Bæredygtigt arktisk byggeri i det 21. århundrede

Vakuumrørsolfangere – Slutrapport til VILLUM KANN RASMUSSEN FONDEN
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Vakuumrørsolfangere – Slutrapport til VILLUM KANN RASMUSSEN FONDEN

Jianhua Fan, Simon Furbo, Janne Andersen, Rikke Jørgensen og Louise Jivan Shah
# Indholdsfortegnelse

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Introduktion

Solenergi er den reneste og naturligst energiform vi overhovedet har. Solindfaldet er så stort på kloden – og i Grønland – at der er mulighed for at udnytte solenergi i stort omfang.

Solenergi kan udnyttes til at reducere brugen af fossile brændsler, fx ved at anvende solvarmeanlæg til boliger. Solvarmeanlæg kan for eksempel benyttes til brugsvandsoptempering eller til kombineret rumopvarmning og brugsvandsoptempering.

Det årlige antal timer med mulighed for solskin er stort set det samme uanset hvor på kloden vi befinder os.


Solindfaldet på en flade afhænger stærkt af fladens lokalitet, orientering og hældning. I København (breddegrad 56°) er solindfaldet størst på en sydvendt 40° hældende flade, mens solindfaldet i Sisimiut i Grønland (breddegrad 67°) er størst på en sydvendt 60° hældende flade. Solindfaldet i København og i Sisimiut er stort set ens på de optimalt hældende flader, ca. 1160 kWh/m²år, se figur 3. I denne forbindelse skal det nævnes at solindfaldet på en lodret sydvendt flade er ca. 20% større i Sisimiut end i København.
SOLAR RADIATION MONTH BY MONTH

SOUTH-facing 40° TILTED SURFACE IN COPENHAGEN, ALBEDO: 0.2
SOUTH-facing 60° TILTED SURFACE IN SISIMIUT, ALBEDO: 0.2 MAY-SEPTEMBER 0.9 OCTOBER-APRIL

Figur 3. Solarindfald på optimalt højdede flader i Sisimiut og i København.

Anvendelsen af solvarmeanlæg varierer stærkt fra land til land. I Europa er Cypern, Østrig og Grækenland, efterfulgt af Tyskland, Danmark og Schweiz, de lande hvor der er installeret flest m² solfangere pr. indbygger. Der er ingen entydig sammenhæng mellem disse landes (relative) succes inden for solvarmeområdet, solindfald og energiprisniveau. Der er eller har været en aktiv solvarmeindustri og politisk opbakning til solvarmeanlæg i form af støtte til forskning, udvikling og demonstrationsprojekter i alle de nævnte lande. Derudover er der, eller har der været, økonomisk støtte til opførelse af solvarmeanlæg.

Solvarmeanlægs rentabilitet afhænger stærkt af energiprisniveauet og energiprisudviklingen. I Danmark har typiske solvarmeanlæg økonomiske tilbagebetalingstider på ca. 10 år og energimæssige tilbagebetalingstider på ca. 1 år, og der er inden for en forholdsvis kort tidshorisont mulighed for teknologisk udvikling så den økonomiske tilbagebetalingstid når ned på ca. 5 år.

Der er et antal barrierer for udnyttelse af solvarme i Grønland. Blandt andet kan det nævnes at:

- energipriserne for fossilt brændsel er lavere i Grønland end i Danmark.
- der ikke er en solvarmeindustri i Grønland.
- der kun er få solvarmeuddannede VVS-installatører i Grønland.
- der ikke er udviklet solvarmeanlæg som er specielt velegnede til Grønland.

Der er dog også en række forhold som gør at solvarmeanlæg er mere velegnede i Grønland end i Danmark. Blandt andet kan nævnes at:

- sne reflekterer en meget stor del af solstrålingen. Derfor er solindfaldet på tagflader i perioder med sne på jorden meget stort i Grønland.
• der er rumopvarmningsbehov i sommerperioden med meget sol i Grønland.
• temperaturen af det kolde brugsvand der tilføres boligerne, er lavere i Grønland end i Danmark.
• den optimale solfangerhældning fra vandret er større i Grønland end i Danmark. Det bevirker at solfangereffektiviteten for den samme solfanger er højere i Grønland end i Danmark.
• der er mere solindstråling fra "alle retninger" i Grønland end i Danmark. I denne forbindelse kan det nævnes at de forholdsvis billige kinesiske masseproducerede vakuumglasrør sandsynligvis er specielt velegnede til grønlandske forhold, da de kan udnytte solstrålingen fra alle retninger, dvs. de kan udnytte solstrålingen i alle døgnets lyse timer hvis blot rørene placeres lodret med frit udsyn til alle sider.

Vakuumrørsolfangere

Vakuumrørsolfangere har i mange år været markedsført i Europa og i USA. Disse solfangere er udformet efter det såkaldte heat pipe princip, se figur 4.

Solfangeren, den såkaldte heat pipe single glass solfanger, består af en række cylinderformede glasrør som øverst er koblet til en kondensator/varmeveksler-enhed. Inde i glasrørene med vakuum er placeret absorberet med selektiv belægning og et rør som
 indeholder et varmetransporterende medium, fx vand. Det varmetransporterende medium fordamer ved et lavt temperaturniveau når absorberen opvarmes af solens stråler, idet der også er vakuum i røret. Dampen stiger opad i røret til en kondensator, hvor dampen kondenserer og derved afgiver varme til solfangervæsken, som pumpes gennem kondensator/varmeveksler-enheden. I kondensatoren kondenserer det varmetransporterende medium, der som væske flyder ned til bunden af røret hvor det igen fordamer hvis temperaturen er højt nok hvorefter processen gentages.

Da der er vacuum i glasrørene er varmetafelt fra absorberne på grund af konvektion og varmeledning meget lille. Varmetabskoefficienten for vakuumrørsolfangere er derfor meget mindre end varmetabskoefficienten for almindelige plane solfangere.

Vakuumrørsolfangere kan, i modsætning til almindelige plane solfangere, udnytte solstråling specielt godt når indfaldsvinklen er stor. Årsagen til dette forhold er dels refleksionsforholdene mellem glasrørene, dels glasrørenes cylinderformede overflade, som tillader at solstråler transmitteres gennem glasset selv ved store indfaldsvinkler på tværs af glasrørene.

Vakuumrørsolfangere udnytter altså solens stråler specielt godt ved høje solfangervæsketemperaturen, ved lave udelufttemperaturer, ved små bestrålingsstyrker og ved store indfaldsvinkler.

De europæiske og amerikanske vakuumrørsolfangere har ikke nået så lavt et prisniveau, at der i Europa og i USA har været i stand til at erobre en væsentlig del af solvæskemarkedet fra almindelige plane solfangere.

For nylig har en række kinesiske firmaer startet masseproduktion af forholdsvis billige vakuumrørsolfangere. Firmaerne producerer forskelligt udformede vakuumrørsolfangere, fx med forskellige absorbere og glasrørdiametre med eller uden reflektorer.

I Asien har vakuumrørsolfangere i modsætning til i Europa og i USA nået så lavt et prisniveau og så høj en effektivitet at det er blevet attraktivt at benytte disse højeffektive solfangere i stedet for almindelige plane solfangere.

De mest anvendte kinesiske vakuumrørsolfangere, de såkaldte all glass solfangere, er udført på en simplice måde end de europæiske og amerikanske vakuumrørsolfangere. De er baseret på dobbeltglasrør, se figur 5, med vacuum i mellemrummet mellem glassene. De udvendige overflader af de inderste glasrør har en høj absorptans og en lav emittans. Når solen skinner på glasrøret bliver det indvendige glasrør derfor meget varmt. Varmen fra det indvendige glasrør kan overføres til solfangervæsken på forskellige måder: Enten kan solfangervæsken strømme igennem det indvendige glasrør i direkte kontakt med glasvæggen eller solfangervæsken kan strømme igennem et metalrør, som er i god termisk kontakt med det indvendige glas. En tredje mulighed er at anvende en heat pipe i god termisk kontakt med det indvendige glasrør.

Der er forskellige muligheder for at sammenkoble sådanne vakuumrør til solfangerpanel og dermed også forskellige muligheder for solfangervæskens passage gennem solfangerpanelet.
Vakuumrørskonceptet er ikke kun interessant for arktiske forhold. Det er interessant for alle klimaforhold og for de fleste typer af solvarmeanlæg. Det er bl.a. fordi der med optimalt designede vakuumrørsolfangere er mulighed for at forbedre solvarmeanlægss rentabilitet mærkbart. Forskningsprojektet har til formål at undersøge hvilke vakuumrørsolfangere der er bedst egnede til Arktis.

![Figur 5. Dobbeltglasrør, som anvendes til all glass solfangere.](image)

**All glas solfangere**

Der er udviklet teoretiske modeller til beregning af termiske ydelser for all glass vakuumrørsolfangere, der udnytter solstrålingen fra alle retninger.

Den teoretiske solfangermodel er sammenholdt med målinger på en prototype solfanger, se figur 7, og det viser sig at modellen gengiver "virkeligheden" med stor nøjagtighed. Modellen er herefter videreudviklet så den nu kan indgå i simuleringsprogrammet TRNSYS. Dette amerikanske simuleringsprogram er et komponent baseret program, som er det mest anvendte og anerkendte simuleringsprogram til solvarmeanlæg.

