigheden er konstant i hver af de to kanaler.
- Der forekommer ikke utætheder i systemet.
- I det tilfælde hvor der forekommer isdannelse i varmeveksleren, antages det at isen "bliver en del af" pladematerialet der adskiller luftstrømmene, svarende til at is og plademateriale har en fælles middeltemperatur.

Yderligere antagelser og simplificeringer vil blive diskuteret i forbindelse med selve modelformuleringen i de konkrete situationer hvor de forekommer.

4.3.2 Matematisk formulering

Hvis man betragter Figur 4.2, kan enthalpien som tilføres kontrolvolumen 1 skrives som:

$$h_{1,n-1} = (c_{pa} \cdot T_{ea,n-1} + x_{ea,n-1} \cdot (\Delta h_v + c_{pv} \cdot T_{ea,n-1})) \cdot \dot{m}_{ea} + \dot{m}_{w,n-1} \cdot c_{pw} \cdot T_{w,n-1} \quad (1)$$

hvor c_{pa} er den specifikke varmekapacitet for luft, $T_{ea,n-1}$ er temperaturen af den luft der tilføres kontrolvolumen 1, $x_{ea,n-1}$ er fugtindholdet af luften der tilføres kontrolvolumen 1, Δh_v er fordampningsvarmen for vand, c_{pv} er den specifikke varmekapacitet for vanddampen, $\dot{m}_{ea}$ er massestrømmen af luft i kontrolvolumen 1, $\dot{m}_{w,n-1}$ er massestrømmen af vand der tilføres kontrolvolumen 1, c_{pw} er den specifikke varmekapacitet for vand og $T_{w,n-1}$ er temperaturen af vand der tilføres kontrolvolumen 1.

Enthalpien som forlader kontrolvolumen 1 kan skrives som:

$$h_{1,n} = (c_{pa} \cdot T_{ea,n} + x_{ea,n} \cdot (\Delta h_v + c_{pv} \cdot T_{ea,n})) \cdot \dot{m}_{ea} + \dot{m}_{w,n} \cdot c_{pw} \cdot T_{w,n} \quad (2)$$

hvor $T_{ea,n}$ er temperaturen af den luft der forlader kontrolvolumen 1, $x_{ea,n}$ er fugtindholdet af luften der forlader kontrolvolumenet, $\dot{m}_{w,n}$ er massestrømmen af vand som forlader kontrolvolumen 1 og $T_{w,n}$ er temperaturen af vandet som forlader kontrolvolumen 1.

I kontrolvolumenet kan der dannes varme hvis enten kondens eller isdannelse forekommer. Kondensationsvarmen kan enten afsættes i skillevæggen mellem de varmevekslende luftstrømme eller i luften (tågedannelse). Den totale kondensationsvarme kan udtrykkes ved:

$$\Phi_{\text{cond}} = \Delta h_v \cdot \dot{m}_{ea} \cdot (x_{ea,n} - x_{ea,n-1}) \quad (3)$$

Den totale latente varme for isdannelse kan udtrykkes som:

$$\Phi_{\text{ice}} = \Delta h_m \cdot (\dot{m}_{w,n-1} + \dot{m}_{ea} (x_{ea,n-1} - x_{ea,n})) \quad (4)$$

hvor Δh_m er smeltevarmen for is.

Varmestrømmen fra kontrolvolumen 1 til varmevekslervæggen (kontrolvolumen 2) kan bestemmes ud fra udtrykket:

$$\Phi_{1,p} = \alpha_{1,p} \cdot (T_{p,n} - T_{ea,n}) - \Phi_{\text{cond}} \cdot K - \Phi_{\text{ice}} \quad (5)$$

hvor $\alpha_{1,p}$ er den konvektive varmeoverføringskoefficient mellem den varme fugtige luft og varmevekslervæggen, $T_{p,n}$ er temperaturen af pladevæggen, og K er en konstant som angiver hvor stor en del af kondensationsvarmen som afsættes i varmevekslervæggen, dvs. svarende til at den resterende

del afsættes i luften (tågedannelse). Som udgangspunkt antages det at $K = 1$, dvs. svarende til at al kondensationsvarmen afsættes i varmevekslervæggen.

For kontrolvolumen 3 kan enthalpien ind og ud samt varmestrømmen fra kontrolvolumen 3 til varmevekslervæggen (kontrolvolumen 2) fastlægges som hhv.:

$$h_{2,n} = (c_{pa} \cdot T_{ia,n} + x_{ia} \cdot (\Delta h_v + c_{pv} \cdot T_{ia,n})) \cdot \dot{m}_{ia} \quad (6)$$

hvor $T_{ia,n}$ er temperaturen af den luft som tilføres kontrolvolumen 3, x_{ia} er fugtindholdet af den luft som tilføres kontrolvolumen 2 og $\dot{m}_{ia}$ er massestrømmen af luften.

$$h_{2,n-1} = (c_{pa} \cdot T_{ia,n-1} + x_{ia} \cdot (\Delta h_v + c_{pv} \cdot T_{ia,n-1})) \cdot \dot{m}_{ia} \quad (7)$$

hvor $T_{ia,n-1}$ er temperaturen af den luft som forlader kontrolvolumen 3.

Varmestrømmen fra kontrolvolumen 3 til varmevekslervæggen (kontrolvolumen 2) kan bestemmes ud fra udtrykket:

$$\Phi_{2,p} = \alpha_{2,p} \cdot (T_{p,n} - T_{ia,n}) \quad (8)$$

hvor $\alpha_{2,p}$ er den konvektive varmeoverføringskoefficient mellem den kolde luft og varmevekslervæggen.

De ubekendte i ligning (1) – (8) er $T_{ea,n}$, $T_{p,n}$, $T_{ia,n-1}$, $T_{w,n}$, $\dot{m}_{w,n}$ og $x_{ea,n}$, hvor dog $\dot{m}_{w,n}$ og $x_{ea,n}$ er implicit afhængige, og derfor ender vi op med 5 ubekendte.

For at kunne fastlægge værdierne af disse ubekendte benytter vi følgende ligninger, som beskriver bevarelsen af energi for hver af de tre kontrolvolumener:

$$0 = h_{1,n-1} - h_{1,n} + \Phi_{1,p} \quad (9)$$

$$0 = h_{2,n} - h_{2,n-1} + \Phi_{2,p} \quad (10)$$

$$0 = \Phi_{1,p} + \Phi_{2,p} + C_{n-1} \cdot (T_{p,n} - T_{p,n-1}) + C_n \cdot (T_{p,n} - T_{p,n+1}) \quad (11)$$

og bevarelse af masse (dvs. relationen mellem $\dot{m}_{w,n}$ og $x_{ea,n}$):

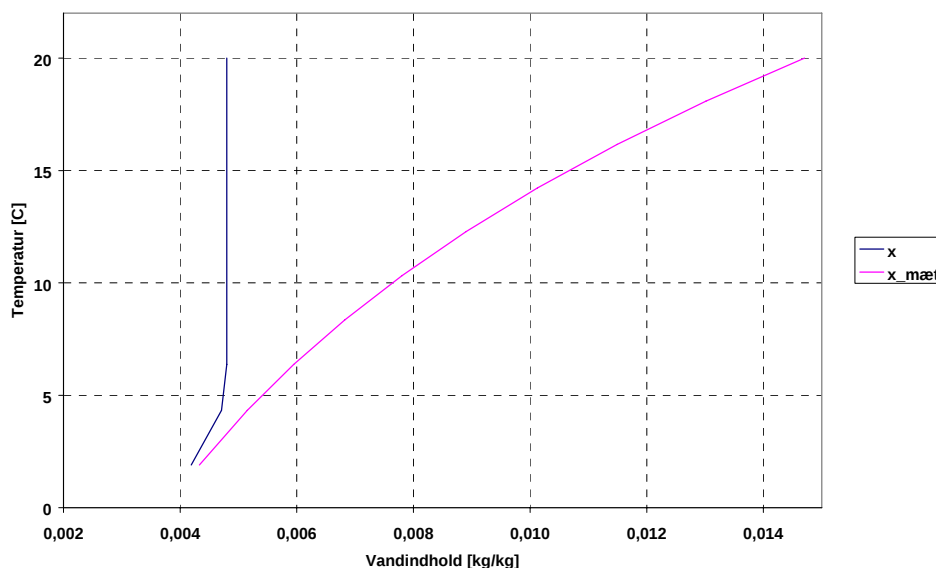
$$m_{w,n} = \begin{cases} m_{w,n-1} + (x_{ea,n-1} - x_{ea,n}) \cdot m_{ea}, & \text{ingen isdannelse} \\ 0, & \text{isdannelse} \end{cases} \quad (12)$$

Vi har således kun 3 ligninger som beskriver energibevarelsen i systemet og vi har behov for at fastlægge 4 temperaturer og massestrømmen af vand som forlader kontrolvolumen 1. Derfor gør vi følgende antagelser;

a) Temperaturen af det vand som forlader kontrolvolumen 1 vil have samme temperatur som varmevekslervæggen ved "udløbet", dvs. $T_{w,n} = T_{p,n}$.

b) Fugtindholdet i luften som forlader kontrolvolumen 1, $x_{ea,n}$, bestemmes som følger:

Det antages at kondenseringen forløber langs en ret linie i et I_x -diagram (enthalpi-vandindhold-diagram), fra indløbsforholdene mod mætningstilstanden ved pladens temperatur (ved udløbet), beskrevet i Danvak [2]. Metoden er illustreret i Figur 4.3. Metoden er egentlig udviklet til brug ved køleflader hvor temperaturforskellen mellem luft og køleflade normalt er væsentligt højere end forskellen på lufttemperatur og varmevekslerflade i en luft-luft varmeveksler, og derfor er det interessant at vurdere hvorvidt metoden er anvendelig i nærværende sammenhæng.



Figur 4.3 I_x -diagram hvor kondenseringen forløber som stykvis lineær mod mætningen.

Mætningsdamptrykket bestemmes ud fra følgende formel:

$$P_{\text{sat}} = \frac{T_{ea,n-1}}{T_{ea,n-1} + 0,62198} \cdot 101325 \quad (13)$$

og herudaf kan fastlægges dugpunktstemperaturen:

$$T_{\text{sat}} = 37,58 - \frac{4042,9}{\ln(P_{\text{sat}}) - 23,5771} - 273,15 \quad (14)$$

Hvis pladevæggens temperatur er lavere end dugpunktstemperaturen, så fastlægges fugtindholdet som:

$$x_{\text{ea},n} = \min \begin{pmatrix} x_{\text{ea},n-1} - \Delta x \\ x_{\text{sat},\text{ea},n} \end{pmatrix} \quad (15)$$

hvor $x_{\text{sat},\text{ea},n}$ er mætningsfugtindholdet for luften ved temperaturen $T_{\text{ea},n}$ og Δx er ændringen i luftens fugtindhold fastlagt ved følgende formel:

$$\Delta x = \frac{h_{1,n-1} - h_{1,n}}{\left(\frac{dI}{dx} \cdot \dot{m}_{\text{ea},n-1}\right)} \quad (16)$$

dI/dx er givet ved følgende udtryk:

$$\frac{dI}{dx} = \frac{\left(\frac{h_{1,n-1}}{m_{\text{ea},n-1}} - \left(c_{\text{pa}} \cdot T_{\text{p},n} + x_{\text{sat},\text{p},n} \cdot (\Delta h_v + c_{\text{pv}} \cdot T_{\text{p},n}) \right) \right)}{(x_{\text{ea},n-1} - x_{\text{sat},\text{p},n})} \quad (17)$$

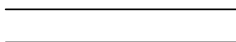
hvor $x_{\text{sat},\text{p},n}$ er mætningsfugtindholdet ved pladetemperaturen $T_{\text{p},n}$.

Hvis pladevæggens temperatur er højere end dugpunktstemperaturen, så er fugtindholdet det samme som i foregående kontrolvolumen, dvs. $x_{\text{ea},n} = x_{\text{ea},n-1}$.

Alternativt til denne metode kunne det antages at kondenseringen i varmeveksleren forløber direkte mod mætningskurven. Hvad forskellen på de to metoder medfører for beregningerne diskuteres i afsnit 4.6.1.

Varmeoverføringskoefficienterne $\alpha_{1,p}$ og $\alpha_{2,p}$ afhænger af hvorvidt luftstrømmene i varmeveksleren er laminare eller turbulente. Hvis luftstrømmene er laminare er Nusselt's tal konstant - dog afhængig af kanalernes tværsnit [3], dvs.:

Tabel 4.1 Nusselts tal for laminar strømning i rør med forskellige tværsnitsudformninger

	Ligesidet trekant	3,1
	Kvadrat	3,6
	Uendelig	8,235

og varmeoverføringskoefficienterne kan bestemmes af:

$$\alpha = \frac{Nu \cdot \lambda_{\text{air}}}{d_h} \quad (18)$$

Hvis luftstrømmene er turbulente kan Nusselt's tal bestemmes af Gnielinski's formel [3]:

$$Nu = \frac{(f/8) \cdot (Re - 1000) \cdot Pr}{1 + 12.7 \cdot (f/8)^{1/2} \cdot (Pr^{2/3} - 1)} \quad (19)$$

Hvor friktionsfaktoren, f , kan udtrykkes som [3]:

$$f = (0.790 \cdot \ln(Re) - 1.64)^{-2} \quad (20)$$

Anvendelsen af ovenstående formel er under antagelse af at følgende er gældende; Reynold's tal skal ligge i intervallet 3000 til $100 \cdot 10^6$, dvs. svarende til at luftstrømmene er turbulente og Prandtl's tal skal ligge i intervallet 0,5 til 1,5, dvs. svarende til at varmevekslervæggen kan antages værende hydraulisk glat.

Reynold's tal kan beregnes ved hjælp af følgende formel [3]:

$$Re = \frac{v \cdot d_h}{\nu} \quad (21)$$

Ved at indsætte (20) og (21) i (19) og benytte dette i (18) kan fastlægges varmeoverføringskoefficienterne for tilfældet hvor luftstrømmene er turbulente.

4.4 Validering af den stationære model

Den 1-dimensionale stationære model valideres ved sammenligninger med målinger på konkrete varmevekslere under kontrollerede forhold. For den stationære model er der gennemført en enkelt sammenligning af modellens resultater med målinger fra en forsøgsopstilling i BYG·DTU's forsøgshal. Den varmeveksler som er brugt i den pågældende forsøgsopstilling er en modstrømsvarmeveksler af mærket Recair Sensitive [4] med en længde på ca. 0,3 m, svarende til at den skulle have en temperatureffektivitet på ca. 90 % ved en luftstrøm på 50 m³/h.

I forsøget sendes fraluften med temperatur 20,0 °C og fugtindhold på 0,0048 kg/kg ind fra den ene side og udeluften 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 tilluften har når den forlader veksleren.

I laboratorieforsøget er følgende resultater opnået:

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

$$x_{\text{afkast}} = 0,0036 \text{ kg/kg}$$

$$T_{\text{tilluft}} = 17,5 \text{ }^{\circ}\text{C}$$

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

$$T_{\text{afkast}} = 1,9 \text{ }^{\circ}\text{C}$$

$$x_{\text{afkast}} = 0,0042 \text{ kg/kg}$$

$$T_{\text{tilluft}} = 17,3 \text{ }^{\circ}\text{C}$$

Sammenligner man de to sæt resultater ses det 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. Vigtigst er det at bemærke at der i modellen ikke udkondenseres ligeså store mængder vand som der gør i forbindelse med målingerne.

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:

$$T_{\text{afkast}} = 0,6 \text{ }^{\circ}\text{C}$$

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

$$T_{\text{tilluft}} = 16,9 \text{ }^{\circ}\text{C}$$

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 1,9 °C til 0,6 °C. Dette viser at kondenseringen har en stor betydning og at det dermed er vigtigt at medtage denne i modellen for at kunne simulere forholdene i varmeveksleren korrekt.

Som tidligere nævnt har opdelingen af modellen indflydelse på nøjagtigheden af beregningen, og derfor vil det være interessant at undersøge hvorvidt en opdeling i 10 delområder er tilstrækkeligt til at opnå en acceptabel beregningsnøjagtighed. Der gennemføres derfor en beregning hvor antallet af delområder fordobles, dvs. at der nu benyttes 20 delområder frem for 10.

$$T_{\text{afkast}} = 1,8 \text{ }^{\circ}\text{C}$$

$$x_{\text{afkast}} = 0,0042 \text{ kg/kg}$$

$$T_{\text{tilluft}} = 17,3 \text{ }^{\circ}\text{C}$$

Modellen med 20 celler får altså en lidt lavere afkasttemperatur, hvilket svarer bedre til målingen, og for tillufttemperaturen og fugtindholdet af afkastluften fås samme værdier som for tilfældet hvor der benyttes 10 celler. Forskellen i de to beregninger er imidlertid så lille at det ikke umiddelbart synes nødvendigt at benytte en mere detaljeret model, idet fugtindhold og tillufttemperatur ikke ændres væsentligt ved den højere detaljeringsgrad. Selvfølgelig vil der kunne forekomme situationer hvor det kan være nødvendigt at operere med flere inddelinger af en varmeveksler, og behovet for dette vil i høj grad afhænge af gradienterne for varmeoverføringskoefficienterne. Man kan altså sige, at desto højere grad af kompleksitet, dvs. specielt i forbindelse med kondens/isdannelse, desto større behov for detaljeringsgrad.

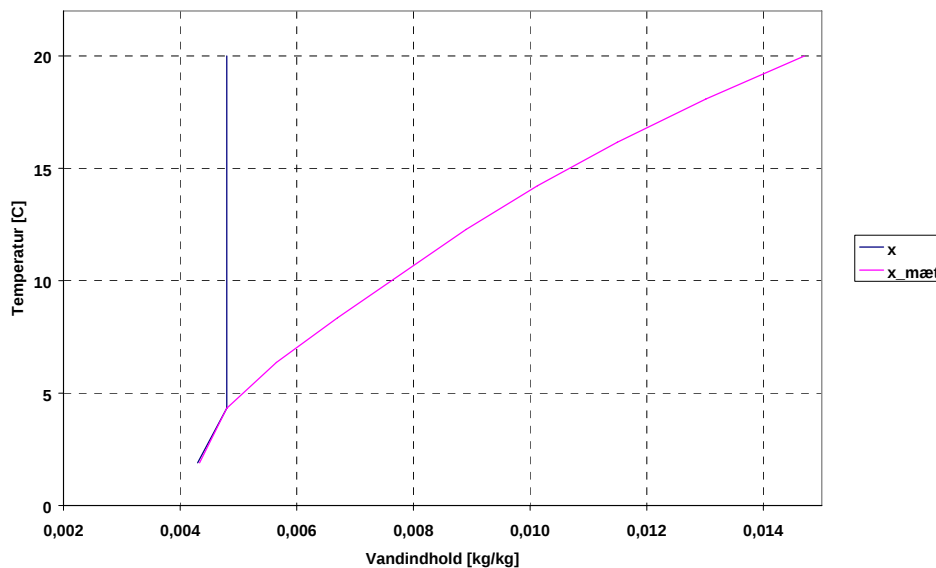
4.5 Beskrivelse af dynamisk model

For at kunne modellere og analysere varmeveksleres dynamiske egenskaber, herunder kondens, isdannelse samt afrimning, er det nødvendigt at kunne gennemføre dynamiske simuleringer, som tager højde for ophobning og fjernelse af vand og is. Den stationære model udvides derfor, således at den kan udføre beregninger under dynamiske forhold.

Som udgangspunkt er Excel [5] benyttet i forbindelse med udviklingen af den stationære model. Excel er imidlertid ikke særligt egnet til at regne på dynamiske forhold, og derfor overføres modellen fra Excel til Simulink [6] (et dynamisk simuleringsmodul under Matlab [7]). Ligningssystemerne bibeholdes fra Excel, men i oversættelsen til Simulink inkorporeres automatisk tidsafhængige forhold, svarende til at Simulink erstatter ligningssystemet beskrevet tidligere med et system af differentilligninger som beskriver varmembalancerne i modellen.

I forbindelse med ”oversættelsen” af modellen fra stationære til dynamiske forhold, er der en række andre forhold der også ændres. I den stationære model blev temperaturer og varmeoverføringskoefficienter beregnet ved ud- og indløb til hvert enkelt kontrolvolumen, se figur 3.1. I den dynamiske model er knudepunkterne for beregningen placeret midt i kontrolvolumenerne. Årsagen til denne forskel er dels, at man under stationære forhold ikke behøver at medtage varmekapaciteten af luft og varmeveksler i beregningerne, og derfor betyder det mindre hvor temperaturerne beregnes, og dels at der i Simulink opstod stabilitetsproblemer i modellen hvis beregningsknuderne blev placeret som i den stationære model.

En anden forskel på de to modeller er, at det i den stationære model blev antaget at kondenseringen forløb langs en ret linie i et I_x -diagram (enthalpi-vandindhold-diagram), fra indløbsforholdene mod mætningstilstanden ved pladens temperatur, som beskrevet tidligere. I den dynamiske model antages det at kondenseringen forløber direkte mod mætning og herefter følger mætningskurven, se figur 4.4. Hvorledes disse forhold påvirker beregningen vil blive diskuteret senere i rapporten.



Figur 4.4 ix -diagram hvor kondenseringen forløber direkte mod mætning og herefter følger mætningskurven.

Den dynamiske model vil kunne regne på både stationære og dynamiske forhold, idet stationære forhold kan opnås ved at påføre modellen ”statiske” randbetingelser. Dette medfører at vi som udgangspunkt kan foretage en række sammenligninger mellem de to modeller, således at det dels sikres at de to modeller opnår samme resultater, svarende til at den statiske model er ”oversat” korrekt, og dels vil de ovennævnte forskelle der er i modellerne kunne vurderes.

I forbindelse med at modellen skal regne på dynamiske forhold er det nødvendigt at indføre varmekapaciteter i beregningen, herunder specielt luftens varmekapacitet samt varmekapaciteten af selve varmevekslermaterialet.

Der er på nuværende tidspunkt ikke medtaget aksial ledning i varmeveksler-materialet i den dynamiske model, idet aksial ledning kan skabe problemer når der i et kontrolvolumen opstår kondens. Ved kondensens indtræden vil temperaturen af pladematerialet i den ene ende af kontrolvolumenet være forholdsmeæssigt meget større end temperaturen i den anden ende af kontrolvolumenet, hvilket medfører en relativt stor varmeledning i pladematerialet. Dette kan medføre stabilitetsproblemer i forbindelse med Simulink’s indbyggede ligningsløser, hvilket medfører ekstremt små tidsskridt, dvs. meget lange beregningstider. Inkorporeringen af aksial ledning i pladematerialet vil derfor udføres i forbindelse med det videre arbejde med udviklingen af modellen, se afsnit 4.8.

4.6 Validering af dynamisk model

Den dynamiske model valideres dels i forhold til den tidligere beskrevne stationære model, og dels i forhold til en række dynamiske målinger på en konkret varmeveksler under forskellige forhold. Valideringen i forhold til den stationære model gennemføres primært for at vurdere om ”oversættelsen” af modellen er foretaget korrekt, men også for at vurdere hvorvidt enkelte mindre afvigelser mellem den stationære og den dynamiske model har væsentlig betydning for resultaterne. Disse

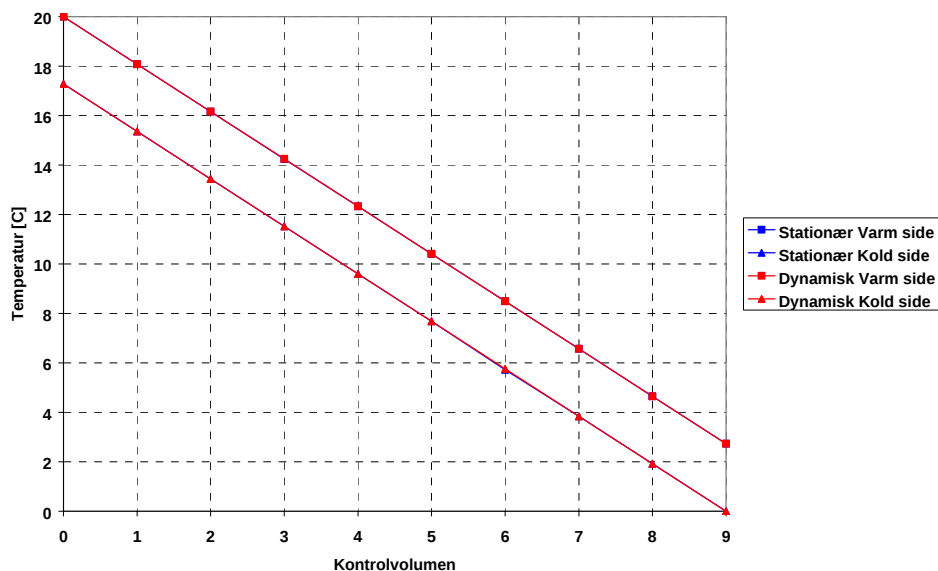
sammenligninger foretages for stationære forhold. Valideringen i forhold til dynamiske målinger på en konkret varmeveksler gennemføres for at vurdere modellens anvendelighed i forbindelse med vurderinger af kondens og frostdannelse i konkrete varmevekslerudformninger, samt muligheden for afrimning af opstået is.