Med modellen er der lavet analyser af, hvilke solfangerydelser man kan forvente i hhv. Danmark og Grønland (Uummannaq). Resultaterne viser, at vakuumrørsolfangerne kan give en meget større ydelse i Grønland end i Danmark, se figur 8.

![Figur 6. Rør der skygger for hinanden.](image-url)
Figur 7. All glass prototype solfanger.

Figur 8. Solfangerydelse i Grønland og i Danmark.

Der er desuden gennemført detaljerede teoretiske undersøgelser af en all glass vakuumrørsolfanger, hvor solfangervæsken opvarmes ved direkte kontakt med de indvendige solopvarmede glas, se figur 9. Undersøgelserne, som er gennemført ved hjælp af CFD
(Computational Fluid Dynamics) beregninger, har klargjort, hvorledes flowfordelingen i solfangeren og solfangereffektiviteten afhænger af volumenstrømmen gennem solfangeren.

Figur 9. Principskitse af all glass vakuumrørsolfanger med solfangervæsken direkte placeret i det indvendige glasrør.

**Heat pipe single glass solfanger**

Der er udviklet teoretiske modeller til beregning af termiske ydelser for heat pipe single glass solfanger. Disse solfanger består af en række cylinderformede glasrør som øverst er koblet til en kondensator/varmeveksler-enhed, se figur 10.

Traditionelle solfangerteorier fra litteraturen er udviklet med henblik på almindelige plane solfanger med plane absorbere og har ikke kunnet benyttes direkte til heat pipe single glass vakuumrørsolfanger. Derfor er der udviklet to nye teoretiske solfangermodeller til heat pipe single
glass vakuumrørsolfangere med hhv. krumme og plane finner. Modellerne udmærker sig især ved at de præcist bestemmer skyggeeffekterne fra rør til rør, ligesom de detaljeret bestemmer temperaturprofilet på finnen, se figur 11.

Figur 11. Heat pipe single glass vakuumrør med hhv. plane og krumme finner.

Målinger i prøvestand

I 2005 blev der på BYG.DTU’s forsøgsareal opbygget en prøvestand hvor 5 vakuumrørsolfangere kan afprøves under ens driftsbetingelser. Prøvestanden er placeret således at solfangerne kan modtage solstråling fra alle retninger uden at solfangerne ofte rammes af skygger, se figur 12.

Figur 12. Prøvestand til 5 vakuumrørsolfangere.
I prøvestanden kan solfangervæske tilføres de 5 solfanger med samme fremløbstemperatur, og volumenstrømmen igennem den enkelte solfangere kan styres så middelsoflangervæsketemperaturen i alle solfangerne bliver næsten den samme igennem en måleperiode. Volumenstrømmen igennem hver solfangere måles med en flowmmåler af mærket Brunata HGQ1-R0, og fremløbs- og returtemperaturen måles for hver enkelt solfangere med kobber/konstantan termoelementtråd type TT. Temperaturdifferens mellem fremløbs- og returtemperaturen for alle solfangerne måles med termosøjler bestående af 5 elementer. Der benyttes også her termoelementtråd af kobber/konstantan type TT. Målenøjagtigheden for flowmålingerne er 1%, for temperaturmålingerne 0,5 K og for temperaturdifferensmålingerne 0,05 K.

Solindfaldet, både total og diffus stråling på en vandret flade og udelufttemperaturen måles igennem hele måleperioden.

Efter en indkørlingsperiode for prøvestanden er der foretaget målinger i perioderne uge 6-25 og uge 29-45 i 2006. I den første måleperiode er der afprøvet 5 vakuumrørsolfangere fra kinesiske producenter, i den anden måleperiode er der afprøvet 4 solfanger fra kinesiske fabrikanter og fra det svenske firma ExoHeat AB, se figur 13, 14, 15, 16, 17 og 18 samt tabel 1.

Figur 13. Seido 5-8 solfangeren i prøvestanden.
Figur 14. Seido 1-8 solfangeren i prøvestanden.

Figur 15. Seido 10-20 solfangeren med krum absorber i prøvestanden.
Figur 16. Seido 1-20 solfangeren med plan absorber i prøvestanden.

Figur 17. SLL solfangeren i prøvestanden.
Figur 18. ExoHeat solfangeren i prøvestanden.

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<th>Seido 1-8</th>
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<th>Seido 10-20 med plan absorber</th>
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<td>0,91</td>
<td>0,91</td>
<td>0,91</td>
<td>0,91</td>
<td>0,91</td>
</tr>
<tr>
<td>Vakuum</td>
<td>&lt; 10⁻³ Pa</td>
<td>&lt; 10⁻³ Pa</td>
<td>&lt; 10⁻³ Pa</td>
<td>&lt; 10⁻³ Pa</td>
<td>&lt; 5x10⁻² Pa</td>
<td>&lt; 5x10⁻³ Pa</td>
</tr>
<tr>
<td>Manifold længde</td>
<td>960 mm</td>
<td>960 mm</td>
<td>1860 mm</td>
<td>1860 mm</td>
<td>2000 mm</td>
<td>2000 mm</td>
</tr>
<tr>
<td>Manifold diameter</td>
<td>28 mm</td>
<td>28 mm</td>
<td>38 mm</td>
<td>38 mm</td>
<td>45 mm</td>
<td>22/38 mm</td>
</tr>
<tr>
<td>Væskeindhold i manifold</td>
<td>0,48 l</td>
<td>0,48 l</td>
<td>0,80 l</td>
<td>0,80 l</td>
<td>1,21 l</td>
<td>2,00 l</td>
</tr>
<tr>
<td>Symbol for solfangere</td>
<td><img src="image1.png" alt="Symbol for solfangere" /></td>
<td><img src="image2.png" alt="Symbol for solfangere" /></td>
<td><img src="image3.png" alt="Symbol for solfangere" /></td>
<td><img src="image4.png" alt="Symbol for solfangere" /></td>
<td><img src="image5.png" alt="Symbol for solfangere" /></td>
<td><img src="image6.png" alt="Symbol for solfangere" /></td>
</tr>
</tbody>
</table>

Tabel 1. Data for de afprøvede solfangere.
Solfangerne fra Sunda Technology Ltd. er fremstillet så absorbernes for- og bagside har samme absorberende og selektive overflade, således at solstråling kan udnyttes uanset retningen hvorfra den kommer. Solfangerne er orienteret 15° mod vest fra syd og de har en hældning fra vandret på 70°. Middelsolfangervæsketemperaturen i solfangerne er blevet ændret to gange i løbet af de to afprøvningsperioder. Solfangernes fremløbstemperatur og volumenstrømme er indstillet således at middelsolfangervæsketemperaturen i solfangerne i uge 6-10 er ca. 44°C, i uge 11-19 ca. 63°C og i uge 2-25 ca. 76°C. I uge 29-36 var middelsolfangervæsketemperaturen i solfangerne ca. 76°C, i uge 37-40 ca. 64°C og i uge 41-45 ca. 44°C, se figur 19, 20 og 21.

**Figur 19. Fremløbstemperaturer for solfangerne i de to afprøvningsperioder.**

**Figur 20. Volumenstrømme gennem solfangerne i de to afprøvningsperioder.**
Figur 21. Midelsofvangervæsketemperaturer i solfangere i de to afprøvningsperioder.

På den måde kan sofangernes ydelser klarlægges for perioder af året med forskellige solbaner over himlen samt for forskellige temperaturniveauer. På figur 22, som viser et soldiagram for København, er solhøjden vist som funktion af solazimuth for den 21./22. for hver måned igennem året. I midten af sommeren er solen på himlen i mange timer hvert døgn og den direkte solstråling rammer sofangere såvel forfra (midt på dagen) som bagfra (om morgenen og om aftenen). Om vinteren rammer den direkte solstråling kun sofangerne forfra, da solen altid befinder sig på den sydlige del af himlen.

De målte sofangerydelser samt det målte totale solindfald på en vandret flade uge for uge er vist på figur 23. Det er svært at vurdere ydelserne for de forskellige sofangere, da de har forskellige størrelser og da ydelserne vairer meget igennem året. Figur 24 viser relative ydelser for sofangere, defineret som:

Q2/Q1: Forholdet mellem ydelsen af Seido 5-8 og ydelsen af Seido 1-8.
Q3/Q4: Forholdet mellem ydelsen af Seido 10-20 med krum absorber og ydelsen af Seido 10-20 med plan absorber.
Q5/Q4: Forholdet mellem ydelsen af Tsinghua sofangeren og Seido 10-20 med plan absorber.

Af figuren ses at Seido 5-8 med den krumme absorber yder mindre end den tilsvarende Seido 1-8 med den plane absorber om vinteren, hvor solen befinder sig på den sydlige himmel. Jo nærmere man kommer på midten af sommeren des bedre klarer Seido 5-8 sig i forhold til Seido 1-8. Det skyldes at den krumme absorber er i stand til at udnytte solstrålingen bedre end den plane absorber om sommeren, hvor solstrålingen kommer fra forskellige retninger. Desuden ses det at ydelsen for Seido 5-8 med den krumme absorber er relativ lav i perioder med høje driftstemperaturer sammenlignet med Seido 1-8 med den
plane absorber. Det skyldes at varmetabet fra den krumme og relativ store absorber er større end varmetabet fra den plane og relativ lille absorber.

Figur 22. Soldigram for København.

Figur 23. Målte solfangerydelser og solindfald på vandret.
Det ses også at Seido 10-20 med den plane absorber yder mere end Seido 10-20 med den krumme absorber. Også for disse solfangere klarer solfangeren sig med den krumme absorber ydelsesmæssigt relativt bedre om sommeren end om vinteren og ved lave driftstemperaturer.

Solfangeren fra ExhoHeat klarer sig i forhold til Seido 10-20 med plane absorbere relativt bedst midt om sommeren. Forklaringen er at absorberen, som udgøres af det indvendige rørs udvendige overflade er i stand til at udnytte solstrålingen fra alle retninger. Den relative ydelse af Exoheat solfangeren i forhold til Seido 10-20 med plane absorbere påvirkes ikke nævneværdigt af temperaturniveauet.

Tsinghua solfangeren klarer sig i forhold til Seido 10-20 solfangeren med plane absorbere meget bedre om sommeren end om vinteren. Forklaringen må være at de vandrette rør med cylinderformede absorbere kan udnytte solstrålingen fra alle retninger og at rørene ikke kaster skygger på hinanden i samme grad som lodrette rør gør. Den relative ydelse af Tsinghua solfangeren i forhold til Seido 10-20 med plane absorbere påvirkes ikke nævneværdigt af temperaturniveauet.

Figur 25 og 26 viser de målte solfangerydelser for de to afprøvningsperioder. Seido 1-8 yder en smule mere end Seido 5-8, mens Seido 10-20 med plane absorbere yder 13% mere end Seido 10-20 med krumme absorbere. Tsinghua solfangeren yder mest af de 6 afprøvede solfangere. Exoheat solfangeren yder næstmest.

Figur 27 og 28 viser solfangernes ydelser pr. rør. Det ses at rørene med de store transparente arealer Seido 1-8 og Seido 5-8 yder mest.

Figur 29 og 30 viser solfangernes ydelser pr. transparent areal. Det ses at Tsinghua solfangeren yder mest pr. transparent areal.


**Validering af simuleringsmodeller**

For de fire afprøvede heat pipe single glass solfangere er målte ydelser sammenlignet med beregnede ydelser med de udviklede simuleringsmodeller for tre udvalgte perioder.


Figur 34. Målt globalstråling og udelufttemperatur i perioden 1.-7. maj, 2006.


Simuleringsmodellen er i stand til med god nøjagtighed at beregne solfangerydelsen for Seido 5-8 for de tre perioder, som har forskellige driftstemperaturer. Simuleringsmodellerne kan også beregne ydelserne for de tre andre heat pipe single glass solfangere med god nøjagtighed i de tre perioder. Tabel 2, 3 og 4 viser beregnede og målte ydelser samt forskellen mellem målt og beregnet ydelse for de tre perioder. Det ses at den største forskel mellem målt og beregnet ydelse er 1,7%.
Undersøgelserne har altså vist at de udviklede simuleringsmodeller for heat pipe single glass solfangere med god nøjagtighed kan beregne solfangerydelsen, både ved forskellige driftstemperaturer og i forskellige perioder igennem året. Derfor kan simuleringsmodellerne benyttes til at beregne årsydelser for solfangere for forskellige lokaliteter ved forskellige temperaturniveauer.

**Ydelsesberegninger**

Der er med de validerede simuleringsmodeller gennemførte beregninger af årsydelser for de fire heat pipe single glass solfangere: Seido 5-8, Seido 1-8, Seido 10-20 med krum absorber og Seido 10-20 med plan absorber. Desuden er for sammenlignings skyld gennemført beregninger af årsydelser for en nyudviklet højeeffektiv plan solfangere til solvarmecentraler, HT solfangere fra Arcon Solvarme A/S. HT solfangere har et transparent areal på 12,53 m² og et bruttoareal på 13,53 m². Beregningerne er gennemført

<table>
<thead>
<tr>
<th>Solfanger</th>
<th>Seido 5-8</th>
<th>Seido 1-8</th>
<th>Seido 10-20</th>
<th>Seido 10-20</th>
</tr>
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<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>med krum</td>
<td>med plan</td>
</tr>
<tr>
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<td>28,8 kWh</td>
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<tr>
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<td>-1,7%</td>
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</table>


<table>
<thead>
<tr>
<th>Solfanger</th>
<th>Seido 5-8</th>
<th>Seido 1-8</th>
<th>Seido 10-20</th>
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<tbody>
<tr>
<td></td>
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<td>med krum</td>
<td>med plan</td>
</tr>
<tr>
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<td>29,7 kWh</td>
<td>41,1 kWh</td>
<td>45,5 kWh</td>
</tr>
<tr>
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<td>27,0 kWh</td>
<td>29,4 kWh</td>
<td>41,1 kWh</td>
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<tr>
<td>Forskel</td>
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<td>1,0%</td>
<td>0,0%</td>
<td>0,9%</td>
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</tbody>
</table>


<table>
<thead>
<tr>
<th>Solfanger</th>
<th>Seido 5-8</th>
<th>Seido 1-8</th>
<th>Seido 10-20</th>
<th>Seido 10-20</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>med krum</td>
<td>med plan</td>
</tr>
<tr>
<td>Beregnet ydelse</td>
<td>24,5 kWh</td>
<td>28,0 kWh</td>
<td>39,3 kWh</td>
<td>44,4 kWh</td>
</tr>
<tr>
<td>Målt ydelse</td>
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<td>28,3 kWh</td>
<td>39,5 kWh</td>
<td>44,4 kWh</td>
</tr>
<tr>
<td>Forskel</td>
<td>0,4%</td>
<td>-1,1%</td>
<td>-0,5%</td>
<td>0,0%</td>
</tr>
</tbody>
</table>

med vejrdatal fra referenceår for Sisimiut (breddegrad 67˚) og Uummannaq (breddegrad 71˚).

Figur 45 viser sydvendte vakuumrørsolfangeres årlige ydelse som funktion af solfangerhældningen, når solfangervæsketemperaturen konstant er 60°C og solfangerne er placeret i Sisimiut. Det ses, at den optimale solfangerhældning er 62˚ for solfangerne med plane absorbere og 67˚ for solfangerne med krumme absorbere.

Figur 46 viser vakuumrørsolfangernes årlige ydelse i Sisimiut som funktion af solfangerorienteringen, når solfangervæsketemperaturen er 60°C og solfangerhældningen er 62˚ for solfangerne med plane absorbere og 67˚ for solfangerne med krumme absorbere. Det ses at solfangerne yder mest når solfangerne vender 40˚ mod vest fra syd.
Figur 46. Årlige ydelser for vakuumrørsolfangere i Sisimiut som funktion af solfangerorienteringen. Solfangervæsketemperaturen er 60°C og solfangerhældningen er 62° for solfangerne med plane absorber og 67° for solfangerne med krumme absorber.

Figur 47 viser vakuumrørsolfangernes årlige ydelser i Sisimiut som funktion af solfangerhældningen når solfangerne vender 40° mod vest fra syd og når solfangervæsketemperaturen er 60°C. Det ses at solfangerne yder mest når solfangerhældningen er 62° for solfangerne med plane absorber og 67° for solfangerne med krumme absorber.

Figur 47. Årlige ydelser for vakuumrørsolfangere i Sisimiut som funktion af solfangerhældningen. Solfangervæsketemperaturen er 60°C og solfangerne vender 40° mod vest fra syd.
Tilsvarende undersøgelser er gennemført for den plane solfanger HT for at bestemme den orientering og hældning for solfangeren som resulterer i den højeste årlige ydelse. Undersøgelserne viste at HT solfangeren i Sisimiut ved en solfangervæsketemperatur på 60°C yder mest når solfangeren vender 9° mod vest fra syd og når solfangervæsketemperaturen er 53°.


Figur 49 viser vakuumrørsolfangernes ydelser pr. rør i Sisimiut. Også her er de optimale solfangerorienteringer og solfangervæsketemperaturer forudsat anvendt. Ikke overraskende ses det at de store rør yder mere end de små rør. Desuden ses det at solfangerne med de plane absorbere yder mere end solfangerne med de krumme absorbere, specielt ved de høje driftstemperaturer.
Figur 49. Årlige ydelser for vakuumrørsolfangere pr. rør i Sisimiut som funktion af solfangervæsketemperaturen. Optimale solfangerorienteringer og -hældninger er forudsat.

Figur 50 viser beregnede årlige ydelser pr. transparent areal for de 4 vakuumrørsolfangere og for den plane HT solfanger som funktion af solfangervæsketemperaturen. Solfangerne er placeret i Sisimiut og den optimale solfangerhældning og solfangerorientering er forudsat anvendt. For lave driftstemperaturer yder den plane solfanger bedst. For temperaturer højere end 30°C yder vakuumrørsolfangerne med de plane absorbere bedst.

Figur 50. Årlige ydelser pr. transparent areal for vakuumrørsolfangere og HT solfangeren i Sisimiut som funktion af solfangervæsketemperaturen. Optimale solfangerorienteringer og -hældninger er forudsat.
Figur 51 viser beregnede årlige ydelser pr. bruttoareal for de 4 vakuumrørsolfangere og for den plane HT solfanger som funktion af solfangervæsketemperatur. Solfangerne er placeret i Sisimiut og den optimale solfangerhældning og solfangerorientering er forudsat anvendt. For lave driftstemperaturer yder den plane solfanger bedst. For temperaturer højere end 65°C yder Seido 1-8 bedst.