4.6.1 Sammenligning mellem stationær og dynamisk model. Ingen kondens.

I den første sammenligning mellem modellerne benyttes der en fraluft med en temperatur på 20 °C, et fugtindhold på 0 kg/kg og en massestrøm på 0,0578 kg/s. Tilluften har en temperatur på 0 °C og en massestrøm på 0,0578 kg/s. Der er således intet fugtindhold i fraluft, og der forekommer dermed hverken kondens eller isdannelse i veksleren.

Den varmeveksler der er modelleret er en kommerciel modstrømsvarmeveksler af polystyren fra firmaet Recair, som typisk benyttes i forbindelse med ventilationsanlæg til enfamiliehuse, se [4].

Som tidligere omtalt medtages effekten af den aksiale ledning i varmevekslermaterialet ikke, og dette gælder generelt for sammenligningerne. Ved at benytte et fugtindhold på 0 kg/kg vil forskellen i modellerne om hvorledes kondensen forløber i Ix-diagrammet være uden betydning. Hermed skabes det simpleste sammenligningsgrundlag. Figur 4.5 viser temperaturforløbet i veksleren.



Figur 4.5 Temperaturforløb i stationær og dynamisk model. Ingen fugt i fraluft.

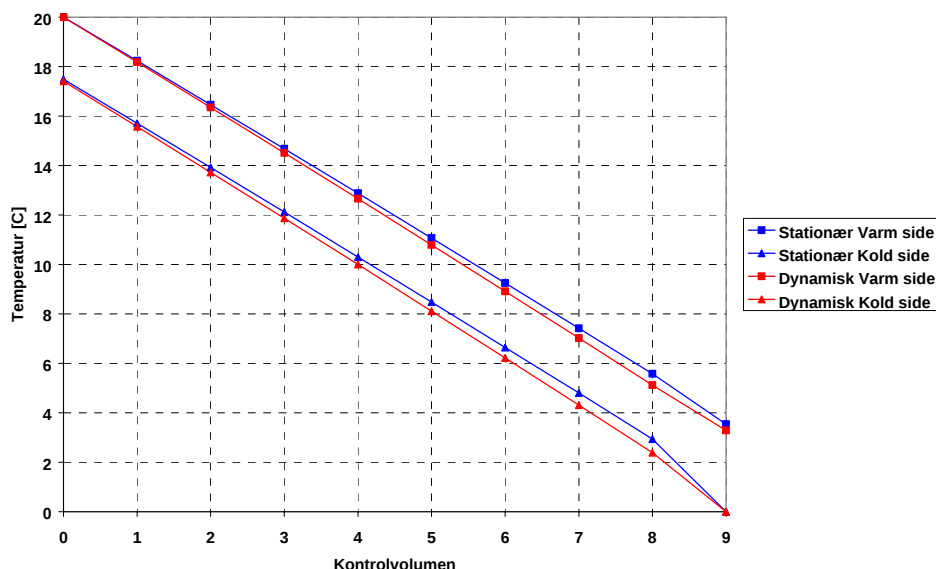
I figur 3.5 er det tydeligt at modellerne giver stort set samme resultater (kurverne ligger oven i hinanden). Der forekommer enkelte afvigelser af størrelsesordenen 0,01 °C, men det må konkluderes at de to modeller regner ens på det meget simple tilfælde hvor der ikke forekommer kondens. Årsagen til at der ikke ses nogen effekt af det forhold, at knudepunkterne for pladematerialets temperatur er defineret forskelligt i de to modeller er, at varmeoverføringen er konstant gennem hele veksleren, og derfor vil denne forskel ikke komme til udtryk i nærværende sammenhæng. Når der modelleres

situationer hvor der forekommer kondens, vil varmeoverføringen ikke være konstant, og så vil en eventuel forskel mellem de to modeller kunne ses.

Af den første sammenligning kan således konkluderes at modellen er oversat korrekt i forbindelse med overgangen fra Excel til Simulink.

4.6.2 Sammenligning mellem stationær og dynamisk model. Let kondens.

I den anden sammenligning mellem modellerne betragtes nu en situation hvor der er et mindre fugtindhold i fraluften. Fraluft har en temperatur på 20 °C, et fugtindhold på 0,005 kg/kg og en massestrøm på 0,0578 kg/s. Tilluften har en temperatur på 0 °C og en massestrøm på 0,0578 kg/s. Hvis man betragter et Ix-diagram vil det være tydeligt at den beskrevne situation vil afstedkomme en lettere kondens i varmeveksleren, og det er derfor interessant at sammenligne modellerne mht. hvor meget vand der udkondenseres, samtidig med temperaturforløbet i veksleren. I figur 4.6 er optegnet temperaturforløbet for de to beregninger.



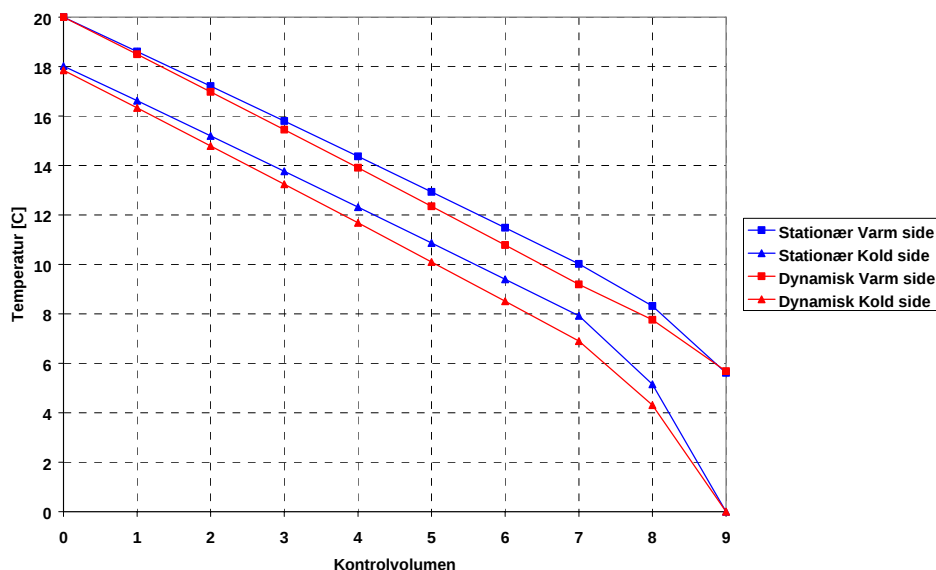
Figur 4.6 Temperaturforløb i stationær og dynamisk model. Lidt fugt i fraluft.

Betragter man figur 4.6 er det tydeligt at modellerne fortsat giver resultater som er stort set ens. Den primære årsag til afvigelsen mellem de to beregninger ligger, som tidligere omtalt, i den metode som benyttes i modellerne til at beskrive hvorledes kondenseringen forløber i Ix-diagrammet. Af figuren ses det at den stationære model generelt giver temperaturer som er højere end den dynamiske model, specielt i den kolde ende af varmeveksleren. Dette skyldes at modelleringen af kondenseringen i den stationære model medfører en hurtigere afgivelse af fugten fra luften til pladematerialet som adskiller de to luftstrømme, og dermed opstår der mere kondenseringsvarme i denne model. I den stationære model udkondenseres der i alt $2,06 \cdot 10^{-5}$ kg/s mens der i den dynamiske model udkondenseres $1,26 \cdot 10^{-5}$ kg/s, hvilket som nævnt medfører de lidt højere temperaturer. Den relati-

ve afvigelse i resultaterne er ca. 39 %, men da der er tale om meget små fugtmængder er det svært at konkludere yderligere vedrørende forskellen i modelleringen af kondenseringen.

4.6.3 Sammenligning mellem stationær og dynamisk model. Mere kondens.

I den tredje sammenligning mellem modellerne øges fugtindholdet i fraluften lidt, således at der forekommer en større kondensation. Fraluften har en temperatur på 20 °C, et fugtindhold på 0,007 kg/kg og en massestrøm på 0,0578 kg/s. Tilluften har fortsat en temperatur på 0 °C og en massestrøm på 0,0578 kg/s. I figur 4.7 er optegnet temperaturforløbet for de to beregninger.



Figur 4.7 Temperaturforløb i stationær og dynamisk model. Mere fugt i fraluft.

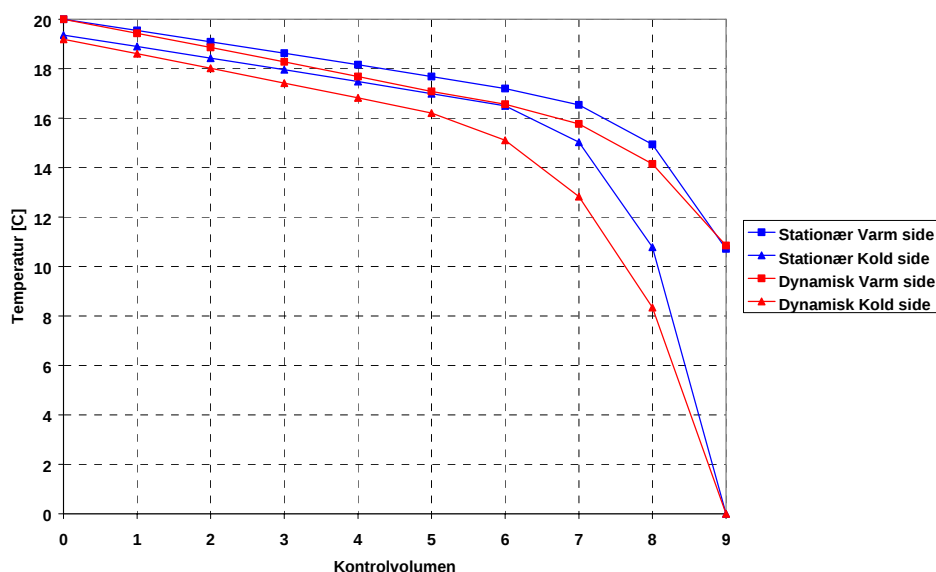
Modellerne giver således fortsat sammenlignelige resultater, dog er afvigelserne i temperaturerne tydeligvis større end i den foregående sammenligning. Igen er det primært forskellen i hvorledes kondenseringen forløber i Ix-diagrammet som giver forskellen, og igen er det tydeligt at temperaturerne i den stationære model generelt er højere end temperaturerne i den dynamiske model.

I den stationære model udkondenseres der i alt $8,04 \cdot 10^{-5}$ kg/s mens der i den dynamiske model udkondenseres $7,75 \cdot 10^{-5}$ kg/s. Den relative forskel mellem resultaterne svarer til ca. 4 %, og hermed er det altså tydeligt, at den relativt store forskel der gjorde sig gældende i den foregående sammenligning skyldes at der er tale om meget små fugtmængder. I sammenligningen hvor der er mere fugt i fraluften er resultaterne meget ens, både hvad angår temperaturerne i varmeveksleren og den udkondenserede fugtmængde.

4.6.4 Sammenligning mellem stationær og dynamisk model. Kraftig kondens.

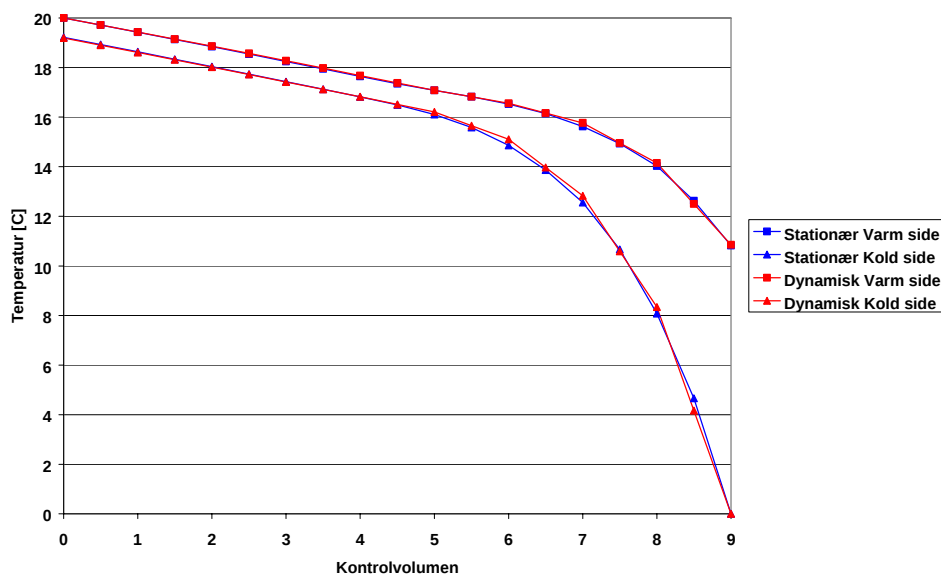
I den fjerde og sidste sammenligning mellem modellerne øges fugtindholdet i fraluften igen, således at der vil forekomme en meget kraftig kondensation i varmeveksleren. Fraluften har nu en tempera-

tur på 20 °C, et fugtindhold på 0,012 kg/kg og en massestrøm på 0,0578 kg/s. Tilluften har fortsat en temperatur på 0 °C og en massestrøm på 0,0578 kg/s. I Figur 4.8 er optegnet temperaturforløbet for de to beregninger.



Figur 4.8 Temperaturforløb i stationær og dynamisk model. Meget fugt i fraluft.

Modellerne giver således fortsat sammenlignelige resultater, dog er afvigelserne i temperaturerne nu endnu større end i de tidligere sammenligninger, og det er fortsat den stationære model der giver de højeste temperaturer i veksleren. Årsagen til de store afvigelser der forekommer i temperaturerne når resultaterne fra de to modeller sammenlignes er, at hvis der dannes kondens i et givet kontrolvolumen i den stationære model, antages kondensen at forekomme i hele det pågældende kontrolvolumen, og dermed påvirkes temperaturerne i kontrolvolumenet kraftigt. I den dynamiske model derimod, vil kondensen først have indflydelse på temperaturerne hvor den forekommer, og denne model er altså væsentligt mere nøjagtig end den stationære. Hvis man således øger antallet af kontrolvolumener i den stationære beregning, vil unøjagtighederne reduceres, og teoretisk set vil den, når der benyttes uendeligt mange kontrolvolumener, være identisk med den dynamiske model. I Figur 4.9 er beregningen gentaget med 20 kontrolvolumener.



Figur 4.9 Temperaturforløb i stationær og dynamisk model. Meget fugt i fraluft. 20 kontrolvolumener.

Betragtes Figur 4.9 er det tydeligt at antallet af kontrolvolumener har stor betydning for hvor godt de to modeller stemmer overens.

I den stationære model udkondenseres der i alt $23,10 \cdot 10^{-5}$ kg/s mens der i den dynamiske model udkondenseres $22,69 \cdot 10^{-5}$ kg/s, svarende til en relativ afvigelse på under 2 %. Igen er der tale om meget små afvigelser mellem de to modellers resultater, og det er ligeledes tydeligt at den relative forskel i mængden af kondens bliver mindre desto mere kondens der er i beregningen.

4.6.5 Sammenfatning af modelsammenligninger.

Hvis man sammenfatter sammenligningerne mellem den stationære og den dynamiske model, er det tydeligt at det har betydning hvorledes kondenseringen foregår i Ix-diagrammet, men at dette gør sig specielt gældende når der er tale om mindre mængder kondensation mens kraftigere kondensation medfører at modellerne regner mere ens. Modellerne er imidlertid udviklet til at simulere situationer med meget lave udetemperaturer, hvor der typisk vil forekomme kraftig kondens eller isdannelse, og derfor må konklusionen være at metoden til beskrivelse af hvorledes kondenseringen foregår i den dynamiske model er fuldt ud så anvendelig som metoden benyttet i den stationære model.

Med hensyn til afvigelserne i de temperaturer som aflæses i den kolde ende af varmeveksleren, er det ligeledes tydeligt at disse bliver større desto mere kondens der kommer i varmeveksleren, og at der generelt forekommer højere temperaturer i den stationære model. Dette forhold skyldes også den metode der benyttes til at beskrive kondenseringen i veksleren. I den stationære model forløber kondenseringen stykkevist lineært mod mætningen, og forekomsten af kondens fastlægges på baggrund af udløbstemperaturen for et givet kontrolvolumen. Hvis der således haves en situation i et givet kontrolvolumen, hvor der forekommer kondens, vil kondensen antages at forekomme i hele

det pågældende kontrolvolumen uanset at den muligvis først forekommer meget tæt på udløbet. Derfor vil antallet af kontrolvolumener i modellen være helt afgørende for hvordan kondenseringen forløber. Desto flere kontrolvolumener, desto mere vil den stationære model altså nærme sig den dynamiske.

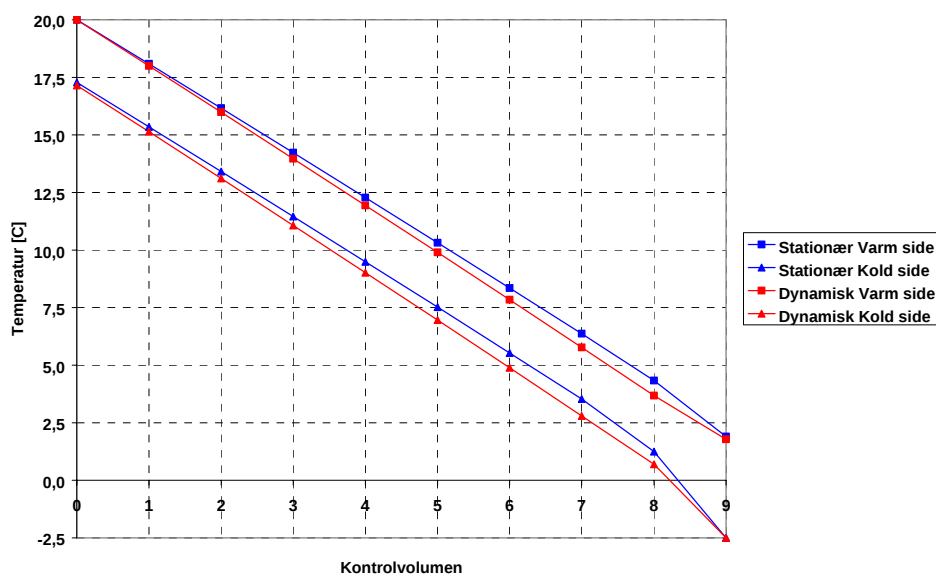
4.6.6 Sammenligning mellem laboratoriemålinger og model. Kommerciel varmeveksler

Den varmeveksler som blev brugt som eksempel ved de foregående sammenligninger mellem de to beregningsmodeller er der lavet målinger på i laboratoriet, og derfor foretages nu en sammenligning med disse målinger.

Der er gennemført en måling på varmeveksleren, hvor fraluften havde en temperatur på 20 °C, et fugtindhold på 0,0048 kg/kg og en massestrøm på 0,0578 kg/s. Tilluften havde i laboratorieforsøget en temperatur på -2,5 °C og en massestrøm på 0,0578 kg/s. Alle disse data indtastes i de to modeller og der gennemføres beregninger.

I forsøget er temperaturen for afkastluften målt til 1,5 °C, fugtindholdet i afkastluften til 0,0036 kg/kg og tillufttemperaturen til 17,3 °C.

I figur 4.10 er optegnet temperaturforløbet for de to beregninger.



Figur 4.10 Temperaturforløb i stationær og dynamisk model. Modelling af laboratorieforsøg.

I den stationære model er temperaturen af afkastluften beregnet som 1,9 °C og fugtindholdet i afkastluften er beregnet som 0,0042 kg/kg. Temperatur af tilluften er beregnet til 17,3 °C og der ud-kondenseres i alt $3,54 \cdot 10^{-5}$ kg/s.

I den dynamiske model er temperaturen af afkastluften beregnet som 1,8 °C og fugtindholdet i afkastluften er beregnet som 0,0043 kg/kg. Temperatur af tilluften er beregnet til 17,2 °C og der udkondenseres i alt $2,95 \cdot 10^{-5}$ kg/s.

I tabel 4.2 er resultaterne opstillet til sammenligning.

Tabel 4.2 Resultater af laboratoriemåling samt resultater for stationær og dynamisk model.

	Laboratoriemåling	Stationær model	Dynamisk model
Temperatur af afkastluft	1,5 °C	1,9 °C	1,8 °C
Temperatur af tilluft	17,3 °C	17,3 °C	17,2 °C
Fugtindhold i afkastluft	0,0036 kg/kg	0,0042 kg/kg	0,0043 kg/kg

Sammenlignes først temperaturerne er det tydeligt at både den stationære og den dynamiske model opnår resultater som stemmer rimeligt godt overens med de målte værdier. Sammenligner man fugtindholdet i afkastluften stiller sagen sig noget anderledes, og både den stationære og den dynamiske model giver resultater som er væsentligt højere end hvad der gjorde sig gældende i laboratoriemålingen, svarende til at der ikke udkondenseres lige så meget fugt.

I laboratoriemålingen er fugtindholdet af afkastluften baseret på en måling af den relative fugtighed, og derfor er det aktuelle fugtindhold beregnet på baggrund af denne størrelse samt temperaturen for afkastluften. Begge disse størrelser er i forbindelse med målingerne forbundet med relativt store usikkerheder og derfor er det interessant at vurdere hvor stor en betydning målenøjagtigheden for det benyttede udstyr kan have for de fremkomne resultater.

I laboratoriemålingen er temperaturen af afkastluften målt til 1,5 °C. Målingerne er foretaget med en type TT termoelement, som typisk har en nøjagtighed på $\pm 0,5$ °C. I beregningen er der i den dynamiske model fastlagt en temperatur for afkastluften på 1,8 °C, hvilket ligger vel indenfor målenøjagtigheden for termokoblerne. Hvis man omregner fugtindholdet fra målingen fra de oprindelige 1,5 °C til en ny temperatur på 1,8 °C, vil man få et fugtindhold af luften på ca. 0,0038 kg/kg.

I laboratoriemålingen er den relative luftfugtighed målt vha. en RF måler af typen HIH-3610 fra Honeywell og den har et måleområde fra 0-100 % RF med en usikkerhed på ± 2 % RH. Hvis vi antager at målinger er 2 % lavere end reelt, vil fugtindholdet være ca. 0,0038 kg/kg. Hvis man kombinerer de to fejlkilder, stadig indenfor målenøjagtigheden af udstyret, vil man altså kunne ende op med at fugtindholdet for afkastluften kunne være ca. 0,0040 kg/kg, dvs. svarende meget godt til de beregnede værdier.

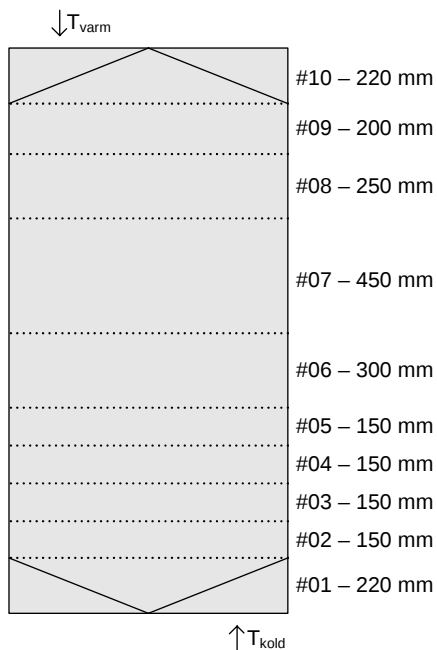
En endnu større usikkerhed i forbindelse med målingerne, forekommer i forbindelse med den nøjagtige placering af måleudstyret, og ikke mindst det forhold, at det er meget svært at kontrollere at temperaturmålingen af afkastluften er en "ren" lufttemperaturmåling, som ikke er påvirket af de omkringliggende overfladers temperatur, dvs. om temperaturføleren er i kontakt med overfladerne.