Figur 52 viser beregnede årlige ydelser for de 4 vakuumrørsolfangere som funktion af rørafstand. Solfangerne er placeret i Sisimiut, solfangervæsketemperaturen er 60°C og den optimale solfangerhældning og solfangerorientering er forudsat anvendt. Seido 10-20 solfangerne med de store solfangerarealer yder mere end Seido 5-8 og Seido 1-8 solfangerne med de små solfangerarealer. For voksende rørafstand forøges solfangerens varmetab på grund af det lange manifoldrør hvorfor ydelsen reduceres. Reduceres rørafstanden vil rørene kaste skygger på naborørene. For solfangerne med de krumme absorbere vil ydelsen derfor reduceres, mens det reducerede varmetab på grund af det kortere manifoldrør overstiger reduktionen i ydelsen forårsaget af skygger for solfangerne med plane absorbere. For disse solfangere fås derfor den højeste ydelse ved at placere rørene helt tæt.

Figur 53 viser beregnede årlige ydelser pr. rør for de 4 vakuumrør-solfangere som funktion af rørafstand. Solfangerne er placeret i Sisimiut, solfangervæsketemperaturen er 60°C og den optimale solfangerhældning og solfangerorientering er forudsat anvendt. De store rør yder naturligvis mere end de små rør. For voksende rørafstand forøges solfangerens varmetab på grund af det lange manifoldrør hvorfor ydelsen reduceres. Reduceres rørafstanden vil rørene kaste skygger på naborørene. For solfangerne med de krumme absorbere vil ydelsen derfor reduceres, mens det reducerede varmetab på grund af det
kortere manifoldrør overstiger reduktionen i ydelsen forårsaget af skygger for solfangere med plane absorbere. For disse solfangere fås derfor den højeste ydelse pr. rør ved at placere rørene helt tæt.

Figur 54 viser beregnede årlige ydelser pr. transparent areal for de 4 vakuumrørsolfangere som funktion af rørafstanden. Solfangere er placeret i Sisimiut, solfangervæsketemperatur er 60°C og den optimale solfangerhældning og solfangerorientering er forudsat anvendt. Solfangere med de plane absorbere yder mere end solfangere med de krumme absorbere, specielt ved små rørafstande. For voksende rørafstand forøges solfangerens varmetab på grund af det lange manifoldrør hvorfor ydelsen reduceres. Reduceret rørafstanden vil rørene kaste skygger på naborørene. For solfangere med de plane absorbere vil ydelsen derfor reduceres, mens det reducerede varmetab på grund af det kortere manifoldrør overstiger reduktionen i ydelsen forårsaget af skygger for solfangere med plane absorbere. For disse solfangere fås derfor den højeste ydelse pr. transparent areal ved at placere rørene helt tæt.

Figur 55 viser beregnede årlige ydelser pr. bruttoreal for de 4 vakuumrørsolfangere som funktion af rørafstanden. Solfangere er placeret i Sisimiut, solfangervæsketemperatur er 60°C og den optimale solfangerhældning og solfangerorientering er forudsat anvendt. Jo mindre rørafstanden er, des højere er ydelsen pr. bruttoreal. Solfangere med de store rør har højere ydelser pr. bruttoreal end solfangere med de små rør, og solfangere med de plane absorbere har højere ydelser pr. bruttoreal end solfangere med de krumme absorbere.

Figur 52. Årlige ydelser i Sisimiut for vakuumrørsolfangere som funktion af rørafstanden. Solfangervæsketemperatur er 60°C og optimale solfangerorienteringer og -hældninger er forudsat.
Figur 53. Årlige ydelser pr. rør i Sisimiut for vakuumrørsolfangere som funktion af rørafstanden. Solfangervæsketemperaturen er 60°C og optimale solfangerorienteringer og -hældninger er forudsat.

Figur 54. Årlige ydelser pr. transparent areal i Sisimiut for vakuumrørsolfangere som funktion af rørafstanden. Solfangervæsketemperaturen er 60°C og optimale solfangerorienteringer og -hældninger er forudsat.
Figur 55. Årlige ydelser pr. bruttoareal i Sisimiut for vakuumrørsolfangere som funktion af rørafstanden. Solfangervæsketemperaturer er 60°C og optimale solfangerorienteringer og -hældninger er forudsat.

Figur 56 viser sydvendte vakuumrørsolfangere’s årlige ydelse som funktion af solfangerhældningen, når solfangervæsken konstant er 60°C og solfangerne er placeret i Uummannaq. Det ses, at den optimale solfangerhældning er 66° for solfangerne med plane absorbere og 70° for solfangerne med krumme absorbere.

Figur 56. Årlige ydelser for sydvendte vakuumrørsolfangere i Uummannaq som funktion af solfangerhældningen. Solfangervæsketemperaturen er 60°C.
Figur 57 viser vakuumrørsolfangernes årlige ydelser i Uummannaq som funktion af solfangerorienteringen, når solfangervæsketemperaturen er 60°C og solfangerhældningen er 66° for solfangerne med plane absorbere og 70° for solfangerne med krumme absorbere. Det ses at solfangerne yder mest når solfangerne vender 10° mod vest fra syd.

Figur 57. Årlige ydelser for vakuumrørsolfangere i Uummannaq som funktion af solfangerorienteringen. Solfangervæsketemperaturen er 60°C og solfangerhældningen er 66° for solfangerne med plane absorbere og 70° for solfangerne med krumme absorbere.

Figur 58 viser vakuumrørsolfangernes årlige ydelser i Uummannaq som funktion af solfangerhældningen når solfangerne vender 10° mod vest fra syd og når solfangervæsketemperaturen er 60°C. Det ses at solfangerne yder mest når solfangerhældningen er 66° for solfangerne med plane absorbere og 70° for solfangerne med krumme absorbere.
Figur 58. Årlige ydelser for vakuumrørsofangere i Uummannaq som funktion af solfangerehældningen. Solfangervæsketemperaturen er 60°C og solfangerne vender 10° mod vest fra syd.

Tilsvarende undersøgelser er gennemført for den plane solfangere HT placeret i Uummannaq for at bestemme den orientering og hældning for solfangeren som resulterer i den højeste årlige ydelse. Undersøgelserne viste at HT solfangeren i Uummannaq ved en solfangervæsketemperatur på 60°C yder mest når solfangeren vender 5° mod vest fra syd og når solfangerehældningen er 56°.

Figur 59 viser beregnede årlige ydelser i Uummannaq for de 4 vakuumrørsofangere og for den plane HT solfangere som funktion af solfangervæsketemperaturen. Den optimale solfangerehældning og solfangereorientering er forudsat anvendt. Ikke overraskende ses det, at den store HT solfangere yder mere end de små vakuumrørsofangere.
Figur 59. Årlige ydelser for vakuumrørsolfangere og HT solfangeren i Uummannaq som funktion af solfangervæsketemperaturen. Optimale solfangerorienteringer og -hældninger er forudsat.

Figur 60 viser vakuumrørsolfangernes ydelser pr. rør i Uummannaq. Også her er de optimale solfangerorienteringer og solfangerhældninger forudsat anvendt. Ikke overraskende ses det at de store rør også i Uummannaq yder mere end de små rør. Desuden ses det at solfangerne med de plane absorbere yder mere end solfangerne med de krumme absorbere, specielt ved de høje driftstemperaturer.

Figur 60. Årlige ydelser for vakuumrørsolfangere pr. rør i Uummannaq som funktion af solfangervæsketemperaturen. Optimale solfangerorienteringer og -hældninger er forudsat.
Figur 61 viser beregnede årlige ydelser pr. transparent areal for de 4 vakuumrørsolfangere og for den plane HT solfanger som funktion af solfangervæsketemperaturen. Solfangerne er placeret i Uummannaq og den optimale solfangerhældning og solfangerorientering er forudsat anvendt. For lave driftstemperaturer yder den plane solfanger bedst. For temperaturer højere end 20°C yder vakuumrørsolfangerne med de plane solfangere bedst.

Figur 62 viser beregnede årlige ydelser pr. bruttoareal for de 4 vakuumrørsolfangere og for den plane HT solfanger som funktion af solfangervæsketemperaturen. Solfangerne er placeret i Uummannaq og den optimale solfangerhældning og solfangerorientering er forudsat anvendt. For lave driftstemperaturer yder den plane solfanger bedst. For temperaturer højere end 70°C yder Seido 1-8 bedst.
Konklusion og aktiviteter fremover

Projektet har vist at vakuumrørsolfangere, som udnytter solstråling fra alle retninger, er specielt velegnede til solvarmeanlæg i Arktis.


Figur 62. Årlige ydelser pr. bruttoareal i Uummannaq for vakuumrørsolfangere og HT solfangeren som funktion af solfangervæsketemperaturen. Optimale solfangereorienteringer og -hældninger er forudsat.
Publikationer


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Foredrag og anden formidling


Artikel i den grønlandske avis Sermitsiaq, bl.a. vedrørende solvarme om vakuumrør-solfangere.