Disse forhold er vanskelige at kontrollere i den pågældende måleopstilling, og derfor vil det i forbindelse med den videre udvikling og verificering af den dynamiske model, være nødvendigt at benytte mere kontrollerede laboratorieforsøg til sammenligningerne.

4.6.7 Sammenligning mellem laboratiormålinger og model. Nyudviklet varmeveksler

Som beskrevet tidligere er der foretaget en række detaljerede laboratiormålinger på den under dette projekt udviklede varmeveksler. Varmeveksleren er nærmere beskrevet i rapportens kapitel 2. Målingerne benyttes nu til at lave en validering af modellen under stationære forhold. Først gennemføres sammenligninger hvor der ikke forekommer kondens i varmeveksleren og efterfølgende sammenlignes situationer med hhv. let kondensering, kraftig kondensering og isdannelse. I forbindelse med situationen hvor der forekommer isdannelse i veksleren gennemføres simuleringer af forholdene med kontinuerlig afrimning af veksleren ved skift mellem de to sektioner.

For at kunne sammenligne beregninger og målinger er det nødvendigt at opdele modellen i kontrolvolumener som passer med de temperaturmålinger der er lavet i kassevarmeveksleren. I figur 4.11 er vist opdelingen af veksleren i kontrolvolumener.



Figur 4.11 Opdelingen af modellen i kontrolvolumener.

For hvert af kontrolvolumenerne skal varmeoverføringskoefficienterne fastlægges. For de kvadratiske kanaler fastlægges varmeoverføringskoefficienten som:

$$h_c = \frac{Nu \cdot \lambda_{\text{luft}}}{D_h} = \frac{3,6 \cdot 0,026}{0,004} = 23,4 \text{ W} / \text{m}^2 \text{ K}$$

For de parallelle plader fastlægges varmeoverføringskoefficienten på tilsvarende vis som:

$$h_c = \frac{Nu \cdot \lambda_{luft}}{D_h} = \frac{8,235 \cdot 0,026}{0,008} = 26,8 W / m^2 K$$

Omkring ind- og udløb til veksleren er det varmevekslende areal halveret i forhold til det fulde areal, idet ind- og udløbet kun dækker halvdelen af det samlede areal, se figur 2.3. Ydermere foregår varmevekslingen i dette område nærmest som i en krydsveksler, og derfor divideres den samlede varmeoverføringskoefficient med endnu en faktor 2, således at den samlede varmeoverføringskoefficient for dette område reduceres med en faktor 4.

De rektangulære kanaler er 4 x 4 mm og mellem de parallelle plader er der også 4 mm. Godstykkelsen for varmevekslerpladerne er 0,5 mm. Der er i alt 10 ribbeplader i varmeveksleren, og hver af disse ribbeplader har en højde på 2,0 m og en bredde på 1,0 m og en tykkelse på 5 mm. En mere detaljeret beskrivelse af varmeveksleren kan findes i rapportens kapitel 3. På baggrund af varmevekslerens dimensioner kan fastlægges det samlede volumen luft og det samlede volumen varmevekslerplade som indeholdes i hvert enkelt kontrolvolumen, og herudfra kan fastlægges varmekapaciteten for hvert enkelt kontrolvolumen. I tabel 4.3 er opstillet de karakteristiske værdier for varmevekslermodellen.

Tabel 4.3 Varmeoverføringskoefficienter og volumen af luft og plade for kontrolvolumener.

	Kvadratiske	Parallelle	Kvadratiske	Parallelle	
	h	h	V, Luft	V, Luft	V, plade
Kontrolvolumen	[W/K]	[W/K]	[m ³ ·10 ³]	[m ³ ·10 ³]	[m ³ ·10 ⁴]
10	12,7	14,6	2,2	2,4	7,5
9	34,8	39,8	3,0	3,3	10,2
8	34,8	39,8	3,0	3,3	10,2
7	34,8	39,8	3,0	3,3	10,2
6	34,8	39,8	3,0	3,3	10,2
5	69,5	79,6	5,9	6,6	20,5
4	104,2	119,4	8,9	9,9	30,7
3	57,9	66,3	5,0	5,6	17,1
2	46,3	53,1	4,0	4,4	13,6
1	12,7	14,6	2,2	2,4	7,5

I figur 3.7 er vist forsøgsopstillingen som er benyttet, og på figuren kan ligeledes ses placeringen af temperaturmålinger i varmeveksleren. I det forsøg som betragtes er der benyttet en fralufttemperatur på 22,7 °C med en massestrøm på 0,0425 kg/s og en relativ fugtighed på ca. 60%. Tilluftens temperatur var 8,1 °C med en massestrøm på 0,0427 kg/s.

Forsøget viste at tillufttemperaturen var 19,4 °C mens temperaturen af afkastluften var 11,7 °C.

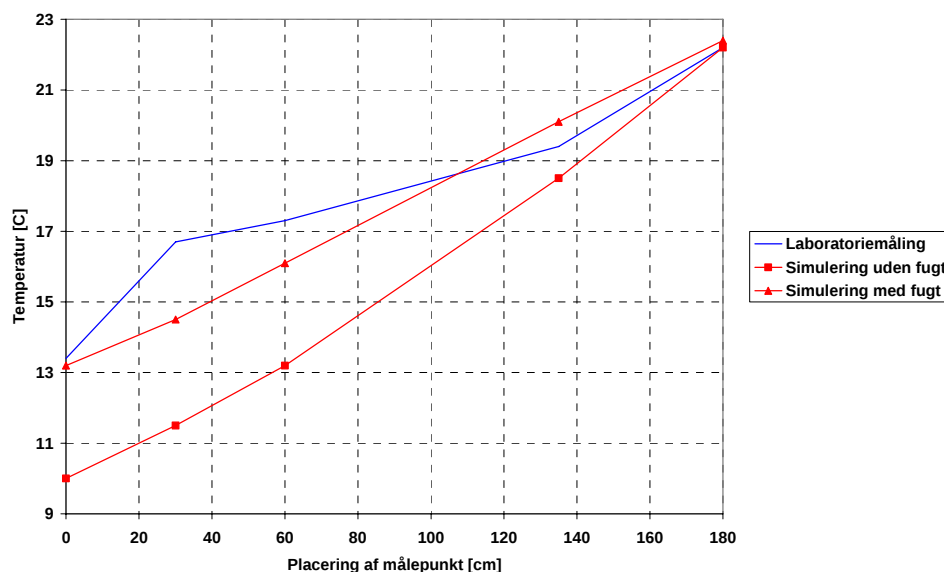
I forbindelse med laboratorieforsøget er der ikke konstateret kondens i varmeveksleren, men det må være forventeligt at der opstår kondens når luft med en temperatur på 22,7 °C og en relativ fugtighed på 60%, svarende til et dugpunkt på ca. 14,5 °C afkøles til 11,7 °C, som forsøget viste. Årsagen til at der ikke er konstateret kondens i varmeveksleren er formentlig, at der har været tale om små mængder som ikke har afstedkommet opsamling af vand.

I tabel 4.4 er opstillet resultaterne af temperaturmålingerne i varmeveksleren. I tabellen er ligeledes opstillet de beregningsresultater der er opnået med den dynamiske model, dels for en situation hvor der ikke forekommer kondens og dels hvor der opstår kondens i varmeveksleren. I det tilfælde hvor der ikke opstår kondens er tilluftens fugtindhold sat til 0 kg/kg i modellen. Denne ekstra beregning medtages for at undersøge hvilken betydning det har for forholdene i varmeveksleren om der forekommer kondens eller ej. Målepunkterne 0, 30, 60, 135 og 180 svarer til punkterne vist med T'er på figur 3.7.

Tabel 4.4 Resultaterne af laboratiormåling samt beregningsresultater med og uden fugt, afkastluft.

	Laboratorie- måling	Simulering uden fugt	Simulering med fugt
Målepunkt	T [°C]	T [°C]	T [°C]
180 (top)	22,2	22,2	22,4
135	19,4	18,5	20,1
60	17,3	13,2	16,1
30	16,7	11,5	14,5
0 (bund)	13,4	10,0	13,2
Afkast	11,7	9,7	12,8
Tilluft	19,4	19,8	20,9

I Figur 4.12 er optegnet temperaturforløbet for laboratiormåling og de to beregninger.



Figur 4.12 Temperaturforløb af afkastluft i laboratorieforsøg samt model med og uden fugt.

Betragtes Figur 4.12 er det tydeligt at simuleringen hvor fugten medtages i beregningen giver resultater som ligger tættere på resultaterne fra laboratorieforsøget. Der er således ingen tvivl om at der også i laboratorieforsøget har været kondens i varmeveksleren, og dette giver endnu et godt billede af fugtens meget store betydning for forholdene i veksleren. Årsagen til at der ikke er observeret kondens ved laboratiemålingen er formentlig, at der har været tale om små mængder, som ikke har afstedkommet opsamlet vand.

Målingen viser endvidere at der ikke er en jævn temperaturstigning igennem varmeveksleren, hvilket der tydeligvis er i beregningsmodellen. Der er mange forskellige mulige forklaringer på denne forskel i temperaturforløbet. For det første kan tidligere omtalte problemer med at aflæse en "ren" lufttemperatur inde i varmeveksleren, være særdeles vanskeligt, idet det er svært at undgå at temperaturfølerne kommer i kontakt med pladematerialet, og dermed måles en temperatur som ikke svarer til luftens temperatur. En anden faktor kan være de ikke-ideelle forhold som naturligvis vil forekomme i veksleren under laboratorieforsøget, svarende til at varmeoverføringskoefficienten ikke er konstant gennem veksleren. Som anført tidligere er det i modellen antaget at varmeoverføringskoefficienten reduceres til en fjerdedel omkring ind- og udløb af varmeveksleren, idet der dels bliver tale om krydsvarmeveksling og dels at kontaktfladen mellem de to luftstrømme halveres. Hvis ikke denne antagelse er i overensstemmelse med de faktiske forhold i laboratiemålingen, vil det naturligvis have betydning for sammenligningerne.

Alt i alt afspejler disse forhold dels vanskelighederne med at gennemføre kontrollerede målinger af temperaturforholdene inden i varmeveksleren og dels at der er behov for at have mere kontrol over hvordan varmevekslerens varmeoverføringskoefficient varierer gennem veksleren. Derfor vil det i forbindelse med videreudviklingen af beregningsmodellen være af afgørende betydning, at de fremtidige målinger der gennemføres i forbindelse med verificeringen af modellen er meget mere kontrollerede.

4.7 Dynamiske simuleringer af varmevekslere

De laboratiemålinger der er lavet for den nyudviklede varmeveksler, beskrevet i afsnit 3.3.5, viser, at der under de givne forhold forekommer isdannelse i veksleren, og at der dermed er behov for at implementere den kontinuerlige afrimning af veksleren for at undgå nedbrud. Hvis man benytter den dynamiske model til at simulere den pågældende situation, viser simuleringen imidlertid at der ikke vil forekomme isdannelse i veksleren, og derfor kan det pågældende laboratorieforsøg ikke benyttes til validering af modellens dynamiske egenskaber.

Årsagen til at der i laboratiemålingen forekommer isdannelse mens der i modellen ikke gør er, at modellen regner på idealiserede forhold, hvor f.eks. luftfordelingen i varmeveksleren antages at være helt jævn. I laboratiemålingen vil luftfordelingen ikke være jævn, og derfor vil der kunne forekomme isdannelse i områder med ikke-ideel luftfordeling (hvor den varme luftstrøm er reduceret i forhold til den kolde), og når først isdannelsen begynder, vil den kunne sprede sig til resten af varmeveksleren efterhånden som luftgennemstrømningen reduceres som følge af isopbygningen.

For således at kunne validere de dynamiske egenskaber for modellen, er det nødvendigt at gennemføre laboratoriemålinger hvor der er bedre kontrol over temperatur- og fugtighedsmålinger, så man bla. kan observere eventuelle lokale fænomener i veksleren. Laboratorieforsøg som kan benyttes til dette formål henføres til det videre arbejde (se afsnit 4.9). Herudover vil det ligeledes være nødvendigt at overveje hvorledes indflydelsen af lokale forhold i givne varmevekslerudformninger kan modelleres, således at modellen i højere grad kan benyttes til at vurdere konkrete styringsstrategier og risici for tilisning mv.

Den dynamiske model benyttes nu til at gennemføre en række simuleringer, med det formål at undersøge hvorvidt den dynamiske, kontinuerlige afrimning af en varmeveksler kan blive problematisk under specielle omstændigheder. F.eks. kunne man forestille sig, at man ved lave udetemperaturer (under 10 frostgrader) og samtidig lav intern fugtproduktion, vil kunne opleve at man ikke kan styre sig udenom en total tilfrysning af varmeveksleren.

I forbindelse med simuleringerne benyttes en teoretisk varmeveksler baseret på den tidligere omtalte kassevarmeveksler. For at give et bedre overblik over forholdene i veksleren er den opdelt i 10 lige store kontrolvolumener, og varmeoverføringskoefficienten er 40 W/K for hver enkelt kontrolvolumen og ens i hhv. de kolde og varme kanaler. Volumen af luft i hvert enkelt kontrolvolumen er fastsat som $4 \cdot 10^{-3} \text{ m}^3$ (i både den varme og kolde side) og volumen af plademateriale er sat til $14 \cdot 10^{-4} \text{ m}^3$. Alt i alt har denne teoretiske varmeveksler varmeoverføringskoefficienter og volumen svarende til førømtalte kassevarmeveksler.

I samtlige simuleringer benyttes en cyklus, hvor der først i en time anvendes 90 % flow i den varme kanal og 100 % flow i den kolde kanal og derefter i en time anvendes 10 % flow i den varme kanal og 0 % flow i den kolde kanal. Dette princip svarer til hvad der blev benyttet i laboratorieforsøget.

4.7.1 Simulering af kontinuerlig afrimning under ”normale” forhold

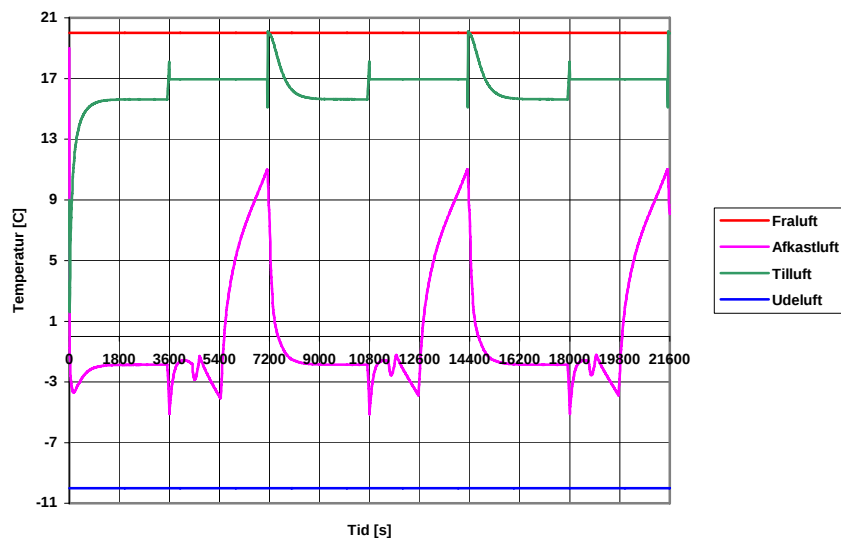
Først gennemføres en simulering af en situation svarende til den som er beskrevet under laboratoriemålingerne, dog med den forskel, at udetemperaturen nu er lavere, således at isdannelse forekommer i modellen. Simuleringen har til hensigt at vise hvorledes det kontinuerlige afrimningsprincip fungerer, og er således en parallel til den tilsvarende laboratoriemåling.

På nuværende tidspunkt er simuleringer relativt tidskrævende, og derfor gennemføres kun en ganske kort række cykler af afrimning. Dette påvirker dog ikke de resultater som fremkommer, idet det af simuleringerne vil fremgå, at der ret hurtigt indstiller sig en periodestabilitet i beregningerne.

Input til beregningen er følgende karakteristika for til- og fraluften:

Fraluft :	T = 20 C
	x = 0,0045 kg/kg (svarende til ca. 30 % RF)
Tilluft:	T = -10 C

Der simuleres 6 timer, og i figur 4.13 er vist temperaturen for fraluft, tilluft, afkastluft og udeluft.

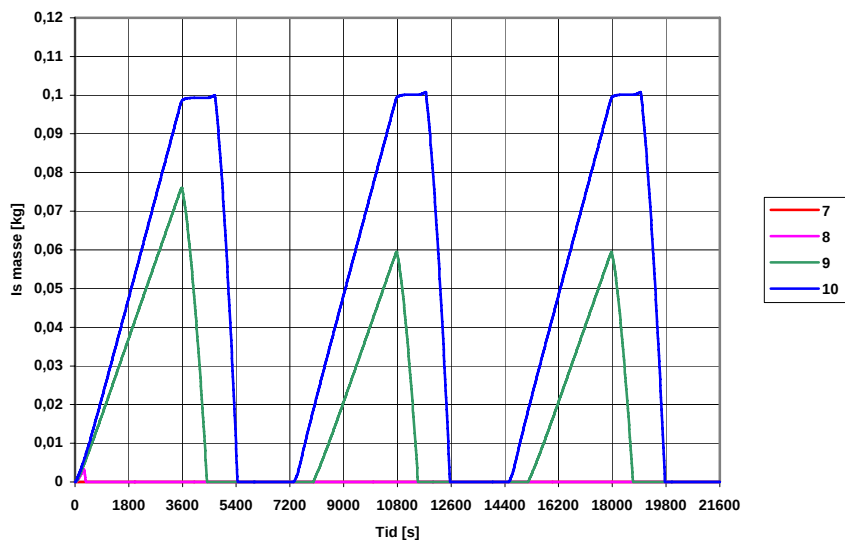


Figur 4.13 Temperaturforløb af fraluft, tilluft, afkastluft og udeluft i simuleringen.

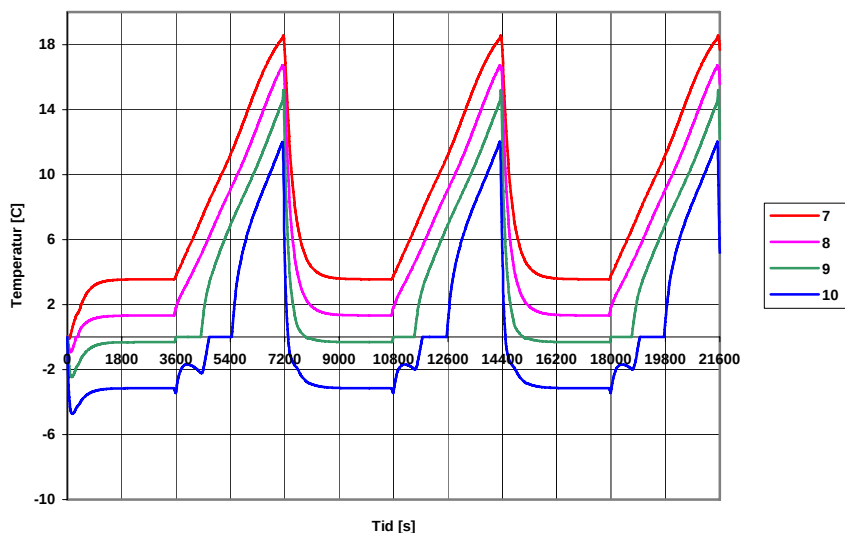
Af figur 4.13 kan det ses, at mens den pågældende varmvekslersektion er aktiv har tilluften en temperatur på ca. 15,5 °C og mens den er passiv ca. 17,0 °C. Figuren viser også at periodestabilitet indtræder allerede efter første cyklus, og det er dermed klart at det ikke er nødvendigt at fortsætte simuleringen længere end de 6 timer, svarende til at der medtages to stabile cykler.

Af figuren kan det også ses, at der i forbindelse med sektionsskift, dvs. ved 3600 s, 7200 s, 10800 s osv. forekommer spring i temperaturerne, f.eks. falder afkastluftens temperatur nærmest momentant fra -1,7 °C til -5,0 °C. Dette forhold skyldes et øjebliks instabilitet i beregningen, hvor flowet i den kolde side af veksleren i løbet af meget kort tid skifter fra den maksimale værdi til 0. Det er ikke muligt helt at undgå disse temperaturspring, men i videreudviklingen af modellen vil det være interessant at bearbejde dette problem, så springene minimeres.

I den pågældende simulering opbygges der frost i varmeveksleren, og i figur 4.14 er vist frostens masse for de sidste 4 kontrolvolumener (kontrolvolumen 7, 8, 9 og 10) inden udløbet. I figur 4.15 er vist pladetemperaturen for de sidste 4 kontrolvolumener inden udløbet.



Figur 4.14 Is masse i de sidste 4 kontrolvolumener i varmeveksleren.



Figur 4.15 Temperatur af plademateriale i de sidste 4 kontrolvolumener i varmeveksleren.

Af figur 4.14 kan det ses at efter periodestabiliteten er indtrådt, forekommer der udelukkende isdannelse i de to sidste kontrolvolumener, mens der i den første cyklus forekommer isdannelse i alle 4 kontrolvolumener. Endvidere kan det ses at der i kontrolvolumen 9 forekommer væsentligt mere isdannelse i første cyklus end i 2. og 3. cyklus. Dette skyldes naturligvis modellens startbetingelser, og derfor er det vigtigt at benytte resultaterne fra 2. eller 3. cyklus hvor der er indtrådt periodestabilitet.

Figur 4.14 viser at der i kontrolvolumen 9 ophobes ca. 60 g is mens der i kontrolvolumen 10 ophobes ca. 100 g is, inden afrimningen påbegyndes. De 60 g henholdsvis 100 g svarer til hhv. ca. $6,0 \cdot 10^{-5} \text{ m}^3$ is og $10,0 \cdot 10^{-5} \text{ m}^3$ is. Hvis vi sammenligner disse størrelser med den volumen luft der er i hvert af de to kontrolvolumener, $4,0 \cdot 10^{-3} \text{ m}^3$, er det tydeligt at isen kun optager ca. 1,5 – 2,5 % af det samlede volumen, og dermed vil kanalen ikke blive tilstoppet under disse forhold. Isens tilste-

deværelse vil naturligvis påvirke luftstrømmene og varmeoverføringskoefficienterne, men på nuværende tidspunkt er disse, meget komplicerede forhold, ikke medtaget i den dynamiske model.

Figur 4.14 viser endvidere at den is som ophobes mens den pågældende sektion af varmeveksleren er aktiv, dvs. i løbet af 1 time, kan afrimes i løbet af ca. $\frac{1}{2}$ time. Dette forhold kan benyttes i forbindelse med en optimering af styringen for varmeveksleren, idet effektiviteten vil afhænge af hvor ofte der skiftes sektion i varmeveksleren, og det er således muligt at optimere varmevekslerens ydelse ud fra de temperatur og fugtforhold som er til stede på et givet tidspunkt.

Sammenholdes figur 4.14 og 4.15 vil man kunne se at pladetemperaturen fastholdes ved 0 °C mens isen smeltes, og at isen først smeltes i kontrolvolumen 9 og først når al isen er smeltet i kontrolvolumen 9 begynder isen i kontrolvolumen 10 at smelte. Herudover kan man også se, at når smeltning af isen i kontrolvolumen 9 påbegynder, stiger pladetemperaturen i kontrolvolumen 10 kraftigt, men når hurtigt et maksimum og falder derefter. Et fald i pladetemperaturen i denne situation vil ikke kunne forekomme under de givne omstændigheder, og det vil derfor være nødvendigt at kigge nærmere på de varmebalancer som er opstillet til at håndtere de pågældende forhold.

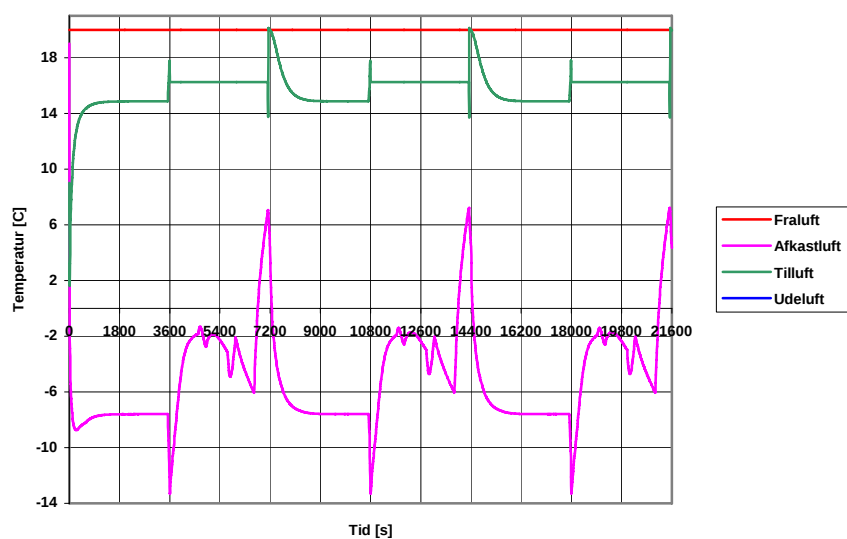
Simuleringen viser altså at princippet om en kontinuerlig afrimning af veksleren er mulig under de pågældende forhold. Man kan imidlertid overveje hvorvidt princippet kan benyttes under alle forhold, eller om der er specielle kombinationer af temperaturer og fugtindhold, som vil afstedkomme problemer som i sidste ende medfører at varmeveksleren fryser helt til. I det efterfølgende er gennemført 3 forskellige simuleringer som har til formål at vurdere dette, for disse situationer betragtes denne første simulering som referencesituation.

4.7.2 Simulering af kontinuerlig afrimning ved udetemperatur på -20 °C

Ved meget lave udetemperaturer vil temperaturerne i varmevekslerens kolde ende naturligvis også blive meget kolde, og derfor vil isdannelse forekomme i et større område. For at vurdere hvorvidt princippet om kontinuerlig afrimning fungerer under sådanne forhold er der gennemført en simulering hvor følgende forhold er benyttet:

Fraluft :	T = 20 C
	x = 0,0045 kg/kg (svarende til ca. 30 % RF)
Tilluft:	T = -20 C

Der simuleres igen 6 timer, og i figur 4.16 er vist temperaturen for fraluft, tilluft, afkastluft og udeluft.

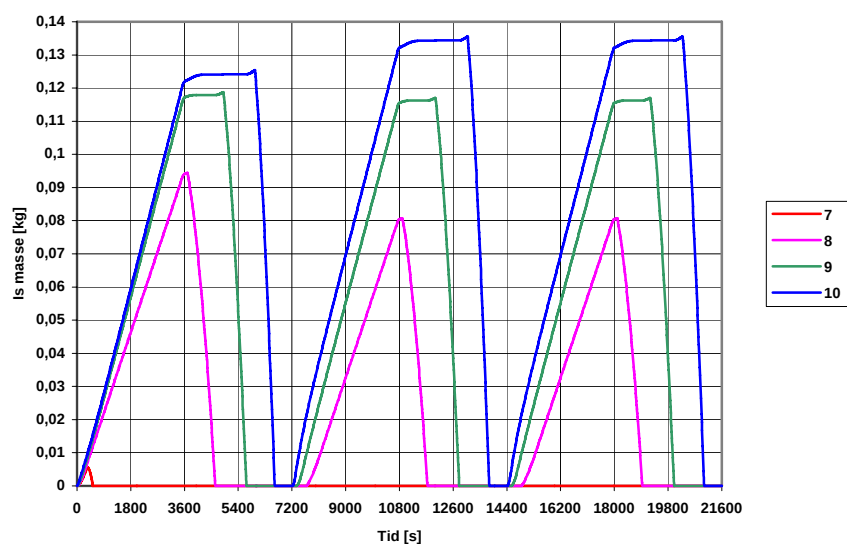


Figur 4.16 Temperaturforløb af fraluft, tilluft, afkastluft og udeluft i simulering.

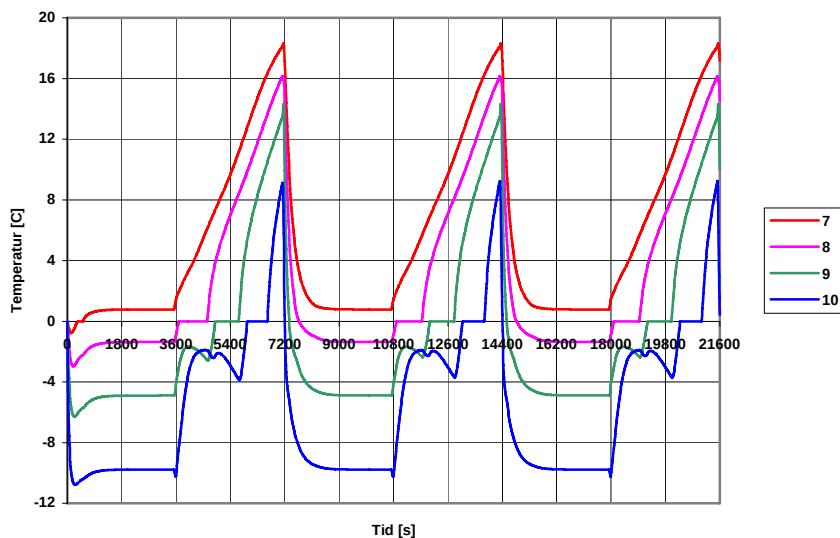
Af figur 4.16 kan det ses, at mens den pågældende varmvekslersektion er aktiv har tilluften en temperatur på ca. 14,9 °C og mens den er passiv ca. 16,3 °C. I forhold til referencesimuleringen er afkasttemperaturen altså kun faldet ca. 0,6 – 0,7 °C, selvom udetemperaturen er faldet 10 °C.

Igen viser figuren at der i forbindelse med sektionsskift, forekommer spring i temperaturerne, og for denne simulering er springene mere voldsomme end i referencesimuleringen.

I den pågældende simulering opbygges der mere frost i varmeveksleren end i referencesimuleringen, og i figur 4.17 er vist frostens masse for de sidste 4 kontrolvolumener inden udløbet. I figur 4.18 er vist pladetemperaturen for de sidste 4 kontrolvolumener inden udløbet.



Figur 4.17 Is masse i de sidste 4 kontrolvolumener i varmeveksleren.



Figur 4.18 Temperatur af plademateriale i de sidste 4 kontrolvolumener i varmeveksleren.

Af figur 4.17 kan det ses at efter periodestabiliteten er indtrådt, forekommer der isdannelse i de tre sidste kontrolvolumener. Figuren viser at der i kontrolvolumen 8 ophobes ca. 80 g is, i kontrolvolumen 9 ca. 115 g is mens der i kontrolvolumen 10 ophobes ca. 135 g is, inden afrimningen påbegyndes. Sammenligner man igen med det tilstedeværende luftvolumen, er der fortsat kun tale om meget små mængder is, og det er derfor ikke forventeligt at kanalen kan blive tilstoppet under disse forhold.

Figur 4.17 viser endvidere at den is som ophobes mens den pågældende sektion af varmeveksleren er aktiv, kan afrimes i løbet af ca. 52 minutter, og vi nærmer os således en situation hvor varmeveksleren ikke længere kan nå at afrime al isen, førend sektionsskifte igen foretages. Hvis afrimningen ikke er fuldendt efter en time, vil varmeveksleren fryse til i løbet af kort tid, og derfor kunne man overveje en styring af afrimningen, som på baggrund af udetemperatur, indetemperatur og fugtindhold i luften, sikrer at der er tid nok til at afrime al den ophobede is.

Af figur 4.18 ses det igen, at når smeltning af isen i kontrolvolumen 8 påbegynder, stiger pladetemperaturen i kontrolvolumen 9 og 10 kraftigt, men når hurtigt et maksimum og falder derefter. Samme fænomen forekommer efter at al isen i kontrolvolumen 8 er smeltet, og isen i kontrolvolumen 9 begynder at smelte.

Simuleringen viser altså at princippet om en kontinuerlig afrimning af veksleren er mulig selv når udetemperaturen er meget lav, men også at man ved endnu lavere temperaturer vil kunne forvente at varmeveksleren ikke vil kunne nå at afrime fuldstændigt indenfor en periode på én time, og at man derfor må medtage dette i en optimering af styringsstrategien for varmeveksleren.

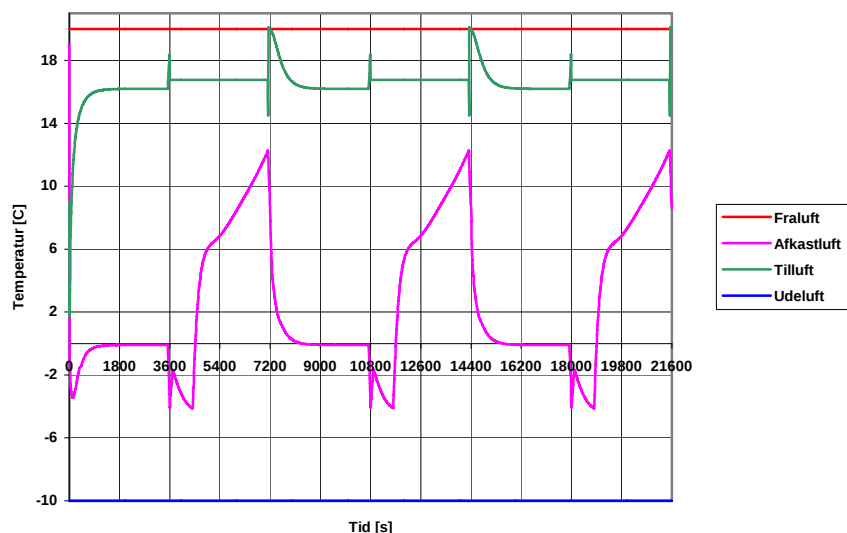
4.7.3 Simulering af kontinuerlig afrimning ved højere fugtindhold i fraluft

Når fugtindholdet i fraluften stiger medfører dette en øget kondens/isdannelse i varmeveksleren, til gengæld vil den øgede kondens medføre generelt højere temperaturer. Derfor er det interessant at

undersøge hvorledes et højere fugtindhold i fraluften påvirker den kontinuerlige afrimning af varmeveksleren. Der er derfor gennemført en simulering med følgende input:

Fraluft : $T = 20\text{ °C}$
 $x = 0,0060\text{ kg/kg}$ (svarende til ca. 40 % RF)
Tilluft: $T = -10\text{ °C}$

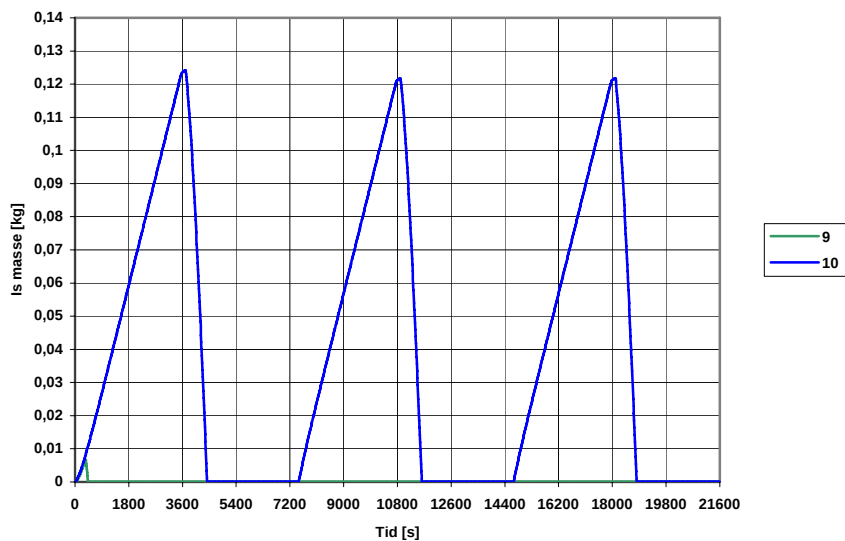
I figur 4.19 er vist temperaturen for fraluft, tilluft, afkastluft og udeluft.



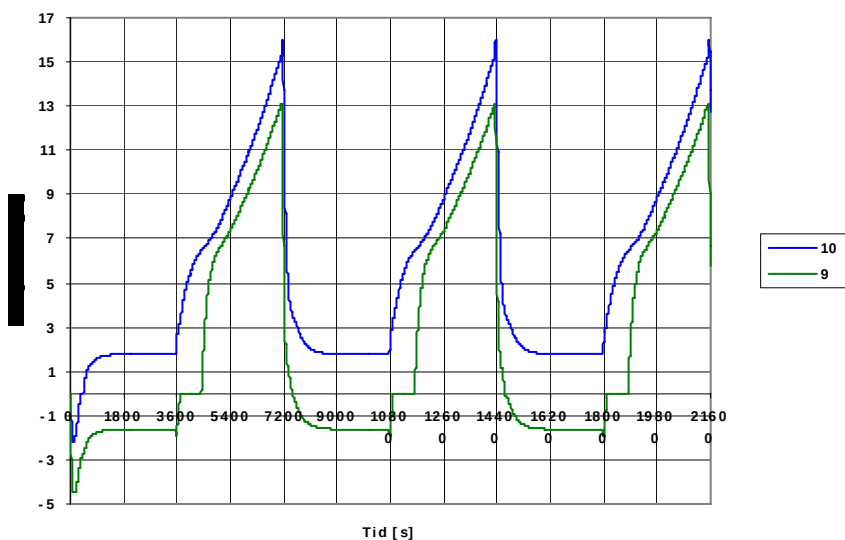
Figur 4.19 Temperaturforløb af fraluft, tilluft, afkastluft og udeluft i simulering.

Af figur 4.19 kan det ses, at mens den pågældende varmevekslersektion er aktiv har tilluften en temperatur på ca. $16,2\text{ °C}$ og mens den er passiv ca. $16,8\text{ °C}$. I forhold til referencesimuleringen er afkasttemperaturen altså steget ca. $0,7\text{ °C}$ i den aktive del af cyklen og faldet ca. $0,2\text{ °C}$ i den passive del af cyklen (afrimningsfasen). Det forøgede fugtindhold i fraluften resulterer således i en relativt beskeden stigning i temperaturen.

I figur 4.20 er vist frostens masse for det sidste 2 kontrolvolumener inden udløbet. I figur 4.21 er vist pladetemperaturen for de sidste 2 kontrolvolumener inden udløbet.



Figur 4.20 Is masse i de sidste 2 kontrolvolumener i varmeveksleren.



Figur 4.21 Temperatur af plademateriale i de sidste 2 kontrolvolumener i varmeveksleren.

Af figur 4.20 kan det ses at efter periodestabiliteten er indtrådt, forekommer der kun isdannelse i det sidste kontrolvolumen. Figuren viser at der i kontrolvolumen 10 ophobes ca. 125 g is, inden afrimningen påbegyndes, hvilket ikke afstedkommer risiko for tilstopning af kanalerne.

Figur 4.20 viser endvidere at den is som ophobes mens den pågældende sektion af varmeveksleren er aktiv, kan afrimes i løbet af ca. 15 minutter og der er altså i den givne situation mulighed for at køre med længere cykler.

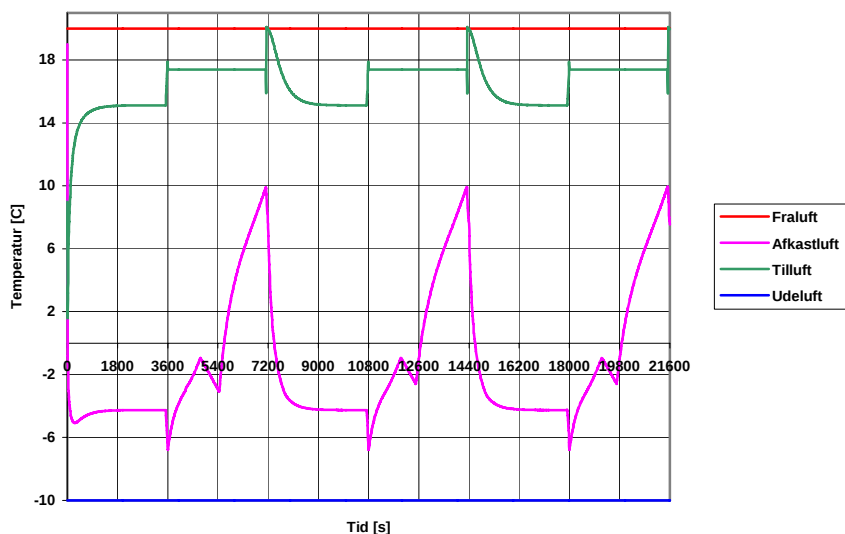
Simuleringen viser altså at en forøgelse af fugtindholdet i fraluften medfører en forøgelse af temperaturen i varmeveksleren, og dermed en totalt mindre mængde isdannelse.

4.7.4 Simulering af kontinuerlig afrimning ved lavere fugtindhold i fraluft

Hvis fugtindholdet i fraluften bliver lavere vil der naturligvis også forekomme mindre kondens i veksleren, og dermed vil temperaturerne generelt blive lavere. Dette vil naturligvis kunne medføre en forøget isdannelse, men da der samtidig er mindre fugt i luften, vil der også være en grænse for hvor meget der kan kondensere/fryse. For at vurdere hvorledes denne situation påvirker forholdene i varmeveksleren er der gennemført en simulering hvor følgende forhold er benyttet:

Fraluft : $T = 20\text{ °C}$
 $x = 0,0029\text{ kg/kg}$ (svarende til ca. 20 % RF)
Tilluft: $T = -20\text{ °C}$

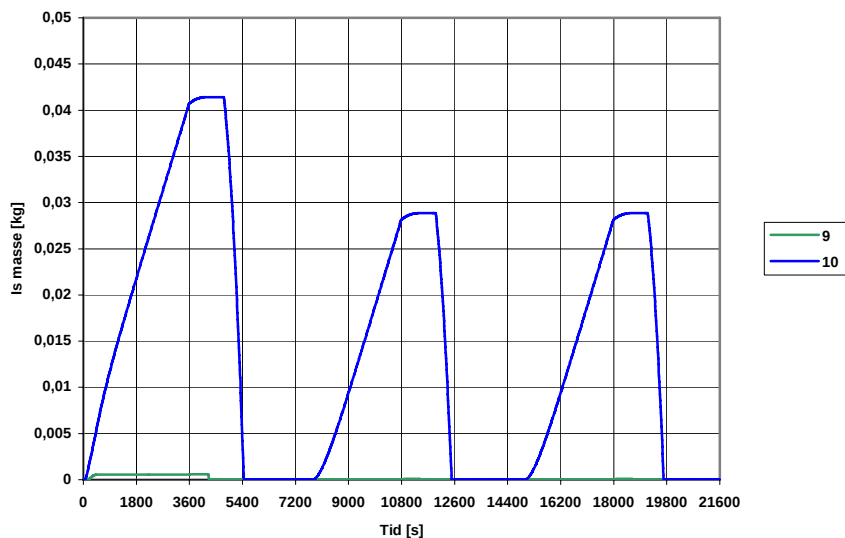
I figur 4.22 er vist temperaturen for fraluft, tilluft, afkastluft og udeluft.



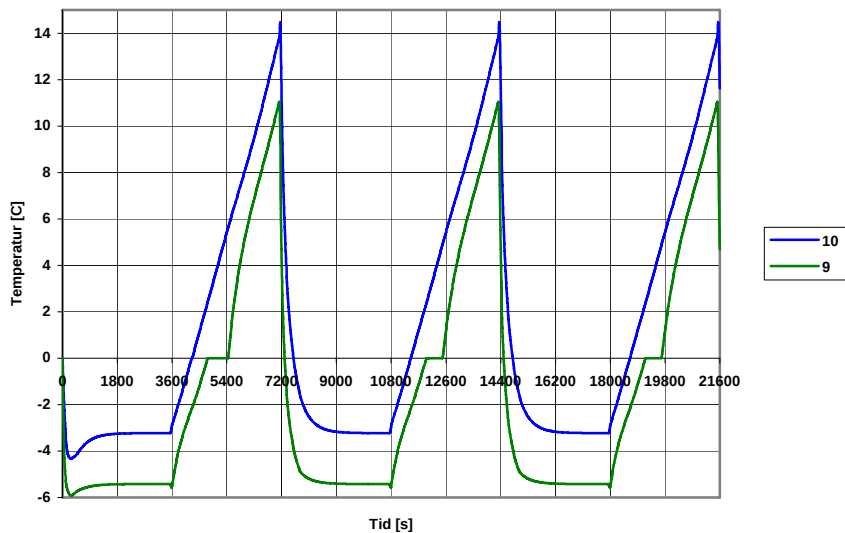
Figur 4.22 Temperaturforløb af fraluft, tilluft, afkastluft og udeluft i simulering.

Af figur 4.22 kan det ses, at mens den pågældende varmvekslersektion er aktiv har tilluften en temperatur på ca. 15,1 °C og mens den er passiv ca. 17,4 °C. I forhold til referencesimuleringen er afkasttemperaturen altså faldet ca. 0,4 °C i den aktive del af cyklen og steget ca. 0,4 °C i den passive del af cyklen (afrimningsfasen). Det forøgede fugtindhold i fraluften resulterer således i en relativt beskedent fald i temperaturen.

I figur 4.23 er vist massen af is for det sidste 2 kontrolvolumener inden udløbet. I figur 4.24 er vist pladetemperaturen for de sidste 2 kontrolvolumener inden udløbet.



Figur 4.23 Is masse i de sidste 4 kontrolvolumener i varmeveksleren.



Figur 4.24 Temperatur af plademateriale i de sidste 4 kontrolvolumener i varmeveksleren.

Af figur 4.23 kan det ses at efter periodestabiliteten er indtrådt, forekommer der udelukkende isdannelse i det sidste kontrolvolumen. Den samlede mængde is i det sidste kontrolvolumen er relativt beskeden, ca. 30 g.

Den is som ophobes i kontrolvolumen 10 afrimes i løbet af ca. 30 minutter.

Simuleringen viser altså at der ikke opstår nær så meget is når fugtindholdet af afkastluften er lavt sammenlignet med situationen hvor fugtindholdet er højere. Til gengæld viser simuleringen også, at på trods af at der nu kun er ca. 30 g is, hvor der i foregående simulering var ca. 125 g, faktisk skal bruges dobbelt så lang tid på at afrime varmeveksleren. Dette forhold viser, at man i forbindelse med lavt fugtindhold i afkastluften kombineret med meget kolde udetemperaturer, kan risikere at en total afrimning ikke kan gennemføres, og dermed at varmeveksleren kan fryse helt til.

4.8 Delkonklusion på dynamisk modellering af varmevekslere

Der er opbygget en beregningsmodel som kan simulere de dynamiske forhold i en varmeveksler. Sammenligninger af modellens resultater med resultater fra laboratoriemålinger har vist en god overensstemmelse, men også at der er behov for mere detaljerede og kontrollerede målinger for at opnå et bedre grundlag for yderligere verificering af modellen. Det er af sammenligningerne endvidere tydeligt at kondensen spiller en væsentlig rolle for varmeoverføringen i varmeveksleren, og at det er nødvendigt at medtage denne effekt i modellen for at opnå beregningsresultater som er sammenlignelige med målingerne.

Der er ligeledes foretaget en række dynamiske simuleringer på en teoretisk varmevekslerudformning som har vist, at princippet om kontinuerlig afrimning af en varmeveksler generelt fungerer godt. Simuleringerne antyder dog at der ved ekstremt lave temperaturer (under $-20\text{ }^{\circ}\text{C}$) og samtidig enten meget højt eller lavt fugtindhold i fraluften, vil kunne forekomme situationer hvor varmeveksleren ikke vil kunne afrimes i løbet af en afrimningsperiode på 1 time. I disse situationer vil det være nødvendigt at ændre styringsstrategi for veksleren, således at en total afrimning opnås indenfor afrimningsperioden.