Der er ydet vejledning til 7 DTU studerende, som har gennemført midtvejsprojekter/eksamensprojekter vedrørende vakuumrørsolfangere til Arktis.
Bilag 1: Artikel optaget i proceedings for ISES SOLAR WORLD CONGRESS, June 14-19, 2003.
THERMAL PERFORMANCE OF EVACUATED TUBULAR COLLECTORS UTILIZING SOLAR RADIATION FROM ALL DIRECTIONS

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Abstract – A prototype collector with parallel-connected evacuated double glass tubes is investigated theoretically and experimentally. The collector has a tubular absorber and can utilize solar radiation coming from all directions.

The collector performance is measured in an outdoor test facility and an efficiency expression for the collector is determined. Further, a theoretical model for calculating the thermal performance is developed. In the model, flat plate collector performance equations are integrated over the whole absorber circumference and the model determines the shade on the tubes as a function of the solar azimuth. Results from calculations with the model are compared with measured results and generally there is a good degree of similarity between the measured and calculated results. However, the comparison shows that the model is suitable only for vertical placed pipes.

The model is used for theoretical investigations on vertically placed pipes at a location in Denmark (Copenhagen, lat. 56°N) and at a location in Greenland (Uummannaq, lat. 71°N). For both locations, the results show that to achieve the highest thermal performance, the tube centre distance must be about 0.2 m and the collector azimuth must be about 45° - 60° towards west. Further, the thermal performance of the evacuated solar collector is compared to the thermal performance of the Arcon HT flat plate solar collector. The Arcon collector is the best performing collector under Copenhagen conditions, whereas the performance of the evacuated tubular collector is highest under the Uummannaq conditions. The reason is that the tubular collector is not optimally tilted in Copenhagen but also that there is much more solar radiation "from all directions" in Uummannaq and this radiation can be utilized with the tubular collector. It is concluded that the collector design is very promising – especially for high latitudes.

1 INTRODUCTION

A new collector design based on evacuated tubular collectors is investigated theoretically and experimentally.

The collector is based on a number of parallel-connected double glass tubes, which are open in both ends. The tubes are annuluses with closed ends and the outside of the inner glass wall is treated with an absorbing selective coating. The collector fluid is floating from bottom to top of the inside of the inner tube where also another closed tube is inserted with the purpose to fill out a part of the tube volume so that less collector fluid is needed. Further, it ensures a high heat transfer coefficient from the inner glass tube to the collector fluid.

Fig. 1 shows the design of the evacuated tubes and Fig. 2 shows the principle of the tube connection.

![Fig. 1. Design of the evacuated tubes. (Top view: Left. Front view: Right)](image1)

![Fig. 2. The tubes connected in a solar collector panel.](image2)

The collector is investigated in an outdoor test facility in order to determine the collector performance experimentally.

For the theoretical investigation of this collector principle, traditional collector theory cannot directly be applied, as the absorbers are tubular. Therefore, to theoretically determine the collector performance a number of conditions must be taken into account, including:

- That solar radiation from all directions can be utilized (also from the "back" of the collector).
- Shadow effects from adjacent tubes.
- Special incident angle modifiers.

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A collector theory for the collector performance, including the above-mentioned considerations, is developed. The theory is compared with the results from the experiments. Based on the theory, the following points are investigated:

- Optimal distance between tubes.
- Optimal collector area.
- Expected yearly thermal performance for different climates.

Finally, a comparison between the thermal performance of a flat plate collector and of the investigated collector is made.

2 COLLECTOR DESIGN

The solar collector panel consists of 14 evacuated tubes placed with a centre distance of 0.867 m. The tubes are connected to two manifold pipes, which are placed in an insulated box. The tubes are 1.6 m long, however, 260.065 m is placed inside the manifold boxes. Thus only 1.47 m is exposed to the sun. The outer diameter of the outer tube is 0.647 m and the inner diameter of the inner tube is 0.327 m. The collector panel is placed on the ground, tilted 45° and facing south. The solar collector areas are described in Table 1 and Fig. 3 shows a photo of the collector. The collector was built by the company SunGain.

Table 1. Solar collector panel areas

<table>
<thead>
<tr>
<th>Gross area [m²]</th>
<th>Outer glass area [m²]</th>
<th>Absorber cross area [m²]</th>
<th>Total absorber area [m²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>18</td>
<td>0.97</td>
<td>0.36</td>
<td>2.36</td>
</tr>
</tbody>
</table>

Fig. 3: The evacuated tubular collector.

3 COLLECTOR PERFORMANCE THEORY

Generally, for a solar collector without reflectors and without parts of the collector reflecting solar radiation to other parts of the collector, the performance equation can be written as:

\[ P_e = P_i + P_f + P_g - P_{loss} \]  

or more detailed described:

\[ P_e = A_e \cdot F_0 + K_e \cdot R_e - G_e \]

\[ + \ A_f \cdot F_s + \ K_{abs} \cdot E_{abs} \cdot G_s \]

\[ + \ A_f \cdot F_s + \ K_{abs} \cdot E_{abs} \cdot G_p \]

\[ - \ A_f \cdot U_e \cdot (T_{col} - T_e) \]

where \( K \) is the incident angle modifier defined as:

\[ K = 1 - \tan(\theta)^2 \]

The incident angle modifiers for diffuse radiation, \( K_{diff} \), and ground reflected radiation, \( K_{geo} \), are evaluated by equation 3 using \( \theta = 3 \).

For tubular collectors, there are several conditions, which make equation (2) more difficult to evaluate. Amongst others the following can be mentioned:

- In flat plate collector theory the areas \( A_p \) and \( A_a \) are typically equal and close to the transparent area. For tubular collectors, however, this is not the case as, depending on the solar azimuth and altitude, parts of the absorber area are exposed to the beam radiation.

- In flat plate collector theory the incident angle modifier, \( K_i \), is independent of the longitudinal and transverse component of the incident angle. The cylindrical geometry in tubular collectors makes it necessary to consider both components.

- In the investigated tubular collector, where the absorber covers the entire inner tube circumference, the radiation coming from the "back" of the collector must be evaluated.

To calculate the thermal performance of the evacuated tubes, the general performance equations (1) and (2) have been integrated over the whole absorber circumference. This means that the tube is divided into small "slices", and each slice is treated as if it was a flat plate collector. In this way, the transverse incident angle modifier is eliminated. For describing the solar radiation on a tubular collector, this method has previously been used by Pyrko 1 (1984).

Integrating over the absorber area, the performance equation can be described as:

\[ P_e = \int (P_i + P_f + P_g - P_{loss}) \, dA \]

In the following each part of equation (4) will be investigated. The investigation is based on a theoretical analysis of a single tube.
Heat loss, $P_{heat}$:
The heat loss can be described as:

$$P_{heat} = \int_{-\infty}^{\infty} A \cdot U \cdot (T_m - T_i) \, d\xi$$

$$= \int_{-\infty}^{\infty} L \cdot T \cdot U \cdot (T_m - T_i) \, d\xi$$

$$= 2\pi L T U (T_m - T_i)$$

Energy from diffuse radiation on collector/tube $P_d$:
The evaluation of the energy contribution from the diffuse radiation is based on an isotropic diffuse model. Thus, the circumsolar diffuse and horizontal brightening contributions are not taken into consideration in this model.

The energy contribution from the diffuse radiation can be written as:

$$P_d = \int_{-\infty}^{\infty} A \cdot F'((\tau_0 \cdot K \cdot G_0) \cdot F_{1-2} \, d\xi$$

$$= 2\pi L T \cdot F'((\tau_0 \cdot K \cdot G_0) \cdot F_{1-2} \, d\xi$$

For a flat plate collector, the view factor from the collector to the sky can be described as:

$$F_{1-2} = \frac{1 + \cos(\beta)}{2}$$

When integrating over the absorber area, the absorber surface tilt, $\beta$, and the absorber surface azimuth, $\xi$, change as illustrated in Fig. 4. This will have an impact on the determination of the incident angle. For instance, when the surface azimuth is 0 (south) the tilt is $\beta$, and when the surface azimuth is $\pm \pi$ (north) the tilt is $\pi - \beta$.

Fig. 4. A tube seen from the top (left) and the side (right). When the performance equation is integrated over the absorber area both the surface azimuth and the surface tilt do change.

The tilt, as a function of the actual absorber azimuth can be written as:

$$\beta = \beta_0 - (1 - \frac{\beta_0}{\pi/2}) \xi$$

$$\xi \geq \frac{\pi}{2}$$

$$\beta = \beta_0 + (1 - \frac{\beta_0}{\pi/2}) \xi$$

(9)

Assuming there are no adjacent tubes, the view factor from the tube to the sky can be described as:

$$F_{c-1} = \int_{-\infty}^{\infty} \left(1 + \cos\left(\beta_1 + \frac{\beta_1}{\pi/2}\xi\right)\right) \frac{1}{2} \, d\xi$$

$$= 0.5$$

In reality, there will be adjacent tubes, which will reduce the view factors to the ground and the sky respectively. This reduction must be taken into consideration.

Fig. 5. Determination of view factors between two tubes.