De dynamiske simuleringer giver også et lille indblik i de muligheder der er for at analysere, karakterisere og optimere forskellige varmevekslere og deres styringsstrategier, og dermed hvilke muligheder der er opnået gennem udviklingen af den dynamiske beregningsmodel.

4.9 Det videre arbejde

Indenfor nærværende projekt er der udviklet en relativt simpel dynamisk model til simulering af varmeoverføringen i en modstrøms luft-luft pladevarmeveksler, hvor der eventuelt forekommer kondens eller isdannelse. Modellen indeholder en række simplificeringer som ønskes behandlet i forbindelse med en videreudvikling af modellen. I det følgende er kort gennemgået de interessante videreudviklingsmuligheder.

I modellen er kondens og isdannelse modelleret i en meget simplificeret formulering, hvor der blandt andet ikke tages hensyn til et eventuelt tryktab og en ændring af luftgennemstrømningen i veksleren, som vil forekomme ved både kondens og isdannelse, og der er endvidere ikke lavet en modellering af hvorledes isdannelse opstår/opbygges eller hvilke egenskaber isen får. Alle disse forhold skal bearbejdes i en videreudvikling af modellen, idet de vil være nødvendige at medtage i forbindelse med vurderinger af muligheden for afrimning og simuleringen af styringsstrategier mv.

I modellen er der ikke taget hensyn til ledning i pladematerialet (hverken aksialt eller på tværs). Dette forhold kan have betydning, specielt når der forekommer kondens eller isdannelse, som resulterer i at varmeoverførkoefficienten ændres væsentligt, og derfor vil ledning i pladematerialet skulle indføres i videreudviklingen af modellen. Årsagen til at den ikke er medtaget for nærværende er, at specielt den aksiale ledning kan skabe problemer med stabiliteten i de dynamiske beregninger,

når der i forbindelse med kondens/isdannelse forekommer relativt stor varmeledning i selve pladematerialet.

I den dynamiske model 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, så det primære behov er her at analysere fænomenet under kontrollerede laboratoriemålinger, og på denne baggrund fastlægge hvorledes man i praksis kan modellere forholdene.

5 Referenceliste

- [1] DSF 447:2004, Norm for mekaniske ventilationsanlæg, Code of practice for mechanical ventilation installations, 2004
- [2] Danvak ApS (1992) Varme- og Klimateknik, Grundbog. Teknisk Forlag A/S, København.
- [3] Mills, A. F. (1992) Heat Transfer. IRWIN.
- [4] Recair Sensitive, <http://www.recair.nl/>
- [5] Excel, Microsoft® Office Excel 2003 SP1,
- [6] Simulink
- [7] Matlab™ (2002), Version 6.5, The MathWorks, Inc., 3 Apple Hill Drive, Natick MA 01760-2098.
- [8] Weather test reference years of Greenland, Nordic Symposium on Building Physics, Reykjavik 13-15 June 2005, Kragh J. et al. (2005), Department of Civil Engineering, Technical University of Denmark

Bilag 1 Data for det benyttede forsøgsudstyr

Ventilatorer

Ventilatorerne er af mærket BESB 250-4-MGE fra firmaet EXHAUSTO A/S.

Datablad for ventilatoren findes på nedenstående web-adresse:

<http://www.exhausto.dk>

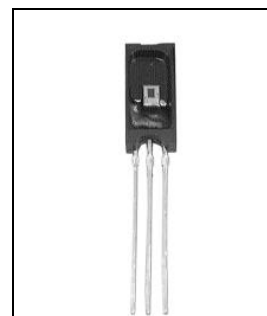


Relative fugtighedsfølere

De to relative fugtighedsfølere fra Honeywell er af typen HIH-3610 og har et måleområde fra 0-100 % RF med en usikkerhed på ± 2 % RH. Følerne kræver en spændingsforsyning på 5 V.

Datablad for føleren findes på nedenstående web-adresse:

http://content.honeywell.com/sensing/prodinfo/humiditymoisture/009012_2.pdf



Målekryds

De to målekryds er af typen MSD-125 (Ø125) fra producenten Halton. Måleområdet for målekrydsne er fra 25 l/s til 120 l/s.

Datablad for målekrydset findes på nedenstående web-adresse:

<http://www.halton.dk/>



Differenstryktransmittere

De to differenstryktransmittere er fra ENOTECH og er af typen DSP-1000P.

Typen har et måleområde fra 0 - 100 / 0 - 200 / 0 - 500 / 0 - 1000Pa afhængigt af jumper placering.

Datablad for differenstryktransmitteren findes på nedenstående web-adresse:

<http://www.enotech.dk/>



Bilag 2

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®, 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 (1st 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 ϵ -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.

ϵ -NTU – Effectiveness – Number of Transfer Units

The ϵ -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 ϵ -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 ϵ -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 ε -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 ε -NTU approaches have been combined. Below are a few examples of some of these studies.

Wang and Sundén (2003) developed a model for designing/optmizing 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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Mechanical ventilation systems with heat recovery in arctic climate

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ABSTRACT

Mechanical ventilations systems with highly effective heat recovery units in arctic climate have problems with condensing water from the extracted humid indoor air. If the condensing water freezes to ice in the heat recovery unit, the airflow rate will quickly diminish due to the increasing pressure drop.

Preheating the inlet air (outdoor air) to a temperature just above 0°C is typically used to solve the problem. To minimize the energy cost, a more efficient solution to the problem is therefore desirable. In this project a new design of a heat recovery unit has been developed to the low-energy house in Sisimiut, which is capable of continuously defrosting itself. The disadvantage of the unit is that it is quite big compared with other units. In this paper the new heat recovery unit is described and laboratory measurements are presented showing that the unit is capable of continuously defrosting itself. Nevertheless, extra heating is still necessary in very cold periods to avoid draft for occupants.

INTRODUCTION

Appropriate ventilation of houses in cold climates is of great importance. The low outside temperature and poor insulated building constructions increases the risks of condensation (on cold surfaces) of the humid room air. This can lead to rot and fungus damages and result in an unhealthy indoor climate.

Traditionally mechanical ventilation system with 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. The inlet diffusers are placed in rooms where the people are present over longer periods of time, i.e. the living room and bedrooms while 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. In this way the moisture and odour from cooking and bathing etc. are removed effectively without polluting the surrounding rooms, while 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 of the fans. In the design phase of a ventilation system, the attention must always be on minimizing the pressure losses in the system, as the electricity used by the fans this is directly proportional to the total pressure drop in the system.

Mechanical ventilation with heat recovery in cold climates can present problems with ice formation in the heat exchanger. That is, when the warm humid room air is brought in contact with the cold surfaces of the exchanger (cooled by the outside air), the moisture in the exhaust air condensates in the heat exchanger. If the outside air temperature is below zero,

the water vapour will freeze resulting in a larger air flow resistance on the exhaust side of the heat exchanger, which in turn decreases the air flow. The decrease in the amount of warm air through the exchanger will result in the exchanger being cooled further, and the system will eventually stop.

LABORATORY TEST OF A STANDARD HEAT RECOVERY UNIT

At the Technical University of Denmark 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 formation 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°C and warm exhaust room air at 21°C (relative humidity of air: 42%). As shown in Figure 1 the exhausted airflow after only 2 hours starts to fall due to the ice formation 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. For Nuuk the heat exchanger would use the built-in feature to avoid ice 30% of the year.

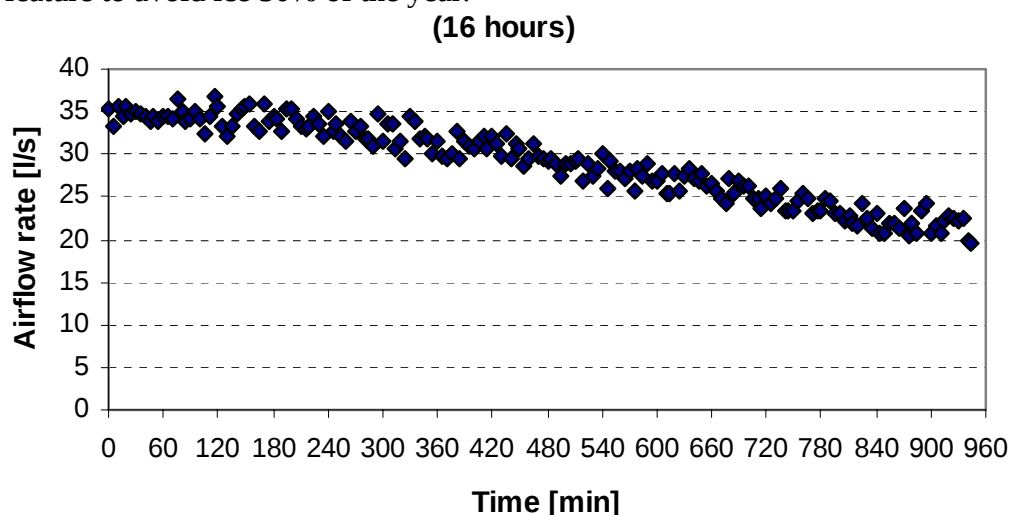


Figure 1 Measurements of the exhaust air flow rate through a standard heat recovery unit at low outside temperature (-5°C).

PREHEATING THE VENTILATION AIR

Preheating the cold inlet air before reaching the exchanger is often used to solve the problem. In Figure 2 an example of the energy used for preheating the outside air is shown for the climate in Sisimiut. The calculation assumed a reference house of 100 m^2 with a room height of 2.5 m and an air change rate of 0.5 h^{-1} according to the building code.

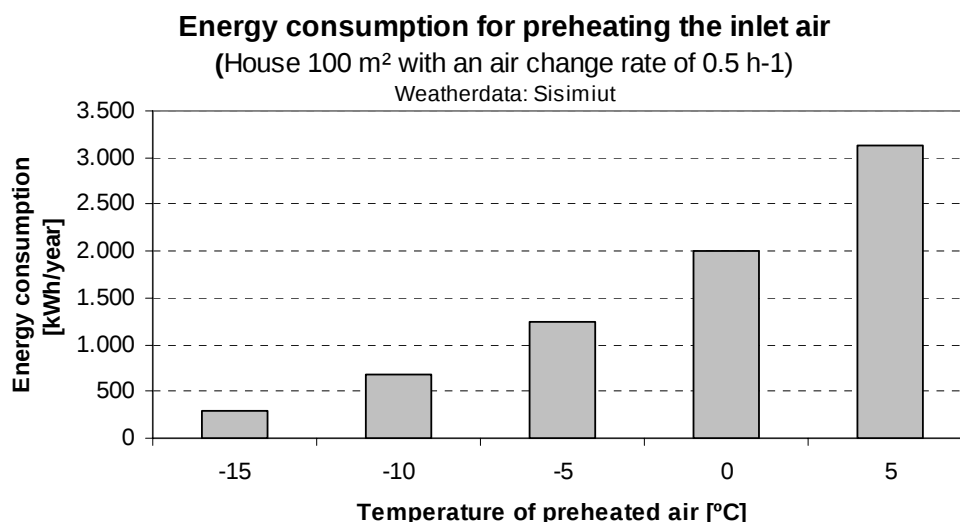


Figure 2 Energy consumption for preheating the ventilation air in a 100 m² house at an air change rate of 0.5⁻¹ according to the building code and a room height of 2.5 m.

As the preheating system requires extra energy and higher installation costs it is not the optimum solution.

THE VENTILATION HEAT LOSS

The ventilation heat loss in a typical 100 m² Greenlandic house located in Sisimiut is approximately 9.000 kWh/year (at an air change rate of 0.5 h⁻¹). In Table 1 the energy consumption for the ventilation system is compared for: ventilation system with no heat recovery, ventilation with heat recovery and preheating system and the future ventilation system with only heat recovery.

Table 1 The ventilation heat loss for a 100 m² house in Sisimiut.

	Ventilation heat loss	Preheating	Fan energy	Savings
	kWh	kWh	kWh	kWh
Ventilation with no heat recovery	9.000	0	0	0
Standard system with preheating	700	2000	600	5.700
Future system with no preheating	900	0	600	7.500

NEW DEVELOPED HEAT RECOVERY UNIT WITH FROST PROTECTION

A new design of a heat recovery unit was developed for the low energy house in Sisimiut in corporation between EXHAUSTO A/S and the Technical University of Denmark. The dimension of the unit is: Length 1.760 mm, width 930 mm and height 660 mm. The unit consists of two highly efficient aluminium counter flow heat exchangers coupled in a serial connection. One valve is able to switch the air flow direction through the units. When ice formation starts to reduce the airflow in the coldest exchanger the air flow direction is switched. The exchangers, valve and filters are mounted in a cabinet with 50 mm insulation, even though the unit is recommend to be placed in a heated place to minimize risks of frost damages from the condensing water. A diagram of the system is shown in Figure 3.

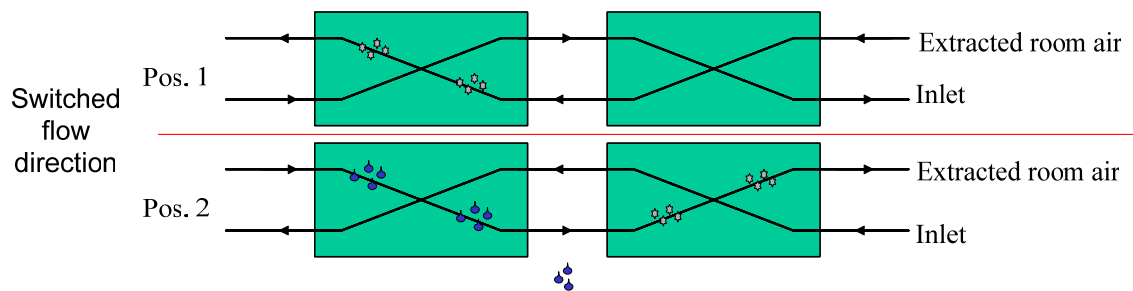


Figure 3 Diagram of the heat recovery unit with two heat exchangers. One valve and a timer switch the flow direction when ice formation reduces the air flow.

Two pictures of the heat recovery unit are shown in Figure 4. One valve and a timer switch the air flow direction, when ice formation reduces the flow. The condensing water is lead out through the bottom of the unit.



Figure 4 Pictures of the heat recovery unit.

LABORATORY TESTS

The heat recovery unit designed for the low energy house in Sisimiut was tested in laboratory facilities at the Technical University of Denmark. The test was carried out using an outside temperature of approximately -8°C and an inlet airflow rate of approximately $260 \text{ m}^3/\text{h}$. The tests showed that the defrosting system functioned as intended. In very cold periods, however, the system still needs extra heating of the inlet air to avoid draft caused by too low inlet temperature. The measurements are shown in Figure 5.

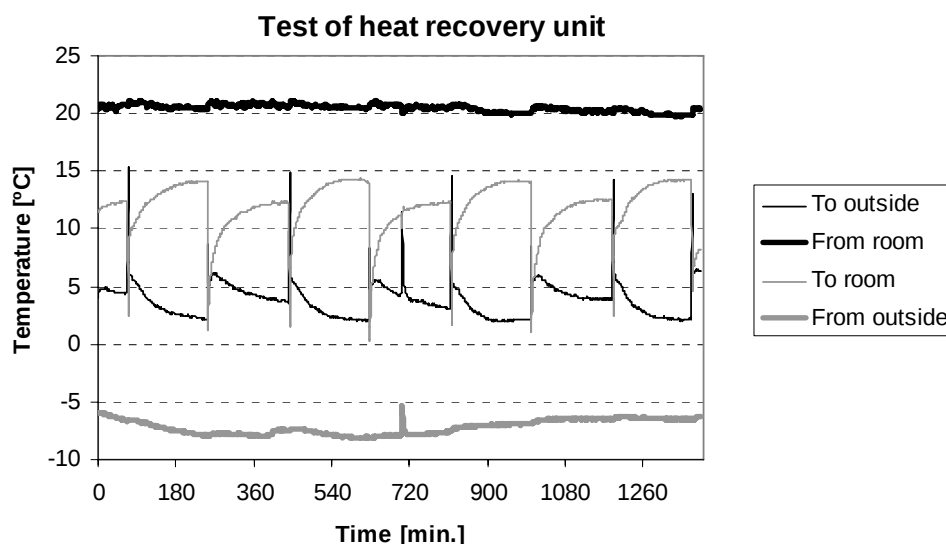


Figure 5 Laboratory measurement of the heat recovery unit.

As it can be seen from Figure 5 it is only necessary to heat the inlet temperature a few degrees before the inlet to the room.

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. Therefore 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 it self. The defrosting system was tested under laboratory facilities showing that the system worked as intended but still there was a need for extra heating minimizing the risk of draft for the occupants.

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Mechanical ventilation with heat recovery in cold climates

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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°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.

1. 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.

2. 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.

2.1 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.

2.2 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.

3. 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.

3.1 Ice formation in the heat exchanger

In cold climates there is a risk of ice formation inside 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.

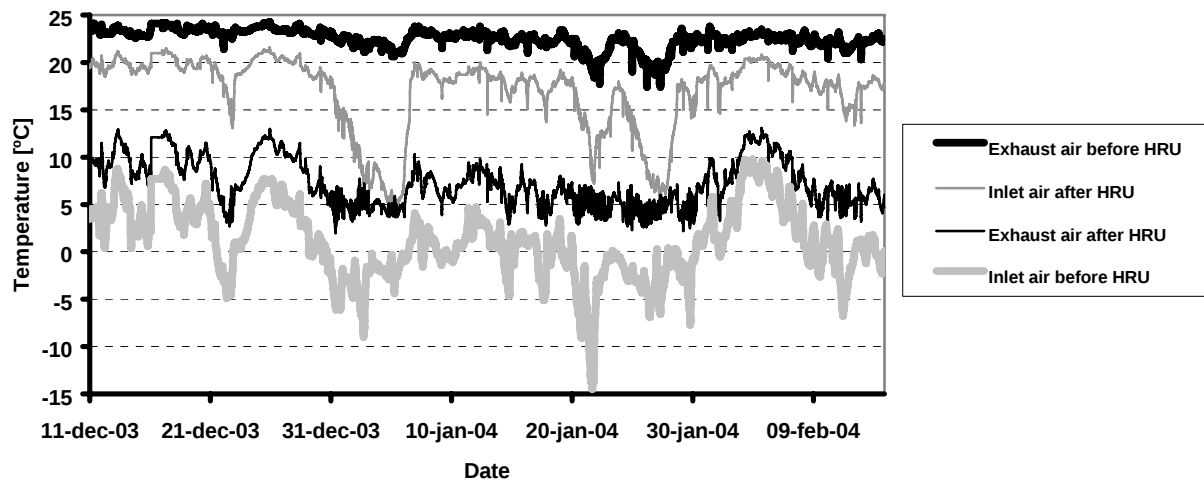


FIG. 1: Measured temperatures in heat recovery unit (HRU). Typical Danish winter.

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.

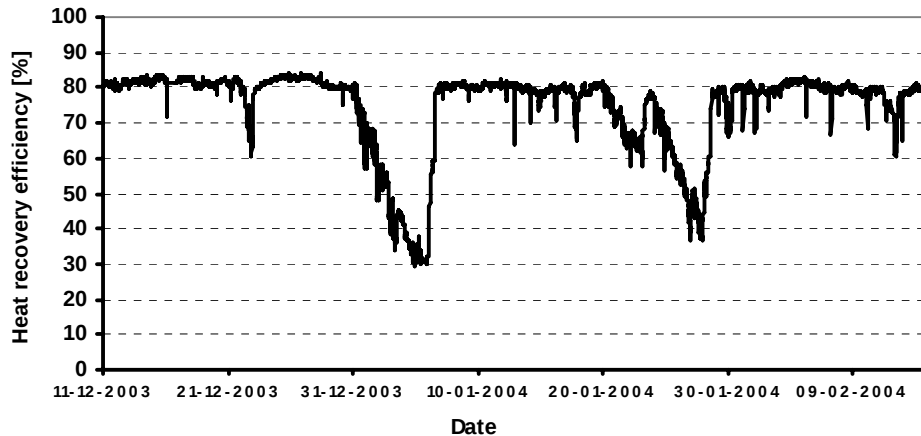


FIG. 2: Temperature efficiency of heat exchanger. Typical Danish winter.

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°C and warm exhaust room air at 21°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.

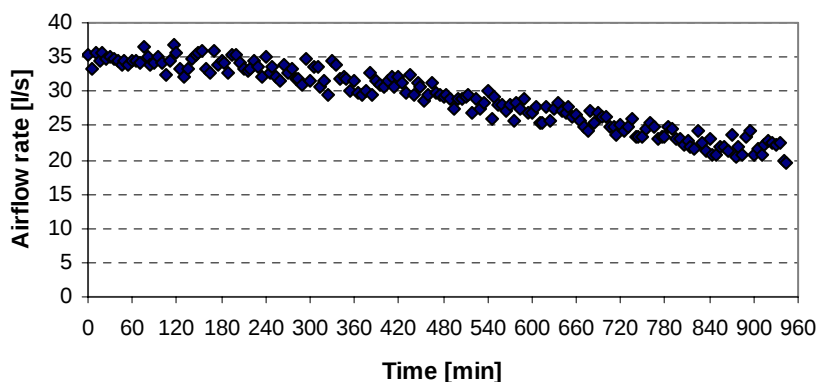


FIG. 3: Measurements of the exhaust air flow rate of a typical heat recovery unit at low inlet temperature (-5°C).

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°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°C . Therefore

development of new designs of heat recovery units able to function all year round in cold or very cold climates is desirable.

3.2 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°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°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°C and a heat recovery unit with a 90% temperature efficiency and balanced airflows, the minimum outside temperature can be determined as -30°C. In Denmark the outside temperature will almost never go below -30°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°C and -5°C respectively, and this would mean that there might be times when the inlet temperature is lower than the accepted 15°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).

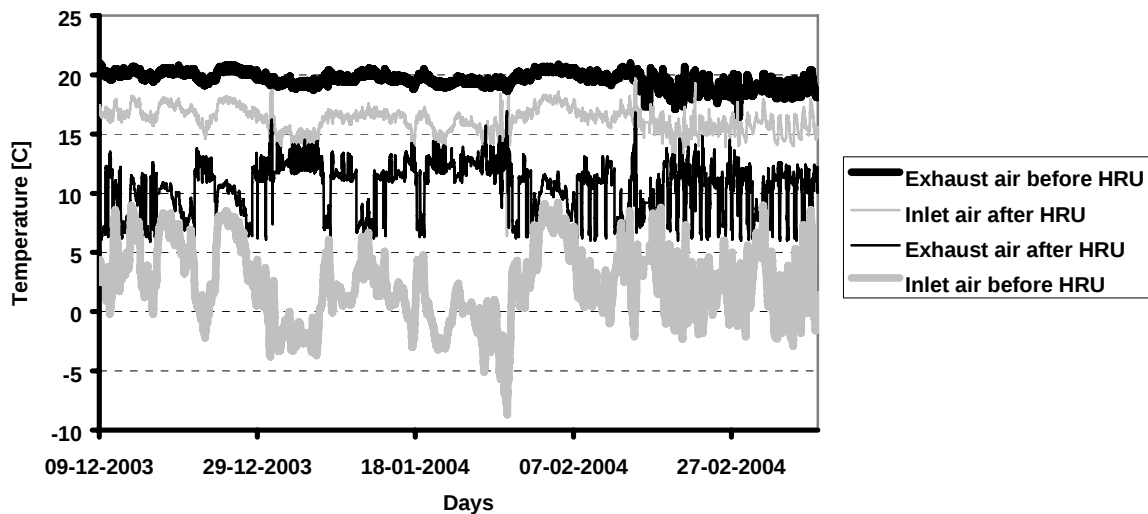


FIG. 4: Temperature measurements for heat recovery ventilation system.

From Fig. 4 it is evident that the inlet air temperature is typically between 15°C and 20°C. However, when the outside temperature reaches the freezing point, the inlet air goes below 15°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°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°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 m² house in Nuuk), and therefore new methods of avoiding low inlet temperatures should be investigated.

4. 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) Munters' solution

The well known Munters' 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. This

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.

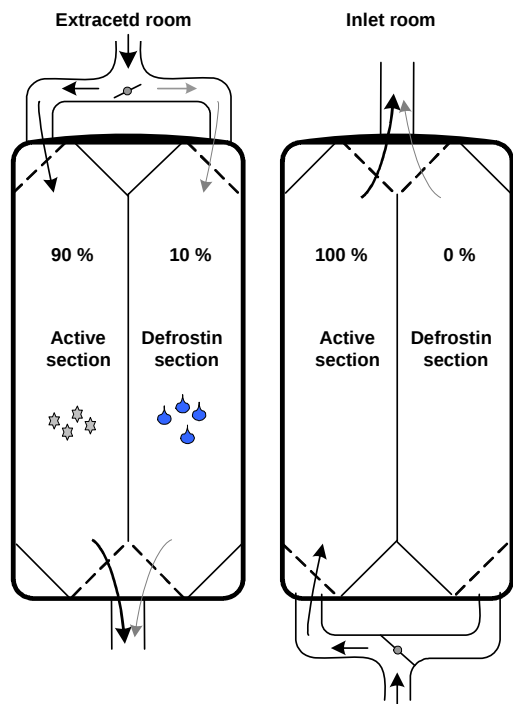


FIG. 5: Drawing of the exchanger at two sectional views. To the left is shown the exhaust airflow and to the right the inlet airflow.

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.

5. 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°C for a short period of time. Measurements also show that the inlet temperature at times reaches levels below 15°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.

6. Acknowledgement

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New counter flow heat exchanger designed for ventilation systems in cold climates

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Abstract

In cold climates mechanical ventilation systems with highly efficient heat recovery will experience problems with condensing water from the extracted humid indoor air. If the condensed water changes to ice in the heat exchanger, the airflow rate will quickly fall due to the increasing pressure drop.

Preheating the inlet air (outdoor air) to a temperature above 0°C before it enters the exchanger is one solution often used to solve the problem, however this method reduces the energy saving potential significantly. To minimize the energy cost a more efficient way to solve the freezing problem is therefore desirable. In this paper the construction and test measurements of a new counter flow heat exchanger designed for cold climates is presented. The developed heat exchanger is capable of continuously defrosting itself without using supplementary heating. Other advantages of the developed heat exchanger are low pressure loss, cheap materials and a simple construction. The disadvantage is that the exchanger is big compared with other heat exchangers. In this paper the new heat exchanger's efficiency is calculated theoretically and measured experimentally. The experiment shows that the heat exchanger is capable of continuously defrosting itself at outside air temperatures well below the freezing point while still maintaining a very high efficiency. Further analysis and development of a detailed simulation model of a counter flow air-to-air heat exchanger will be described in future articles.

Keywords: Ventilation, counter flow heat exchanger, heat recovery unit, defrosting, energy consumption and cold climates.

1. Introduction

In cold or arctic climates it is often difficult to get a traditional mechanical ventilation system to function as intended during the long winter. The ventilation heat loss without heat recovery is significant due to the large number of degree days and draft will also cause problems. If the outdoor air in cold regions is ventilated directly into the building through openings or diffusers, the occupants will normally feel draft because of the very low outside temperature. This is probably the reason why occupants often block the ventilation openings resulting in poor indoor climate and higher risk for moisture problems. Continuous moisture problems will typically cause damage to the building, e.g. as rot or mould fungus, which have been documented several times to be unhealthy for human beings.

If cold inlet air considerable below the freezing point is preheated using a heat exchanger, ice formation on the extracted airside will normally occur within a few hours, depending on the efficiency of the heat exchanger. In [1] and [2] the problem with ice formation in the heat exchanger for ventilation systems in cold climates is recognized,

but only the preheating of the supply air is discussed as a possible solution, and rejected due to the findings of [3] that suggest that this solution significantly reduces the recovered energy. In new buildings, preheating the inlet air above 0°C with heating panels often solves the ice and draft problem but, as mentioned earlier, this solution uses a lot of extra energy. The ice problem is not only seen in arctic climates, but also in places where the outside temperature for long periods stays a few degrees below the freezing point [1] and [4].

In this paper, the construction and test of a highly efficient counter flow heat exchanger capable of continuously defrosting itself without using supplementary heating is described. The measurements described in this paper are used to validate a detailed simulation model described in [5]

In this paper the airflows in the ventilation system are denoted as:

- Inlet air: Cold fresh air from outside ventilated into the building
- Extracted air: Warm humid indoor air extracted from the building

2. Description of the developed counter flow heat exchanger

In [6] measurements of several air handling units are presented, and it is concluded that the efficiency of the heat air handling unit mainly depends on leakages and the energy needed for the fans. In the design phase of the new heat exchanger, the following criteria were established:

- Continuous defrosting without supplementary heating
- Low pressure drop, i.e. low electricity consumption for fans
- Low risk of mixing the two air flows
- Simple construction (possibility for local construction/production)
- High efficiency (counter flow exchanger)
- Robust construction able to manage the rough arctic climate

Recognizing that the condensing water can not be avoided, the exchanger has to be designed in a way that easily removes the water without extra pumps or valves. By having an exchanger in which the airflows are vertical 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. From the bottom of the exchanger, the water is caught in a drain or simply drops to the ground if the exchanger is placed outside.

To achieve 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 heat exchanging surface of the exchanger is made of 5 mm double polycarbonate plates with a wall thickness of 0.5 mm. An example of the plate is shown in Figure 2.1.

The final exchanger was made of 10 plates separated by 4 mm's of air as shown in Figure 2.2. The cold inlet air flows in the ducts of the polycarbonate plate and the warm humid extracted air flows in the gap between the plates.

To minimize the influences from the surrounding environment, the exchanger is insulated with 100 mm polystyrene and covered with a 16 mm plywood board.

This design of the exchanger does not prevent the ice problem. Some of the condensing water will always change to ice in the lower part of the exchanger and over time reduce the extracted airflow rate. Therefore the exchanger consists of two

identical sections as shown in Figure 2.3 for cyclic defrosting of one of the sections.

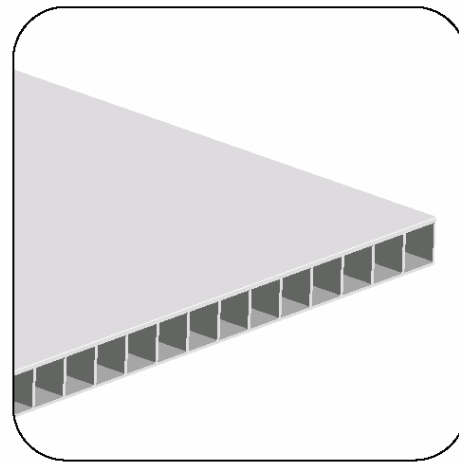


Figure 2.1 shows the 5 mm double wall polycarbonate plates

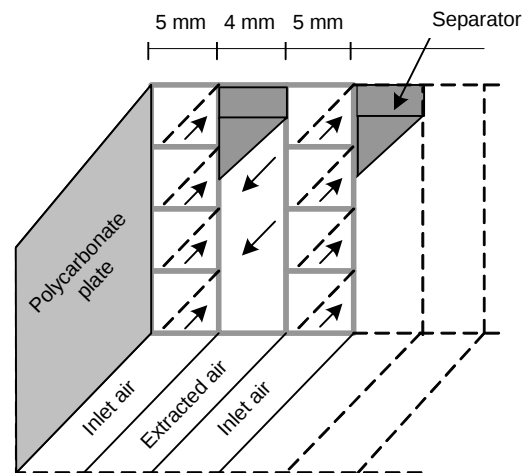


Figure 2.2 shows a sectional view (from the top) inside the exchanger

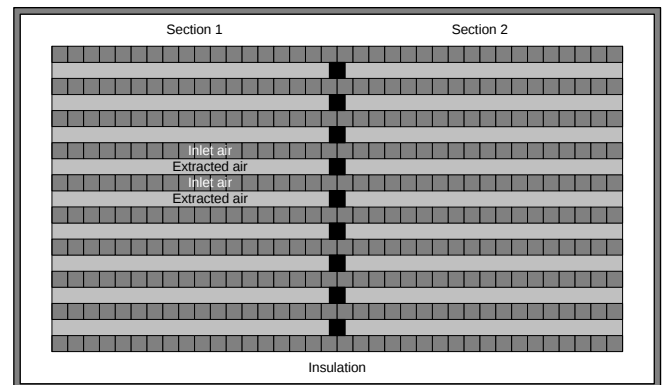


Figure 2.3 shows a horizontal cross section of the exchanger

This design ensures the possibility of defrosting by switching the airflows between the two sections. In that way one section is active and one section is passive regarding the heat exchange.

Two electrical valves control the airflows to the two sections. The flow rate of extracted room air is adjusted so the flow through the active and passive section is 90% and 10% respectively. After an adjusted time interval the airflows switch.

The inlet airflow also switches between the two sections but is always 100% or 0%. The idea is then that the 10% of airflow (warm extracted air) is able to defrost the passive part of the exchange.

In warm periods when no freezing occurs, both sections are used simultaneously, maximizing the heat transfer area and thereby the efficiency of the exchanger. In Figure 2.4 the exchanger's functionality is shown.

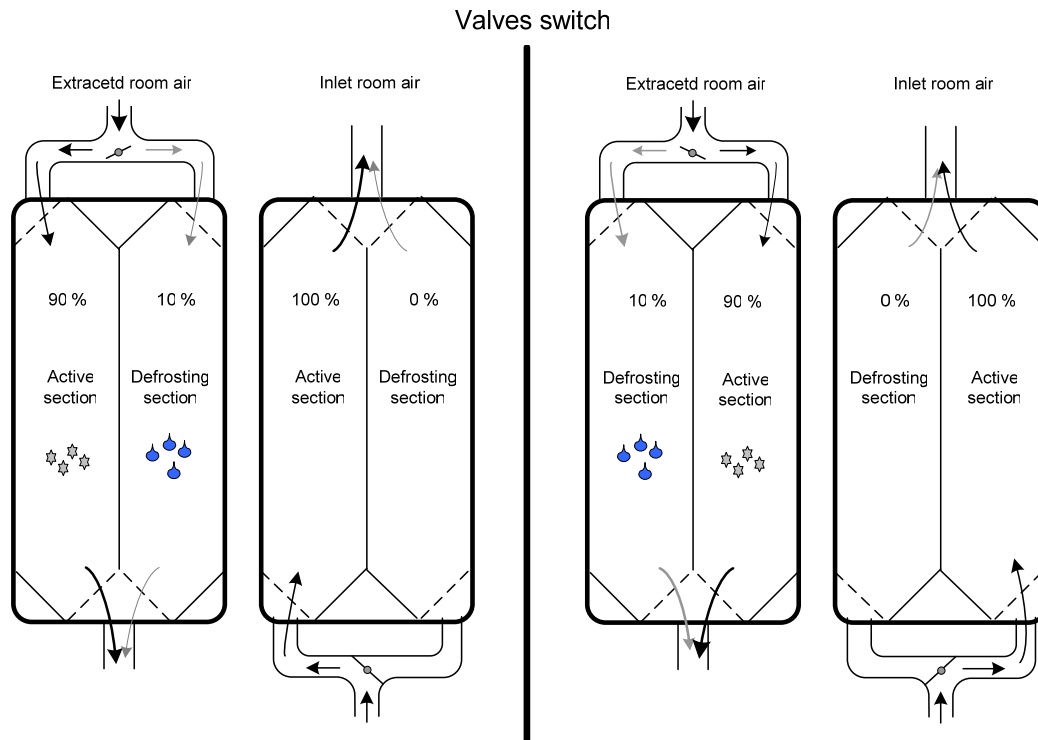


Figure 2.4 shows a drawing of the exchanger at two sectional views and with the valves switch (left and right of the centre line).

Table 2.1 Typical mean weather data measured in Nuuk, Greenland [10]. Enthalpy of the air at typical temperatures and humidity levels inside and outside.

Outside air		Heated	
Temperature	°C	-10	20
Relative humidity	%	80	10
Absolute humidity	kg/kg	0.0014	0.0014
Enthalpy	kJ/kg	-6.6	23.8
Difference		≈ 30 kJ/kg	
Inside air		Cooled	
Temperature	°C	20	-10
Relative humidity	%	30	100
Absolute humidity	kg/kg	0.0043	0.0018*
Enthalpy	kJ/kg	31.1	-5.7
Difference		≈ 37 kJ/kg	

*Condensation assumed and 100% relative humidity at -10°C.

Table 2.1 shows an example of typical weather data from Nuuk, Greenland [10]. The outside air is heated from -10°C to 20°C giving an increase of the enthalpy of 30 kJ/kg dry air. This is then compared with inside air cooled from 20°C to -10°C giving an enthalpy decrease of 37 kJ/kg dry air, which is approximately 19% more than the outside air was increased. Therefore the 10% reduction of the extracted air flow used to the defrosting should not be a problem.

3. Theoretical design of the exchanger

The exchanger was designed for a typical single-family house with a floor area of 100 - 140 m². Fulfilling the demands of the building code (Denmark [7] and Greenland [8]), the necessary air change rate should be 0.5 h⁻¹, which is approximately 150 m³/h (or 42 l/s). It could be discussed if the necessary air flow rate during winter

could be lowered, as the drying potential of the outside air is significantly better due to the low absolute humidity level compared with summer conditions.

The width of the exchanger (both sections not including the insulation) was 1.1 m. The 10 polycarbonate plates were mounted with an air space between each plate of 4 mm, giving a total depth of the exchanger of 86 mm. The height of the exchanger was determined by a calculation of the efficiency as a function of the height. Though this is obviously not the case, it is assumed that no condensation occurs, in order to simplify the calculations. This assumption will influence the efficiency in a conservative way and is therefore acceptable in the design phase. The condensation of water on the extracted airside will release more energy to the inlet airflow, resulting in a higher heat transfer and thereby efficiency in practise.

3.1 Heat transfer coefficient

In the following the heat transfer coefficient, H , between the two airflows is calculated. The cross sectional area (A_{cross}) of the inlet and extracted flow inside the exchanger is assumed to be equal. Given the section width ($w_s = \frac{1.1\text{m}}{2} = 0.55\text{m}$), and the depth (d) to be 0.086 m yields the cross sectional area of each flow:

$$A_{\text{cross}} = \frac{1}{2} \cdot w_s \cdot d \quad (3)$$

$$= \frac{1}{2} \cdot 0.55\text{m} \cdot 0.086\text{m} = 0.024\text{m}^2$$

(Cross sectional area of each flow in one section)

The velocity of the air flows is therefore:

$$v = \frac{q}{A_{\text{cross}}} = \frac{150\text{m}^3/\text{h}}{0.024\text{m}^2} \cdot \frac{1}{3600\text{s/h}} \approx 1.74\text{m/s} \quad (4)$$

Calculating the hydraulic diameter of the space between the plates and of the ducts in the plates, respectively, yields:

$$D_{h,\text{ducts}} = \frac{4 \cdot A_{\text{duct}}}{P} = \frac{4 \cdot 0.004\text{m} \cdot 0.004\text{m}}{2 \cdot (0.004\text{m} + 0.004\text{m})} = 0.004\text{m} \quad (5)$$

$$D_{h,\text{plates}} = \frac{4 \cdot A_{\text{plates}}}{P} = \frac{4 \cdot 0.55\text{m} \cdot 0.004\text{m}}{2 \cdot (0.55\text{m} + 0.004\text{m})} \approx 0.008\text{m}$$

Assuming a kinematic viscosity (ν) of the air to be $15 \cdot 10^{-6} \text{m}^2/\text{s}$ at 15°C (assumed mean temperature) the Reynolds number (Re) is calculated to be [9]:

$$Re_{\text{ducts}} = \frac{v \cdot D_h}{\nu} = \frac{1.74\text{m/s} \cdot 0.004\text{m}}{15 \cdot 10^{-6} \text{m}^2/\text{s}} \approx 464 \quad (6)$$

$< 2300 \Rightarrow$ Laminar

$$Re_{\text{plates}} = \frac{v \cdot D_h}{\nu} = \frac{1.74\text{m/s} \cdot 0.008\text{m}}{15 \cdot 10^{-6} \text{m}^2/\text{s}} \approx 928$$

$< 2300 \Rightarrow$ Laminar

As both flows are laminar the Nusselts' number is then according to [9] found to be 8.24 (laminar flow between two walls) and 3.6 (laminar flow in a square cross section). The thermal conductivity (λ) of the air is assumed to be 0.0254W/mK [9] which then yields a heat transfer coefficient, α , from the fluid (air) to the wall:

$$\alpha_1 = \frac{Nu_{\text{plates}} \cdot \lambda}{D_h} = \frac{8.24 \cdot 0.0254 \text{W/(m} \cdot \text{K)}}{0.008\text{m}} = 26.3 \text{W/m}^2\text{K} \quad (7)$$

(warm air flow)

$$\alpha_2 = \frac{Nu_{\text{square}} \cdot \lambda}{D_h} = \frac{3.6 \cdot 0.0254 \text{W/(m} \cdot \text{K)}}{0.004\text{m}} = 22.9 \text{W/m}^2\text{K}$$

(cold air flow)

The exchanger walls are made of 0.5 mm polycarbonate with a thermal conductivity (λ) of 0.20W/mK . The total heat transfer coefficient is then calculated as:

$$H = \left(\frac{1}{\alpha_1} + \frac{\delta}{\lambda} + \frac{1}{\alpha_2} \right)^{-1} \quad (8)$$

$$= \left(\frac{1}{26.3 \text{W/m}^2\text{K}} + \frac{0.0005\text{m}}{0.20 \text{W/mK}} + \frac{1}{22.9 \text{W/m}^2\text{K}} \right)^{-1}$$

$$\approx 11.9 \text{W/m}^2\text{K}$$

The influence of the internal fins in the cold inlet air flow is calculated using the simulation tool Therm [11]. The above described heat transfer coefficients are applied to the model. The results are presented in Table 3.1.

Table 3.1 Influence of the internal fins (inside the plates)

No fins	11.9	W/m ² K
With fins	13.6	W/m ² K

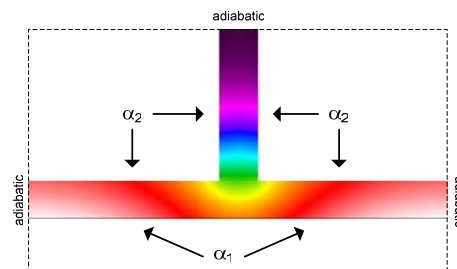


Figure 3.1 Simulation model of the fin

3.2 The efficiency of the heat exchanger by N_{TU} method in summer conditions

It is fairly extensive to calculate the efficiency of the heat exchanger at winter conditions due to the effects of the condensation and perhaps ice formation. The N_{TU} method assumes uniform heat transfer, which is unlikely when condensation starts to occur. The measurements at low temperature are therefore not comparable with the following calculation of the efficiency.

Calculating the capacity ratio (κ) of the two airflows at summer conditions (e.g. 10°C outside air temperature and 20°C inside air temperature) gives [9]:

$$C_{inlet} = \frac{150 \text{ m}^3/\text{h} \cdot \rho(20^\circ\text{C}) \cdot c_p(20^\circ\text{C})}{3600 \text{ s/h}} = 50.5 \text{ W/K}$$

$$C_{extracted} = \frac{150 \text{ m}^3/\text{h} \cdot \rho(10^\circ\text{C}) \cdot c_p(10^\circ\text{C})}{3600 \text{ s/h}} = 52.5 \text{ W/K}$$

Then $C_{inlet} = C_{min}$ and $C_{extracted} = C_{max}$

$$\kappa = \frac{C_{min}}{C_{max}} = \frac{50.5 \text{ W/K}}{52.5 \text{ W/K}} = 0.96 \quad (9)$$

The efficiency (ε) as a function of the exchanger's height (h) is then calculated by using the NTU method [9]:

$$N_{TU}(h) = \frac{H \cdot A(h)}{C_{min}} \quad (10)$$

$$\varepsilon = f(N_{TU}(h), \kappa) \quad (11)$$

$$= \frac{1 - \exp[-(1 - \kappa) \cdot N_{TU}(h)]}{1 - \kappa \cdot \exp[-(1 - \kappa) \cdot N_{TU}(h)]}, (\kappa \neq 1)$$

In Figure 3.2 equation (11) is shown as a function of the exchanger's height. It was decided that the height of the exchanger should be 1.8 meter due to practical limits (The height of 1.8 is not including inlet and outlet connections at the top and bottom of the exchanger). In Figure 3.2 the efficiency of the specific exchanger (1.8 m) is shown as a function of the air flow rate. The figure shows that the theoretical efficiency of the heat exchanger at 150 m³/h is approximately 91% using both sections simultaneously and 87% using one section only. Though as presented in Figure 3.3 the efficiency is very dependent on the actual flow rate.

Further analyses of this specific heat exchanger design are investigated and described in [5], where a

detailed simulation model is developed for calculating the complex heat transfer mechanisms that occur in a counter flow air-to-air heat exchanger with phase changes.

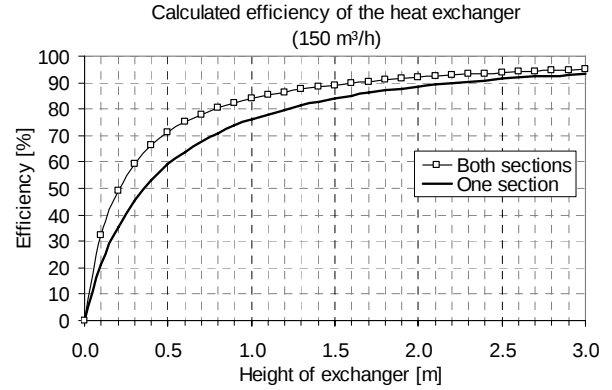


Figure 3.2 shows the calculated efficiency as a function of the height of the exchanger

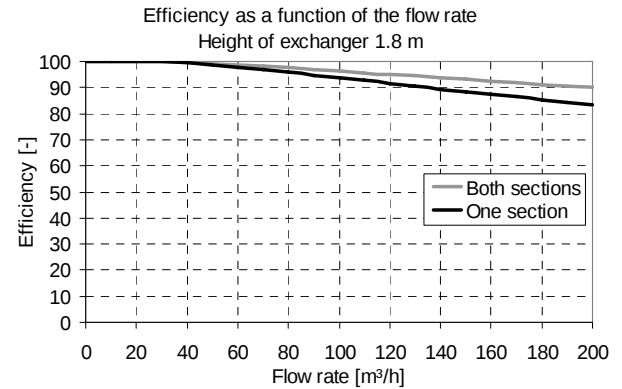


Figure 3.3 shows the efficiency as a function of the flow rate. Exchanger height of 1.8 m

4. Experimental test of the heat exchanger

The heat recovery unit described above was constructed and tested in laboratory facilities at the Technical University of Denmark. The primary purpose of the test was to investigate:

- Function of the cyclic defrosting method
- Efficiency of the defrosting

The test facility was able to supply an air temperature of approximately -10°C depending on the air flow rate. In Figure 4.1 a drawing of the test facility is shown. The temperatures of the two airflows were measured every minute before and after entering the exchanger. The temperatures were measured with thermocouple wires of type TT. Also, the two airflow rates and the relative humidity of the indoor air were measured. During the measurements the flows were adjusted to give a balanced air flow. The position of the thermocouples and measurement of the airflows are also shown in Figure 4.1.

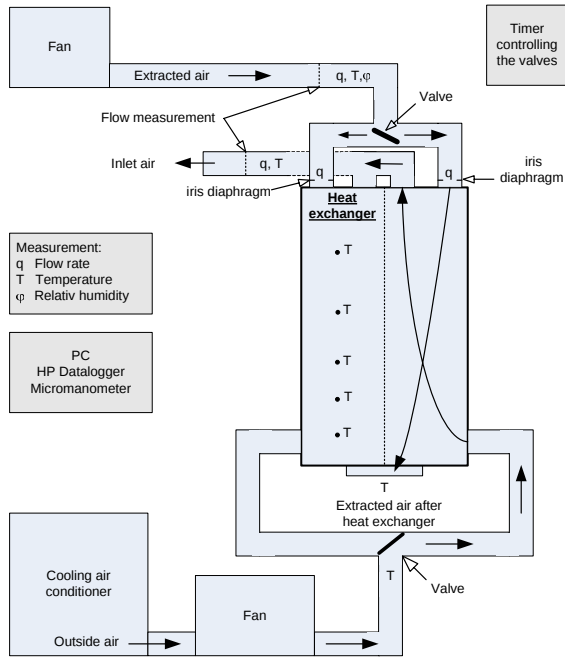


Figure 4.1 Schematic drawing of the test facility with position of temperatures (T) and flow rate (q) measurements

4.1 Measurement of dry efficiency using both sections simultaneously

The first test was carried out at an air flow rate of approximately 46 l/s, an outside temperature of 11.5°C and an indoor air temperature and relative humidity of 20.2°C and 53%, respectively. The valves were adjusted to achieve equal flows in both sections simultaneously. During the test no condensation was observed, which is in agreement with the theoretically calculated dew point temperature of 10.3°C. In Figure 4.2 the air flow rates are shown and in Figure 4.3 the temperatures measured before and after the heat exchanger are shown for the last 24 hours. It is seen that the inlet temperature is raised from 11.5°C to 18-19°C.