Fig. 5 shows two adjacent tubes. The view factor $F_{1-2}$ between the absorber of tube 1 and tube 2 can be described as:

$$A_1 \cdot F_{1-2} = \frac{1}{2} \sum \text{length of the crossing curve}$$

$$-\frac{1}{2} \sum \text{length of the non crossing curve}$$

(11)

Here $A_1$ is the absorber perimeter of tube 1. The curves $O_1$ and $O_2$ between the points $P_{1,2}$ and $P_{1,2}$ respectively can be described by:

$$O_1 = \frac{\text{length of the crossing curve}}{2\pi}$$

$$= (\xi_1 - \xi_2) \cdot 2\pi$$

(12)

and

$$O_2 = \frac{\text{length of the non crossing curve}}{2\pi}$$

$$= (\xi_2 - \xi_1) \cdot 2\pi$$

(13)

Here the angles $\xi_1$ and $\xi_2$ are defined by:

$$\xi_1 = \frac{\pi + \xi_1}{C}$$

(14)
\[ x_i = \cos \left( \frac{\pi - x_i}{C} \right) \] \hfill (15)

If the centre of tube 1 has the coordinates \((0,0)\), the coordinates of the points \(P_1\) and \(P_2\) and thus the distance, \(z\), between the two points can be found as:

\[ P_2 = [C + \tau_1 \cos(x_i) + \tau_2 \sin(x_i)] \]
\[ P_1 = [C + \tau_1 \cos(x_i) - \tau_2 \sin(x_i)] \]

\[ z = \sqrt{\left(\frac{C + \tau_1 \cos(x_i) + \tau_2 \sin(x_i)}{\tau_1} \right)^2 + \left(\frac{\tau_1 \sin(x_i) - \tau_2}{\tau_2} \right)^2} \] \hfill (17)

By inserting equations (12), (13), and (17) into equation (11), the view factor from tube 1 to tube 2 can be written as:

\[ F_{12} = \frac{1}{2\pi \tau_1} \left[ (\pi - x_i - x_i) \tau_1 \right] \]
\[ + \left( \frac{C + \tau_1 \cos(x_i) + \tau_2 \sin(x_i)}{\tau_1} \right)^2 \]
\[ + \left( \frac{\tau_1 \sin(x_i) - \tau_2}{\tau_2} \right)^2 \] \hfill (18)

The final view factor from tube to sky, including shading adjacent tubes can thus be described as:

\[ F_{s1} = F_{1s} = F_{12} \] \hfill (19)

### Energy from ground reflected radiation on collector/tube, \(P_g\):

The energy contribution from the ground reflected radiation can be written as:

\[ P_g = \int_{-\pi/2}^{\pi/2} F^*(\gamma_0) G_{\lambda, \gamma} F_{s1} G_{\lambda, \delta} d\delta \]
\[ = 2\pi \tau_1 L F^*(\gamma_0, \lambda) K_{\lambda, \gamma} G_{\lambda, \delta} \int_{-\pi/2}^{\pi/2} F_{s1} d\delta \] \hfill (20)

with

\[ G_{\lambda, \delta} = p_g \cdot (G_1 + G_2) \] \hfill (21)

For a flat plate collector, the view factor from the collector to the ground can be described as:

\[ F_{s1} = \frac{1 - \cos(b)}{2} \] \hfill (22)

In a similar way as for the diffuse radiation, the view factor from the tube to the ground, assuming that there are no neighbouring tubes, can be described as:

\[ \int_{0}^{1} \frac{1 - \cos(\beta - (1 - \frac{\beta}{\pi/2}) \xi)}{2} d\xi \]

\[ F_{s1} = \frac{1}{2\pi} \int_{0}^{2\pi} \frac{1 - \cos(\beta - (1 + \frac{\beta}{\pi/2}) \xi)}{2} d\xi \] \hfill (23)

Including the adjacent shading tubes, the view factor from tube to ground becomes:

\[ F_{s1} = F_{1s} - F_{12} \] \hfill (24)

### Energy from beam radiation on collector/tube, \(P_b\):

The energy contribution from the beam radiation can be written as:

\[ P_b = \int_{\lambda_{low}}^{\lambda_{up}} F^*(\lambda, \gamma_0) G_{\lambda, \gamma} F_{s1} K_{\lambda, \lambda} R_{\lambda} d\lambda \]
\[ = F^*(\lambda, \gamma_0) G_{\lambda, \gamma} \int_{\lambda_{low}}^{\lambda_{up}} K_{\lambda, \lambda} R_{\lambda} d\lambda \] \hfill (25)

Notice that there is now integrated over only a part of the circumference. This is because only part of the absorber surface is exposed to the beam radiation. Assuming that the pipes are placed vertically, Fig. 6 shows the three critical angles, when the solar azimuth, \(\gamma_0\), is between 0 and \(\pi/2\) (Lasco, 2000)).

When the solar azimuth is smaller than the solar azimuth is between \(\gamma_0\) and \(\pi/2\) the tubes are partly shaded. In Table 2 the critical angles are defined for all four quadrants of the circle.

![Critical angles determining the exposed area, when the pipes are placed vertically.](image)
Table 2. Critical angles determining the exposed area, when the pipes are placed vertically.

<table>
<thead>
<tr>
<th>Critical Angle</th>
<th>$N_0$</th>
<th>$N_1$</th>
<th>$N_2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$-\infty &lt; \gamma &lt; -w/2$</td>
<td>$\cos\left(\frac{\pi + \gamma}{C}\right) - \kappa$</td>
<td>$\cos\left(\frac{\pi}{C} \right) - \kappa$</td>
<td>$\cos\left(\frac{\pi}{C} \right) - \kappa$</td>
</tr>
<tr>
<td>$-w/2 &lt; \gamma &lt; w/2$</td>
<td>$-\sin\left(\frac{\gamma}{C}\right)$</td>
<td>$-\sin\left(\frac{\gamma}{C}\right)$</td>
<td>$-\sin\left(\frac{\gamma}{C}\right)$</td>
</tr>
<tr>
<td>$w/2 &lt; \gamma &lt; \infty$</td>
<td>$\cos\left(\frac{\gamma}{C}\right) + \kappa$</td>
<td>$\cos\left(\frac{\gamma}{C}\right) + \kappa$</td>
<td>$\cos\left(\frac{\gamma}{C}\right) + \kappa$</td>
</tr>
</tbody>
</table>

In Fig. 7 - Fig. 10, the angle $\alpha$ represents the tube area exposed to beam radiation, for different solar azimuths. Further, the angles $\xi_{\text{ext}}$ and $\xi_{\text{emp}}$ used in equation (25) are shown.

As a function of the solar azimuth, the angle $\alpha$, $\gamma$, and the integration borders, $\xi_{\text{ext}}$ and $\xi_{\text{emp}}$, are described in Table 3.

Table 3. Angles determining the exposed area, when the pipes are placed vertically and the collector panel azimuth is $0^\circ$.

<table>
<thead>
<tr>
<th>Critical Angle</th>
<th>$\xi_{\text{ext}}$</th>
<th>$\xi_{\text{emp}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$-w/2 &lt; \gamma &lt; 0$</td>
<td>$\pi - \alpha$</td>
<td>$\alpha$</td>
</tr>
<tr>
<td>$-w/2 &lt; \gamma &lt; 0$</td>
<td>$\frac{\pi}{2} + \alpha$</td>
<td>$\alpha$</td>
</tr>
<tr>
<td>$-w/2 &lt; \gamma &lt; 0$</td>
<td>$\pi - \alpha$</td>
<td>$\alpha$</td>
</tr>
<tr>
<td>$-w/2 &lt; \gamma &lt; 0$</td>
<td>$\frac{\pi}{2} + \alpha$</td>
<td>$\alpha$</td>
</tr>
</tbody>
</table>

Fig. 7. Relationship between the solar azimuth, $\gamma$, the beam exposure angle, $\alpha$, and the integration borders $\xi_{\text{ext}}$ and $\xi_{\text{emp}}$ for $-w/2 < \gamma < 0$, when the pipes are placed vertically.

Fig. 8. Relationship between the solar azimuth, $\gamma$, the beam exposure angle, $\alpha$, and the integration borders $\xi_{\text{ext}}$ and $\xi_{\text{emp}}$ for $0 < \gamma < w/2$, when the pipes are placed vertically.

Fig. 9. Relationship between the solar azimuth, $\gamma$, the beam exposure angle, $\alpha$, and the integration borders $\xi_{\text{ext}}$ and $\xi_{\text{emp}}$ for $0 < \gamma < w/2$, when the pipes are placed vertically.
The incident angle, $\beta$, and the geometric factor, $R_0$

In equation (25), the incident angle, $\theta$, and the geometric factor, $R_0$, still need to be addressed. As earlier mentioned, when integrating over the absorber area, both the surface tilt and the surface azimuth change will have an impact on both $K_0$ and $R_0$. The incident angle, $\theta$, can be described as (Duffie J.A. and Beckman W.A. (1991)):

$$\cos(\theta) = \sin(\delta)\sin(\phi)\cos(\beta)$$

$$+ \cos(\delta)\cos(\phi)\sin(\beta)$$

$$+ \cos(\delta)\sin(\phi)\cos(\gamma)\sin(\beta)$$

Here the tilt, $\beta$, is described in equation 8 and 9 as a function of the absorber surface azimuth, $\gamma$.

The geometric factor, $R_0$, can be described as (Duffie J.A. and Beckman W.A. (1991)):

$$R_0 = \frac{\cos(\theta)}{\cos(\theta)_{corrected}}$$

$$= \frac{\cos(\theta)}{\cos(\delta)\cos(\phi)\cos(\alpha) + \sin(\delta)\sin(\phi)}$$

Solving the performance equation:

All the parameters involved in the performance equation (1) and (2) have now been described in the equations (3) - (27).