Using equation (12) [9] gives an efficiency of approximately 85%, as shown in Figure 4.4.

$$\varepsilon = \frac{C_{\text{inlet}} \cdot (T_{C,\text{out}} - T_{C,\text{in}})}{C_{\text{min}} \cdot (T_{H,\text{in}} - T_{C,\text{in}})} \quad (12)$$

where C is the thermal capacity of the flow, T is the temperature and index H and C refers to the hot and cold flow respectively. In the literature, the temperature efficiency is often used to evaluate the performance of a heat exchanger. The temperature efficiency is in the following defined as equation (13) and is also presented in Figure 4.4.

$$\eta = \frac{T_{C,\text{out}} - T_{C,\text{in}}}{T_{H,\text{in}} - T_{C,\text{in}}} \quad (13)$$

The efficiency of the heat exchanger calculated in chapter 3.2 (by the N_{TU} method) was carried out at a flow rate of 42 l/s (150 m³/h) and at slightly lower temperature levels. The measured efficiency is therefore not directly comparable to the calculated efficiency and therefore we recalculate at an air flow rate of 46 l/s and at the same temperature levels as the measurements. Doing this yields an efficiency of approximately 89%, which compared with the measurements (average 85%), seems in good agreement. The measured temperature efficiency is seen in average to be approximately 82%.

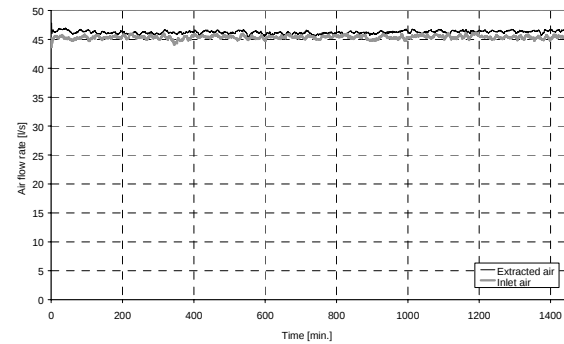


Figure 4.2 The measured air flow rate using both sections simultaneously

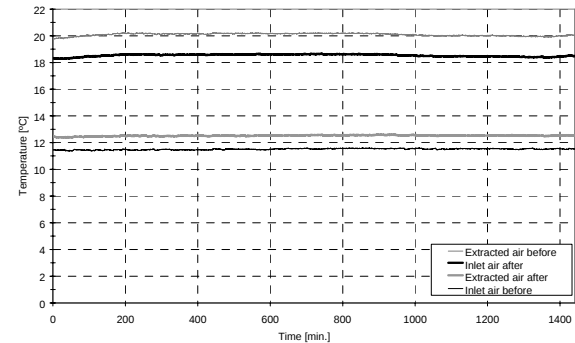


Figure 4.3 The measured temperatures using both sections simultaneously

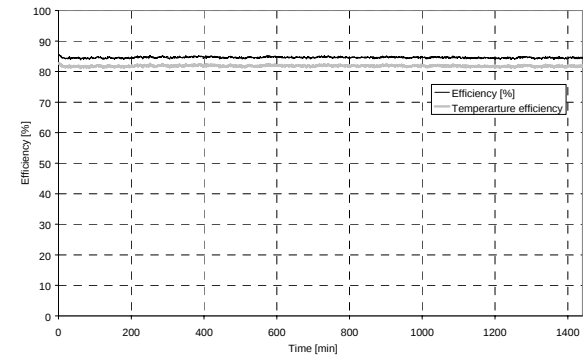


Figure 4.4 The measured efficiency and temperature efficiency using both sections simultaneously

4.2 Measurement of the efficiency with condensation in one section

The second test was carried out with only one section active and using an outside temperature of 8-9°C to obtain condensation inside the heat exchanger. The temperature and relative humidity of the indoor air was in average 22.5°C and 55 – 60% respectively, which results in a dew point temperature of approximately 13°C. The air flow rate was in average 49.5 l/s. During the 24 hour test period 1.275 kg of condensation (water) was caught, corresponding to approximately 50 grams pr. hour. In Figure 4.5 and Figure 4.7 the measured air flow rates and temperatures are shown, and in Figure 4.6 the corresponding efficiency and temperature efficiency are presented. Figure 4.8 shows the measured relative humidity of the extracted air during the test.

It is seen from Figure 4.5 and Figure 4.7 that the inlet temperature after the exchanger varies between 18-19°C giving an average efficiency of 78% and a temperature efficiency of 74%.

Again recalculating the efficiency as described in chapter 3.2 at 49.5 l/s and at the same temperature levels as the measurements gives an efficiency of approximately 80%, which then seems reliable compared with the measurements.

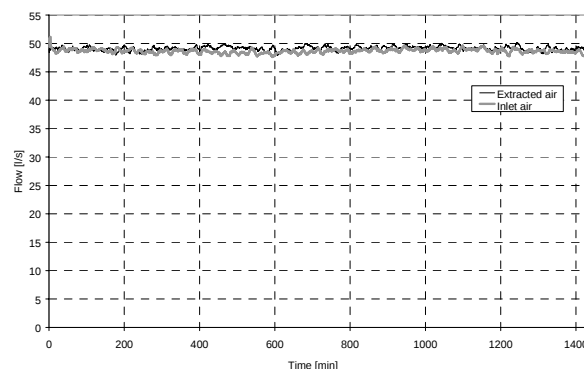


Figure 4.5 The measured air flow rates

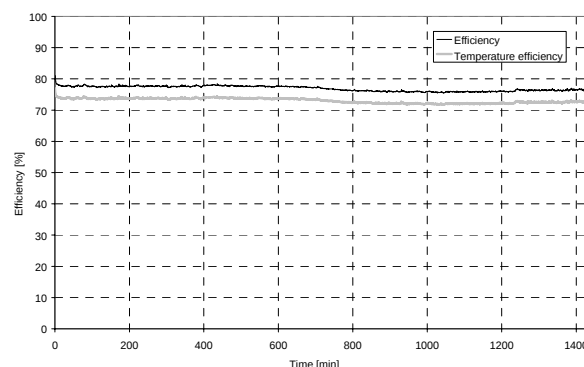


Figure 4.6 The measured efficiency and temperature efficiency

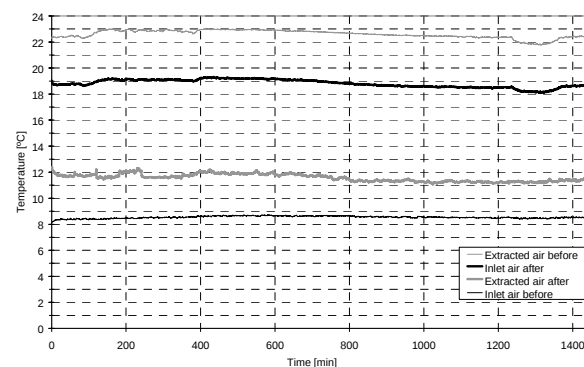


Figure 4.7 The measured temperatures

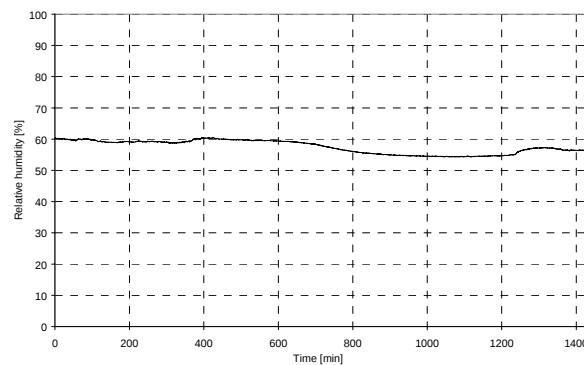


Figure 4.8 The relative humidity of the extracted air before the exchanger

4.3 Measurement with defrosting system activated

The same measurement was carried out at an average outside temperature of -7°C and at an air flow rate of $25 - 27 \text{ l/s}$. The low flow rate was necessary to maintain a low constant temperature, but as previously mentioned the demand of an air change rate of 0.5 h^{-1} could be lower during winter in cold climates. The time between the section switch was approximately 30 minutes. Figure 4.9 shows the total flow rate of the inlet air and extracted air during the test. The dynamic behaviour of flow is due to a small difference in the pressure drop of the two sections. Ideally the total flow should have been more or less constant. Figure 4.10 shows the temperatures measured in the ducts before and after the exchanger, and Figure 4.11 and Figure 4.12 shows the measured temperature efficiency and relative humidity of the extracted indoor air respectively.

It is seen from Figure 4.10 that the temperature of the inlet air drops to $15-16^{\circ}\text{C}$ at the end of each switch. The difference in the minimum temperature between each switch is again explained by the uneven flow rate in the two sections. It is also seen from Figure 4.11 that the temperature efficiency increases significant immediately after a section switch.

This increase can be explained by the energy stored as heat capacity in the inactive section during the defrosting period. In average the temperature efficiency is approximately 88%. This result is not compared to the previous efficiency calculations as the assumption of uniform heat transfer is not fulfilled at internal freezing conditions in the heat exchanger.

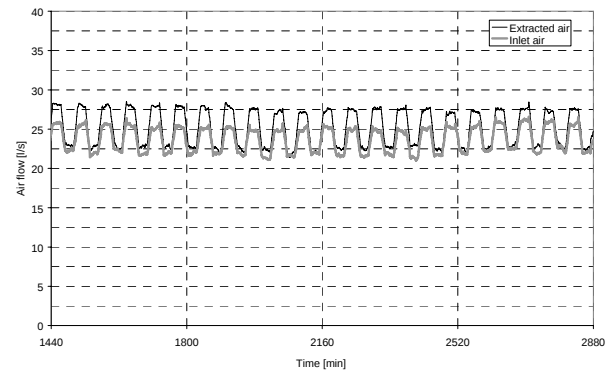


Figure 4.9 The airflow measurements

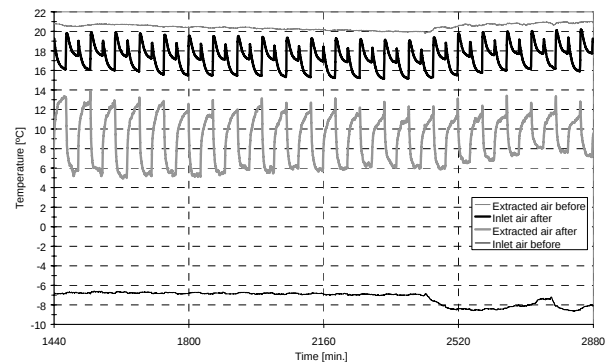


Figure 4.10 The temperature measurements

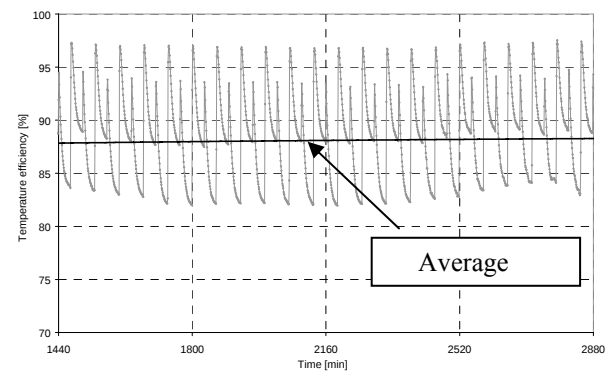


Figure 4.11 The measured temperature efficiency

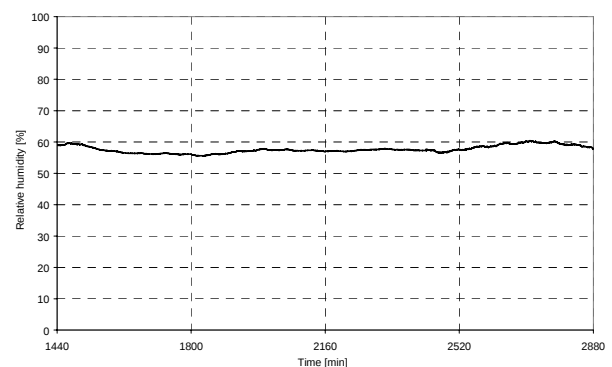


Figure 4.12 The relative humidity of the extracted air

4.4 Defrosting inactive

To document the freezing problem using high effective heat exchangers in cold climates, the same test was carried out with no switch between the sections and at a flow rate starting at 35 l/s. The measured flow rates are presented in Figure 4.13 and the measured temperatures in Figure 4.14.

As it is seen from Figure 4.13 the extracted air flow rate drops significantly, and after 23 hours the cold inlet flow is stopped manually to protect the heat exchanger. As expected, the test clearly shows the need for a defrosting method to protect the heat exchanger in cold climates. It is also seen that the defrosting time is above 200 minutes in this case. A solution where the ventilation is stopped until the exchanger is defrosted again is therefore assessed to be unsuitable.

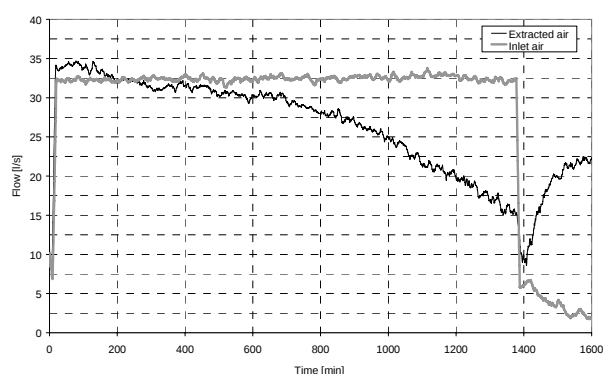


Figure 4.13 The measured air flow rates

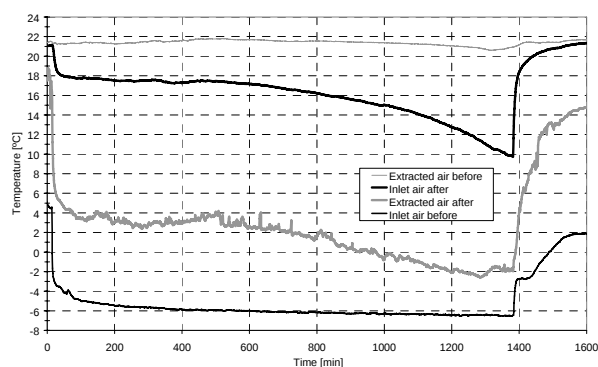


Figure 4.14 The temperature measurements

4.5 Long time defrosting test

A test of the defrosting system was carried out during a 4-day test period. The section switch time was set to 60 minutes, and the flow rate was adjusted to be approximately 25 l/s in average. In Figure 4.15 the measured air flow rates are shown and in Figure 4.16 the measured temperatures. A random zoom of the measurements is presented below in Figure 4.17 and Figure 4.18.

The long time test showed no sign of a defrosting problem at an inlet temperature to the exchanger of -6°C. Accumulation of ice that was not defrosted between the section switch would have influenced the flow during this test. It is also seen that the lowest inlet temperature in one period is 16°C, which could be a problem if the temperature drops even further at higher flow rates.

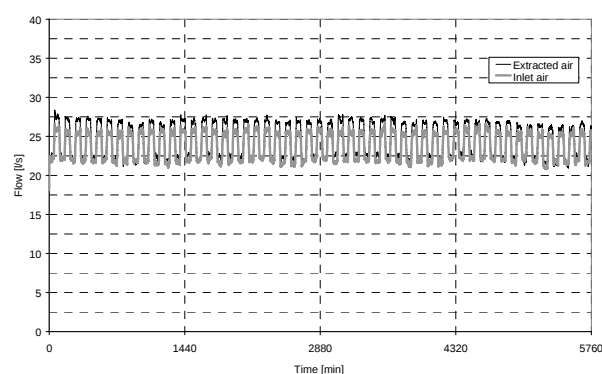


Figure 4.15 The air flow measurements

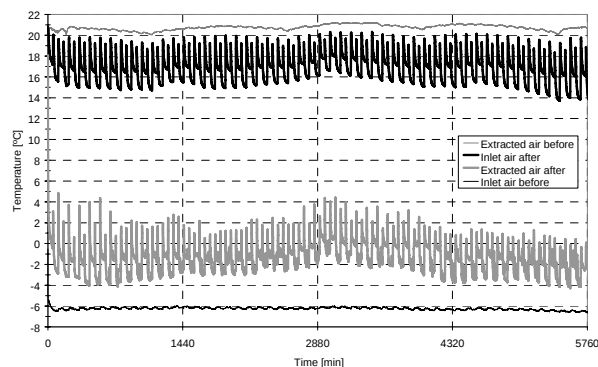


Figure 4.16 The temperature measurements

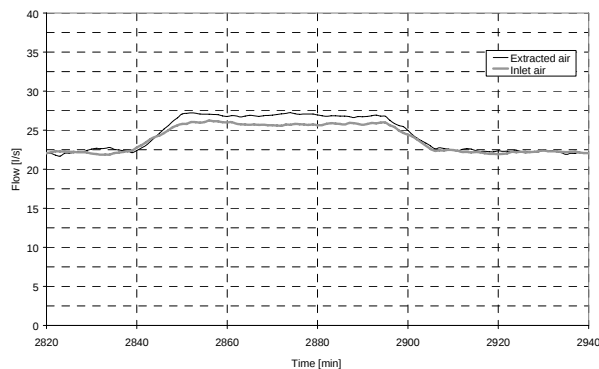


Figure 4.17 Zoom of the flow measurements

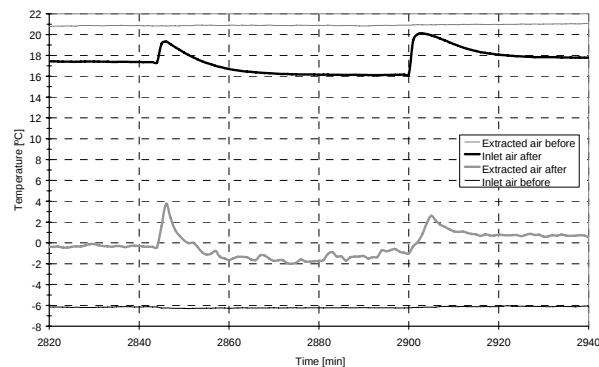


Figure 4.18 Zoom of the temperature measurements

5. Discussion and Conclusion

The work of designing a high effective heat exchanger capable of continuously defrosting itself seems to be successful. Good accordance between the calculated and measured efficiency was found at dry condition. At flow rates of 150 m³/h and using both sections simultaneously the efficiency was found to be approximately 85% and the temperature efficiency 82%. At freezing condition and using the defrosting system, the measured temperature efficiency was 88%.

Experimental test at an inlet temperature of -6°C didn't influence the mean air change rate over a period of 4 days at a flow rate of 25 l/s.

The design criteria of a simple construction and cheap materials were fulfilled, which gives the possibility of local production in the arctic regions. The volume of the exchanger could be a barrier and therefore further development/optimization of the design is necessary (minimizing the volume).

Further development could be an integration of the unit in the building envelope or an improved controlling system that gives the possibility of controlling the section switch time as a function of the inlet temperature. Also the distribution of the extracted air flow (90% and 10%) when defrosting could be optimized.

6. Acknowledgement

The work and development of the heat recovery unit presented in this paper was financed by the VILLUM KANN RASMUSSEN FOUNDATION.

7. References

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Nomenclature

LATIN SYMBOLS:

$A(h)$	Total surface area as a function of the height	$[m^2]$
A_{cross}	Cross section area	$[m^2]$
A_{duct}	Cross section area of one duct in the exchanger	$[m^2]$
c	Specific heat	J/kgK
C_{inlet}	Capacity flow of inlet air	$[W/K]$
$C_{extracted}$	Capacity flow of extracted air	$[W/K]$
d	Section dept of exchanger	$[m]$
D_h	Hydraulic diameter	$[m]$
H	Total heat transfer coefficient	W/m^2K
Nu	Nusselt number	$[-]$
P	Perimeter	$[m]$
q	air flow rate	$[m^3/h]$
Re	Reynolds number	$[-]$
v	Velocity	$[m/s]$
w_s	Section width of exchanger	$[m]$

GREEK SYMBOLS:

α	Heat transfer coefficient	W/m^2K
δ	Thickness	$[m]$
ν	Kinematic viscosity	m^2/s
λ	Conductivity	W/mK
κ	Capacity ration	$[-]$
ε	Efficiency	$[-]$
η	Temperature efficiency	$[-]$

Mechanical Ventilation System with Heat Recovery in Arctic Climates

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Technical University of Denmark

Energy-efficient building
April 12th – 14th 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

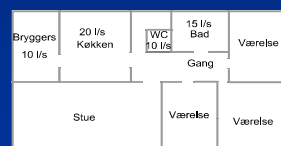
- Poor indoor climate
- Higher risk of mould and rot in building constructions

Ventilation demands in the coming building code

Total air change rate: 0.5 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 m^2 .