In order to evaluate the performance of the tubular collector on a yearly basis, the above theory was implemented into a numerical program. All the integrals could be solved analytically, except the integral in equation (25), which was solved by using the trapezoidal formula for solving integrals numerically. 360 integration steps were used in the numerical integration.

The program is based on weather data with hourly data for global radiation, diffuse radiation on horizontal and outdoor temperature. However, the incident angle and thus the collector performance were calculated every half hour.

4 MEASUREMENTS

The performance of the collector was measured in an outdoor test facility where the inlet temperature, the outlet temperature and the volume flow rate was measured. The temperatures were measured with copper-constantan thermocouples (Type TT) and the volume flow rate was measured with a HFGQ flow meter. A 31% glycol/water mixture was used in the solar collector loop. Further, the global radiation and the diffuse radiation on horizontal were measured with two Kipp&Zonen CM5 pyranometers.

The power from the solar collector was determined from the measurements by:

$$P_i = V P C_p (T_{i,in} - T_{i,out})$$

Further, an efficiency expression for the solar collector panel was determined from the measurements. Only
measurements where the below mentioned conditions were fulfilled were used by determination of the efficiency expression:

- \( G_{	ext{day}} \) was larger than 800 W/m²
- The incidence angle on a south facing 45° tilted surface was smaller than 20°
- The diffuse radiation was less than 22% of the global radiation
- Stationary conditions during at least 15 min.

The solar collector efficiency, based on the absorber area and on the tube cross section area, could thus be determined by:

Absorber area:
\[
\eta = \frac{P_{	ext{in}}}{G_{	ext{day}} A_{	ext{abs}}}
\]

Tube cross section area:
\[
\eta = \frac{P_{	ext{in}}}{G_{	ext{day}} A_{	ext{tube}}}
\]

(29)

Fig. 12 shows the measured efficiency expressions based on the absorber area and the tube cross section area, respectively.

In Fig. 14 the measured and calculated collector outlet temperature are compared. Generally, it can be seen that there is a good degree of similarity between the measured and calculated temperatures. However, it can also be seen that the model has a tendency to calculate too low outlet temperatures in the morning and in the evening. This tendency is especially clear for the last 3-4 days in Fig. 14. The reason for this problem is that the collector is tilted 45°, whereas the model is developed only for horizontally or vertically placed pipes. As earlier described, with a tilted collector both the solar azimuth and the solar altitude will influence the size of the shadowed area of the pipes. The larger differences between the calculated and the measured outlet temperature occur especially when the solar altitude is dominating the size of the area exposed to beam radiation. In the model the shadow calculations are based on only the solar azimuth (for vertical pipes) or on only the solar altitude (for horizontal pipes).

Table 5. Data describing the collector in the model.

<table>
<thead>
<tr>
<th>No. of Pipes</th>
<th>( L ) [m]</th>
<th>( t_1 ) [°C]</th>
<th>( t_2 ) [°C]</th>
<th>( C ) [kW/K]</th>
<th>( F )</th>
<th>( \tau_{	ext{th}} )</th>
<th>( n )</th>
</tr>
</thead>
<tbody>
<tr>
<td>14</td>
<td>1.47</td>
<td>0.0255</td>
<td>0.0165</td>
<td>2.09</td>
<td>2.09</td>
<td>0.059</td>
<td>3.0</td>
</tr>
</tbody>
</table>

Fig. 12. Measured efficiency expressions based on the absorber area and the tube cross section area.

Fig. 13. Total solar irradiance on the collector plane, ambient temperature, collector inlet temperature and collector volume flow rate during the test period.

Fig. 14. Measured and calculated outlet temperature for the test period.

5 COMPARING THE MODEL WITH MEASUREMENTS

A period of 13 days (17/5-30/5 2003) has been selected for validating the computer model of the collector. In Fig. 13 the total solar irradiance on the collector plane, the ambient temperature, the inlet temperature to the collector and the volume flow rate in the collector are shown. These values are used as input data to the model. The necessary data for describing the collector are shown in Table 5. The heat loss coefficient, \( k_0 \), was determined from the efficiency measurements (Fig. 12) and split into two parts for the evacuated tubes and the manifold pipes respectively. \( F' \) was calculated from theory (Duffie J.A. and Beckman W.A. (1991)), (Incropera F.P. and de Witt D.P. (1990)) and \( t_0 \) and \( \alpha \) were calculated with a simulation program for determining optical properties (Svendsen S. and Jønne F.F. (1994)).
6 THEORETICAL INVESTIGATIONS

In this section, the model will be used for theoretical investigations. Only vertically placed pipes will be analysed, as the model is yet unfit to calculate on tilted pipes. The collector performance is investigated for two locations:

- Copenhagen, Denmark, lat. 56°N, yearly average ambient temperature: 7.8°C. Weather data: DRY (Lund H. (1995)).
- Uummannaq, Greenland, lat. 71°N, yearly average ambient temperature: -4.2°C. Weather data: TRY (Kraft et al. (2002)).

Fig. 15 shows the sum of the solar radiation on the front and the back of vertical planes with different orientations. For Copenhagen, there is symmetry around 0° (south) whereas for Uummannaq the minimum solar radiation on the plane is around -30° (towards east). The reason for this asymmetrical behaviour is that there is a mountain east of Uummannaq, which reduces the radiation coming from east.

As a function of the collector azimuth, Fig. 16 shows the utilized solar energy per tube in a panel assuming a tube centre distance of 0.657m and a constant collector fluid temperature throughout the year of 50°C. For both Uummannaq and Copenhagen, there is an optimum at a collector azimuth of about 45°-60°. The results are caused both by the distribution of solar radiation and by higher afternoons temperatures.

Fig. 17 shows the utilized solar energy per tube as a function of the tube centre distance for a collector fluid temperature of 50°C. For both locations, the utilized energy increases for tube centre distances up to 0.2 m, which is due to reduced shaded areas. For larger distances the utilized energy decreases again, due to the increasing heat loss from the manifold pipes.

For the two locations, Fig. 18 and Fig. 19 show the utilized solar energy per tube as a function of the collector azimuth and the tube centre distance for a collector fluid mean temperature of 50°C. Here it can be seen that the tendencies in Fig. 16 are true for tube distances in the range of 0.047 m - 0.3 m.

Finally Fig. 20 and Fig. 21 show the utilized solar energy per m² as a function of the collector fluid mean temperature assuming a tube centre distance of 0.047 m and a collector azimuth of 50°. Further, the thermal performance of the newest (Vejle N.K. (2003)) Arcon HT collector is shown. The Arcon HT collector is facing south and tilted 45° in Copenhagen. In Uummannaq the Arcon HT collector is facing south and tilted 60°.

It is interesting that the Arcon collector is the best performing collector under Copenhagen conditions, whereas the thermal performance of the evacuated tubular collector based on the outer tube cross section area, is the highest under the Uummannaq conditions. The reason for the change in the ranking of the collectors is that the tubular collector is not optimally tilted in Copenhagen but also that there is much more solar radiation "from all directions" in Uummannaq and this radiation can be utilized with the tubular collector.

![Fig. 15. Solar radiation on the front and the back of vertical planes with different orientations.](image1)

![Fig. 16. Utilized solar energy per tube as a function of the collector azimuth. Tube centre distance=0.657 m.](image2)

![Fig. 17. Utilized solar energy per tube as a function of the tube centre distance for a collector fluid mean temperature of 50°C.](image3)
Fig. 18. Ummannaq: Utilized solar energy per tube as a function of the collector azimuth and the tube centre distance. Collector fluid mean temperature: 50°C.

Fig. 19. Copenhagen: Utilized solar energy per tube as a function of the collector azimuth and the tube centre distance. Collector fluid mean temperature: 50°C.

Fig. 20. Copenhagen: Utilized solar energy per m² as a function of the collector fluid mean temperature at a tube distance of 0.047 m.

Fig. 21. Ummannaq: Utilized solar energy per m² as a function of the collector fluid mean temperature at a tube distance of 0.047 m.

7 DISCUSSION AND CONCLUSION

A prototype collector with parallel-connected evacuated double glass tubes is investigated theoretically and experimentally. The collector has a tubular absorber and can thus utilize solar radiation coming from all directions. The collector performance is measured in an outdoor test facility and an efficiency expression for the collector is determined from the measurements.

Further, a theoretical model for calculating the thermal performance is developed. In the model, the flat plate collector performance equations have been integrated over the whole absorber circumference. In this way, the transverse incident angle modifier is eliminated. Also, the model determines the shades on the tubes as a function of the solar azimuth in order to calculate the energy from the beam radiation correctly. The calculation of the energy from the diffuse and ground reflected radiation is based on an isotropic diffuse sky model.

The calculations with the model is compared with measured results and generally, there is a good degree of similarity between the measured and calculated results. However, the comparison also shows that the model is suitable only for vertical placed pipes.

The model is used for theoretical investigations on vertically placed pipes placed in Copenhagen, Denmark and in Ummannaq, Greenland. For both locations, the results show that to achieve the highest thermal performance the tube distance should be about 0.2 m and the collector azimuth should be about 45°-60° towards west. The thermal performance of the evacuated solar collector is also compared to the thermal performance of the Arcon HT flat plate solar collector. These results
show that Arcon collector is the best performing collector under Copenhagen conditions, whereas the performance of the evacuated tubular collector is the highest under the Ummannaaq conditions. The reason is that the tubular collector is not optimally tilted in Copenhagen but also that there is much more solar radiation "from all directions" in Ummannaaq and this radiation can be utilized with the tubular collector. It is therefore concluded that the collector design is very promising — especially for high latitudes.