Outside temperature -10°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	$\eta = 60 - 70 \%$
Rotating heat exchangers	$\eta = 60 - 80 \%$
Counter flow heat exchangers	$\eta = 85 - 95 \%$

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



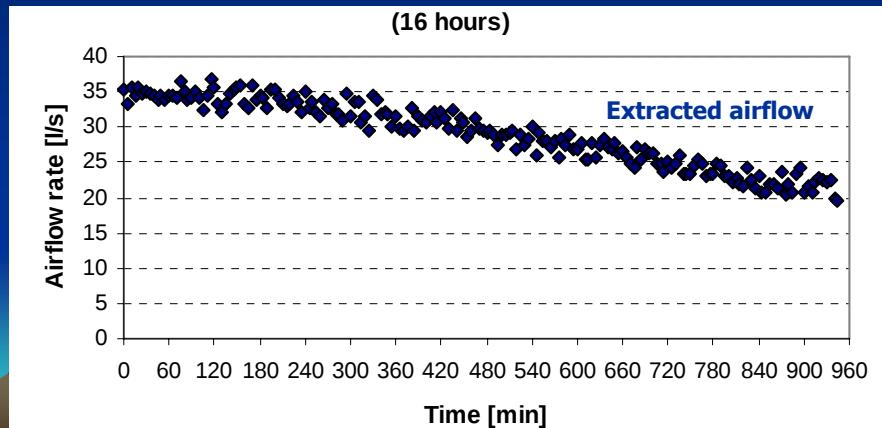
Test of small compact standard heat recovery unit

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

Room temperature: 21°C

Relative humidity: 42%

Outside temperature: -5°C

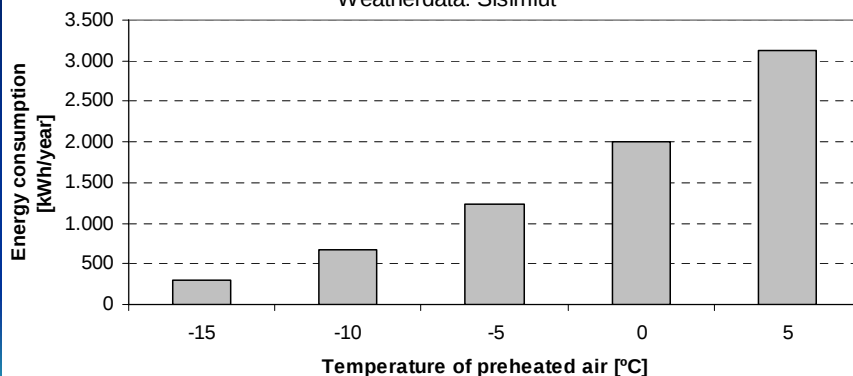


Preheating the inlet air

Energy consumption for preheating the inlet air

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

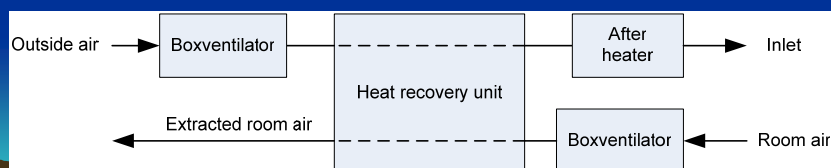
Weatherdata: Sisimiut



Annual energy saving potential (100 m² house in Sisimiut)

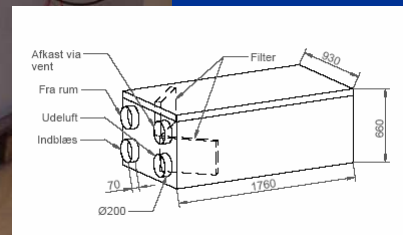
	Ventilation heat loss	Preheating	Fan energy	Savings
	kWh	kWh	kWh	kWh
Ventilation with no heat recovery	9,000	0	0	0
Today's solution: Heat recovery (60%) Preheating (-10°C)	3,300	700	600	4,400
Improved system: Heat recovery (90%) No preheating	900	0	600	7,500

The ventilation system in the low-energy house

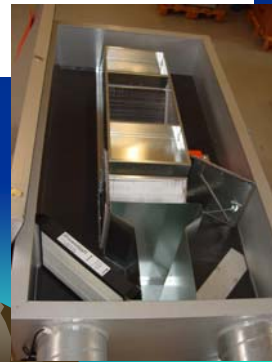
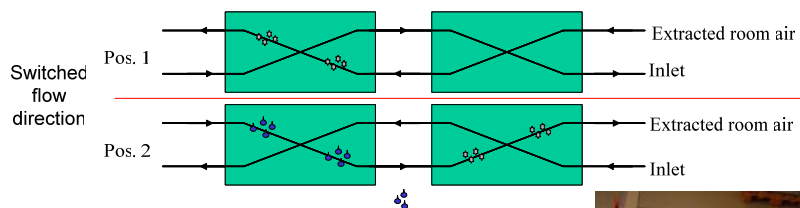


The ventilation system in the low-energy house

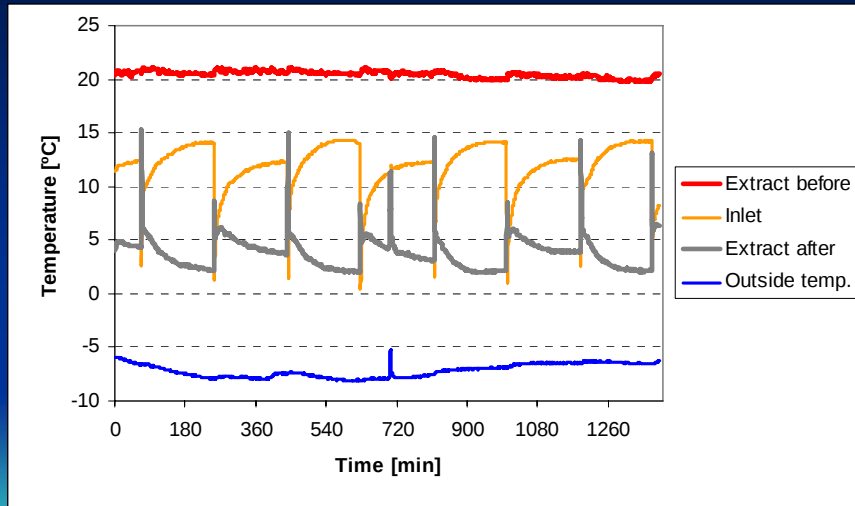
Heat recovery unit developed for the low-energy house in Sisimiut in cooperation between EXHAUSTO A/S and the Technical University of Denmark



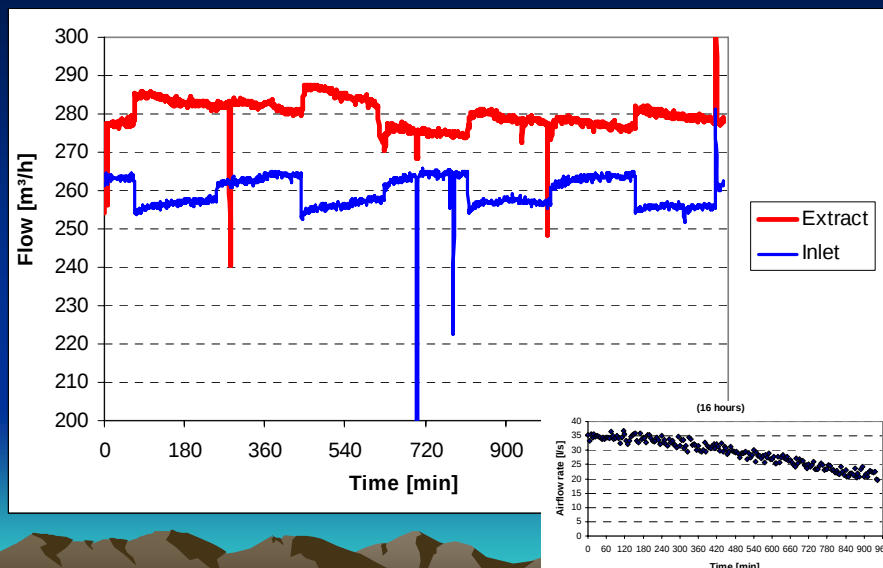
The ventilation system in the low-energy house



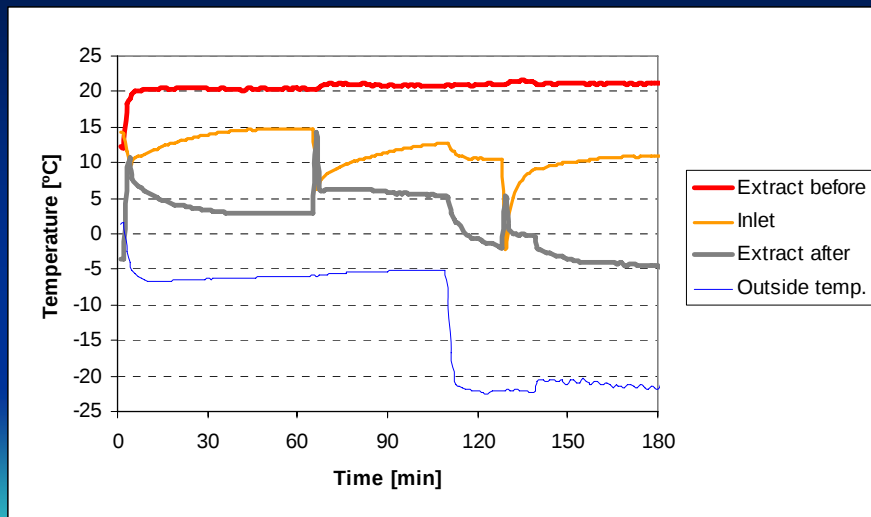
Laboratory test at DTU



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.



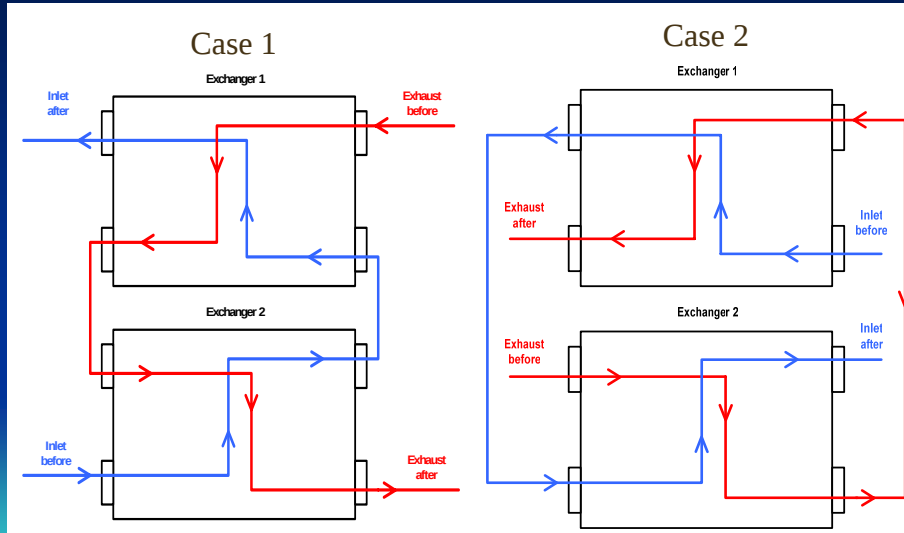
Other solutions of the frost protection

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.

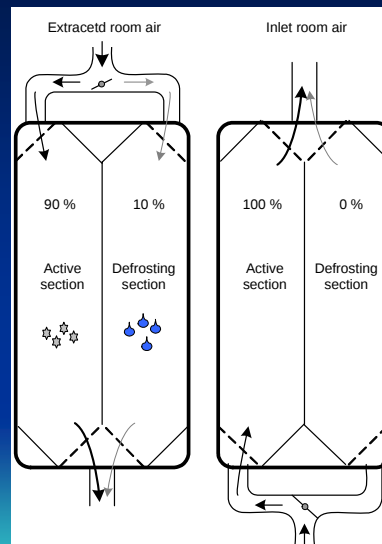


Serial connection of two exchangers

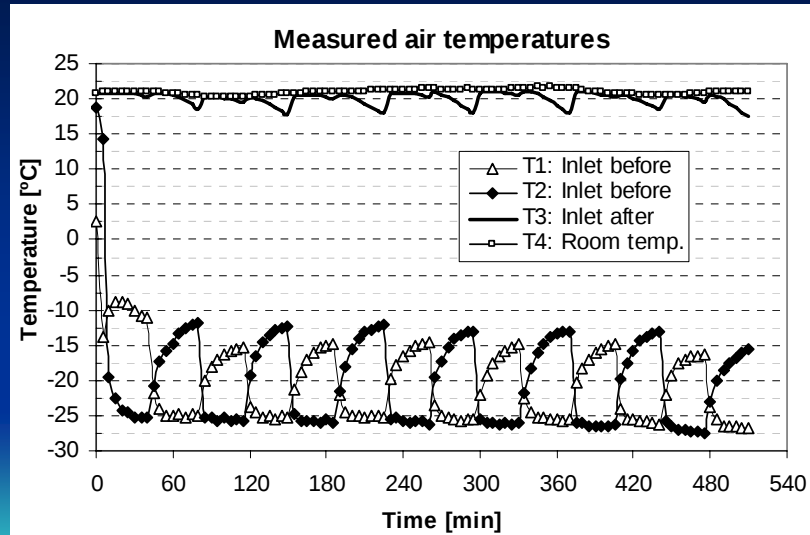


Alternating flow in two parallel coupled heat exchangers

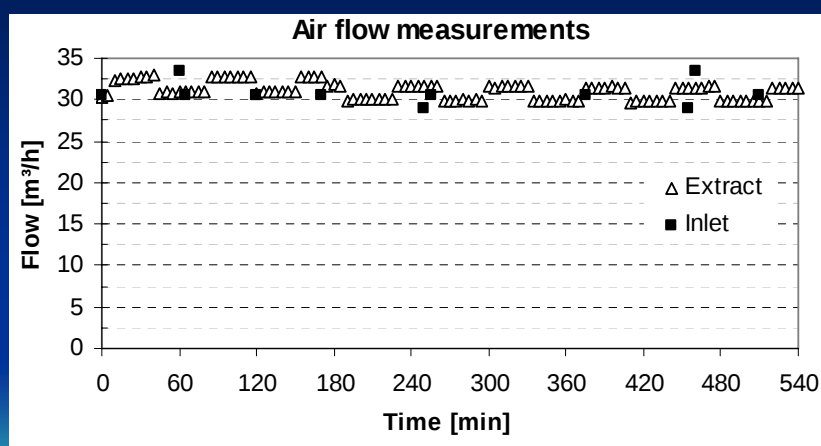
A large counter flow box exchanger made of polycarbonate plate. Since condensing water cannot be avoided, the exchanger has to be designed in a way that allows for easy removal of the water without extra pumps or valves.



Other solutions of the frost protection



Test of the alternating heat recovery unit



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

Mechanical Ventilation System with Heat Recovery in Cold Climates

Jesper Kragh
Assistant research professor
Technical University of Denmark

Nordic Symposium on Building Physics,
Reykjavik 13-15 June 2005

Agenda

- The heat recovery unit
- Heat recovery in cold climate – freezing problem
- Different solutions of the freezing problem
 - Munter's sledge solution
 - Serial connection of two exchangers
 - Alternating flow in two exchangers (parallel)
- Conclusion

The heat recovery unit

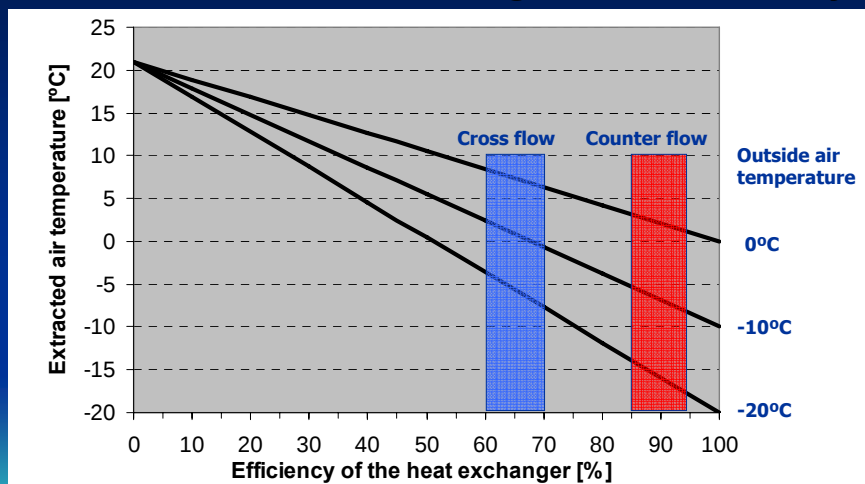
Consist typically of two ventilators, exchanger and filters.

The principle is that 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	$\eta = 60 - 70 \%$
Rotating heat exchangers	$\eta = 60 - 80 \%$
Counter flow heat exchangers	$\eta = 85 - 95 \%$

Extracted air temperature as function of the heat exchanger's efficiency



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



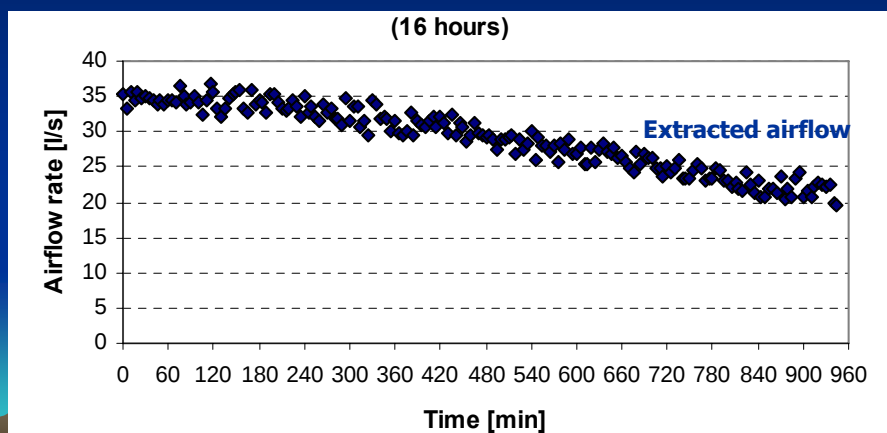
Test of small compact standard heat recovery unit

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

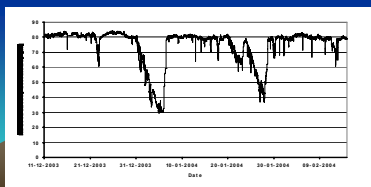
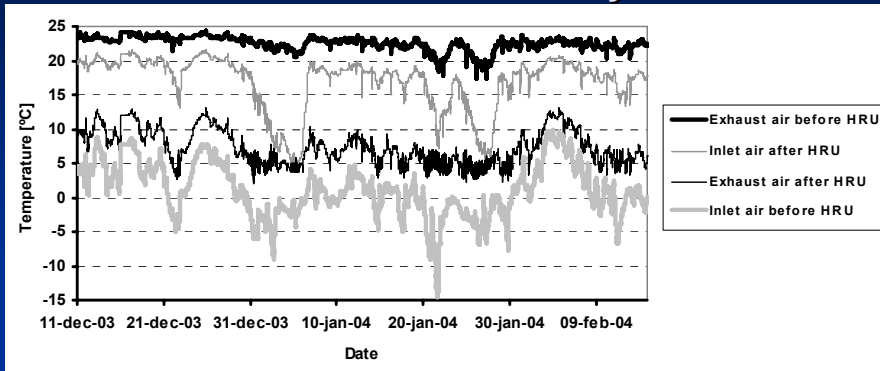
Room temperature: 21°C

Relative humidity: 42%

Outside temperature: -5°C



Measurements from ventilation system with heat recovery



Possible solutions of the frost protection

- Munter's sledge solution
- Serial connection of two exchangers
 - Switch airflow direction
 - Switch order of the exchangers
- Alternating flow in two parallel coupled heat exchangers

Possible solutions of the frost protection (2)

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.



Possible solutions of the frost protection (2)

Condensation and ice formation in the exchanger during the test

The flow starts to fall after only a few hours

The sledge solution didn't work at -5 °C.



Possible solutions of the frost protection (3)

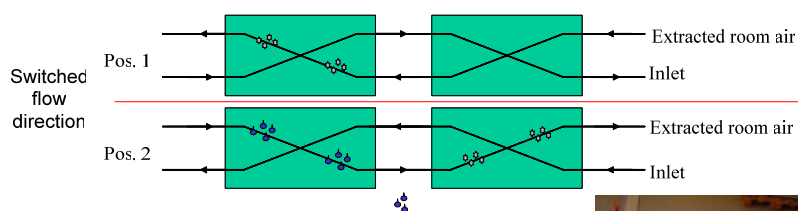
Serial connection of two exchangers – switch order

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.

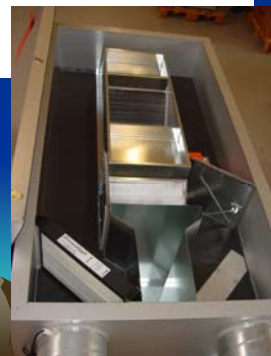
Problem: Too high pressure loss in the duct connection



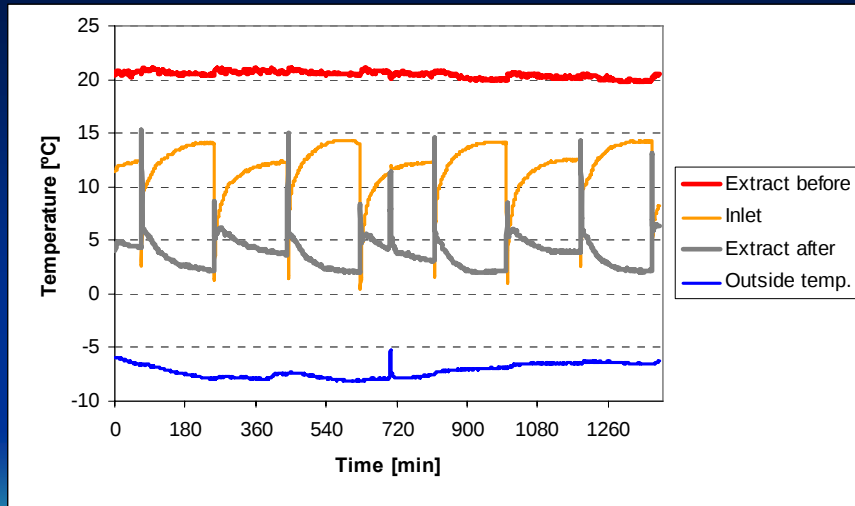
Serial connection of two exchangers Switched flow direction



Heat recovery unit developed in cooperation between EXHAUSTO A/S and the Technical University of Denmark



Laboratory test at DTU

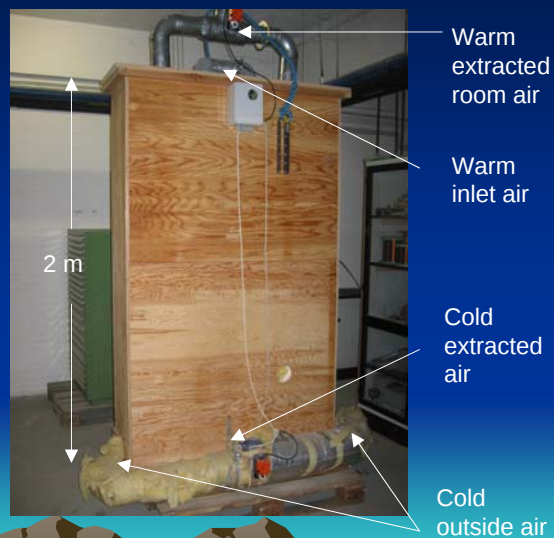


Defrosting works but problem with low inlet temperature

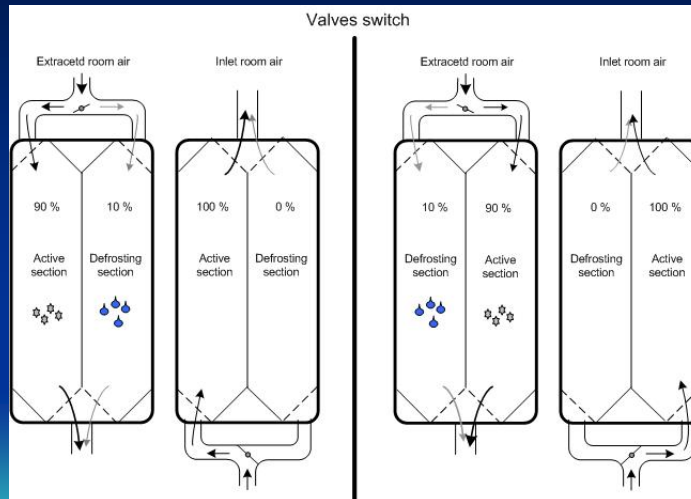
Alternating flow in two parallel coupled heat exchangers (1)

A large counter flow box exchanger made of polycarbonate plate.

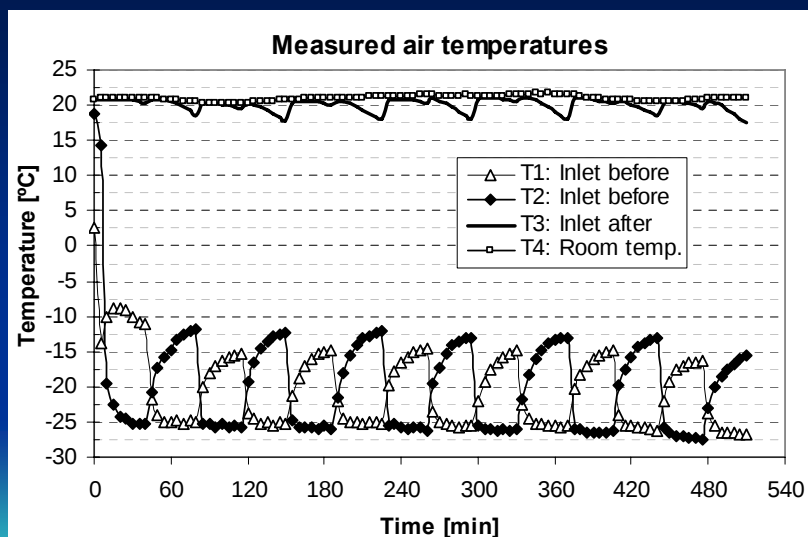
Since condensing water cannot be avoided, the exchanger has to be designed in a way that allows for easy removal of the water without extra pumps or valves.



Alternating flow in two parallel coupled heat exchangers (2)



Other solutions of the frost protection



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.

Tested solutions of the frost protection at DTU:

- Munter's sledge solution ↓
- Serial connection of two exchangers ↓
 - Airflow direction switch ↓
 - Order of the exchanger switch →
- Alternating flow in two parallel coupled heat exchangers ↑