The theoretical results presented in this paper are based on a collector model, which needs to be further developed. First of all, the model must be able to calculate on tilted pipes. The reflections between the pipes must be included in the model and also an anisotropic diffuse sky model should be included. This extended model must of course be thoroughly validated with measurements.

Finally, though the collector design seems very promising the collector reliability and durability must be examined before any final conclusions can be drawn.

ACKNOWLEDGEMENT

The study is financed by VILLUM KANN RASMUSSEN FOUNDATION.

NOMENCLATURE

LATIN SYMBOLS:

- $\theta$: Incident angle modifier constant
- $A_1$: Absorber perimeter length [m]
- $A_a$: Absorber area [m$^2$]
- $A_{ab}$: Absorber area exposed to beam radiation [m$^2$]
- $A_{ad}$: Absorber area exposed to diffuse radiation [m$^2$]
- $a_{tube}$: Tube cross section area [m$^2$]
- $C_C$: Collector fluid heat capacity [J/kg K]
- $F_r$: Collector efficiency factor [-]
- $F_{12}$: View factor from tube 1 to tube 2 [-]
- $F_{2g}$: View factor from tube to ground [-]
- $F_{49}$: View factor from tube to sky [-]
- $F_{4g}$: View factor from tube to ground without adjacent tube [-]
- $F_{4s}$: View factor from tube to sky without adjacent tube [-]
- $G_b$: Beam radiation on horizontal [W/m$^2$]
- $G_d$: Diffuse radiation on horizontal [W/m$^2$]
- $G_{	ext{global}}$: Global radiation [W/m$^2$]
- $G_{	ext{total}}$: Total radiation on collector plane [W/m$^2$]
- $S_u$: Solar altitude [rad]
- $I_{	ext{c}}$: Collector heat loss coefficient [W/m$^2$ K]
- $I_{	ext{c}}$: Incident angle modifier for beam radiation [-]
- $I_{	ext{d}}$: Incident angle modifier for diffuse radiation [-]
- $I_{	ext{g}}$: Incident angle modifier for ground reflected radiation [-]
- $L$: Pipe length [m]
- $L_1$: Absorber length [m]
- $L_2$: Absorber length [m]
- $P_a$: Energy from beam radiation on collector/tube [W]
- $P_d$: Energy from diffuse radiation on collector/tube [W]
- $P_g$: Energy from ground reflected radiation on collector/tube [W]
- $P_{	ext{total}}$: Energy from total radiation on collector/tube [W]

GREEK SYMBOLS:

- $\beta$: Collector tilt [rad]
- $\beta_a$: Absorber surface tilt when absorber surface azimuth is zero [rad]
- $\eta$: Solar collector efficiency [-]
- $\gamma$: Solar azimuth [rad]
- $\phi$: Incidence angle [rad]
- $\rho$: Collector fluid density [kg/m$^3$]
- $\theta$: Incident angle [rad]
- $\zeta$: Effective transmittance-absorptance product [-]
- $\zeta$: Absorber surface azimuth [rad]
- $\zeta$: Integration border [rad]
- $\zeta$: Integration border [rad]
- $\zeta$: Latitude [rad]
- $\omega$: Solar time [rad]

REFERENCES


NEW TRNSYS MODEL OF EVACUATED TUBULAR COLLECTOR WITH CYLINDRICAL ABSORBER
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Introduction
A new collector design based on parallel-connected double glass evacuated tubes has previously been investigated theoretically and experimentally (Shah, L.J. & Furbo, S. (2004)). The tubes were annuluses with closed ends and the outside of the inner glass wall was treated with a selective coating. The collector fluid was floating inside the inner tube where also another closed tube was inserted so less collector fluid was needed. The collector design made utilization of solar radiations from all directions possible. Fig. 1 shows the design of the evacuated tubes and the principle of the tube connection.

The investigations resulted in a validated collector model that could calculate the yearly thermal performance of the collector based on hourly weather data. The advantages of the model were that shadows, the solar radiation and the incidence angle modifier for each tube were precisely determined for all solar positions, including solar positions on the “back” of the collector. However, the model could be improved further as the model was only valid for vertically tilted pipes and as the model was not developed for a commonly used simulation program.

In the present paper, the theory is further developed so it can simulate solar collector panels of any tilt and based on the theory a new TRNSYS (Klein, S.A. et al. (1996).) collector type is developed. This model is validated with the measurements from outdoor experiments.

TRNSYS simulations of the yearly thermal performance of a solar heating plant based on the evacuated solar collectors are carried out and among other things it is investigated how the distance between tubes and the collector tilt influences the yearly thermal performance. The calculations are carried out for two locations: Copenhagen, Denmark, lat. 56°N, and Uummannaq, Greenland, lat. 71°N.

Further, the results are compared to the calculated thermal performance of the solar heating plant based on traditional flat plate collectors.
Collector performance theory for tubular absorbers

In Shah, L.J. & Furbo, S. (2003) and Shah, L.J. & Furbo, S. (2004), a theoretical model for calculating the thermal performance of evacuated collectors with tubular absorbers was developed. The principle in the model was that flat plate collector performance equations were integrated over the whole absorber circumference. In this way, the transverse incident angle modifier was eliminated. The model was valid only for vertically tilted pipes.

In this section, the principle of the model will shortly be summarized. Further, the newest development that improves the model to be able to also take tilted pipes into calculation will be described.

Generally, for a solar collector without reflectors and without parts of the collector reflecting solar radiation to other parts of the collector, the performance equation can be written as:

\[ P_u = P_b + P_d + P_{gr} - P_{loss} \quad (1) \]

or more detailed described:

\[ P_u = A_e \cdot F'_{e_c}(\tau \alpha) \cdot K_{e_c, d} \cdot G_e + A_e \cdot F'_{e_c}(\tau \alpha) \cdot K_{e_c, gr} \cdot F_{e,g} \cdot G_{gr} - A_e \cdot U_{l,c}(T_m - T_a) \quad (2) \]

where \( K_0 \) is the incident angle modifier defined as:

\[ K_0 = 1 - \tan^2 \left( \frac{\theta}{2} \right) \quad (3) \]

The incident angle modifiers for diffuse radiation, \( K_{\theta, d} \), and ground reflected radiation, \( K_{\theta, gr} \), are evaluated by equation 3 using \( \theta = \pi/3 \).

To calculate the thermal performance of the evacuated tubes, the general performance equations (1) and (2) have been integrated over the whole absorber circumference. This means that the tube is divided into small “slices”, and each slice is treated as if it was a flat plate collector. In this way, the transverse incident angle modifier is eliminated. For describing the solar radiation on a tubular geometry, this method has previously been used by Pyrko J. (1984).

Integrating over the absorber area, the performance equation can be described as:

\[ P_u = \int \left( P_b + P_d + P_{gr} - P_{loss} \right) d\xi \quad (4) \]

where,

\[ P_{loss} = \int L \cdot r_p \cdot U_p \cdot (T_m - T_a) \cdot d\xi = 2 \pi L \cdot r_p \cdot U_p \cdot (T_m - T_a) \quad (5) \]

\[ P_d = \int A_e \cdot F'_{e_c}(\tau \alpha) \cdot K_{e_c, d} \cdot G_{e,c} \cdot d\xi = 2 \pi \tau_p \cdot L \cdot F'_{e_c}(\tau \alpha) \cdot K_{e_c, d} \cdot G_{e,c} \cdot F_{e-c} \cdot d\xi \quad (6) \]

\[ P_{gr} = \int A_e \cdot F'_{e_c}(\tau \alpha) \cdot K_{e_c, gr} \cdot F_{e,g} \cdot G_{gr} \cdot d\xi = 2 \pi \tau_p \cdot L \cdot F'_{e_c}(\tau \alpha) \cdot K_{e_c, gr} \cdot G_{gr} \cdot F_{e-g} \cdot d\xi \quad (7) \]

\[ G_{gr} = \rho_{gr} \cdot (G_e + G_d) \quad (8) \]

\[ F_{e-c} = 0.5 - F_{c-1} \quad (9) \]

\[ F_{e-g} = 0.5 - F_{g-1} \quad (10) \]
Power from beam radiation on collector/tube, $P_b$:
The power contribution from the beam radiation can be written as:

$$P_b = \frac{1}{2} \int \int \int F'(\tau \alpha) G_b \cdot A_b \cdot K_b \cdot R_b \cdot d\xi$$

Notice that there is now integrated over only a part of the circumference. This is because only part of the absorber surface is exposed to the beam radiation due to shadows from the neighbour tube. The task is now to determine the size of this area, thus determining the size and position of the shadowed area. In vector notation, the position of the sun can be described by:

$$\vec{S} = \begin{pmatrix} \sin \theta \cdot \cos \gamma \\ \sin \theta \cdot \sin \gamma \\ \cos \theta \end{pmatrix}$$

and a “cross section circle” (see Fig. 2) on the absorber of one tube can be described by:

$$\vec{N} = r_p \begin{pmatrix} \cos \left( \frac{\pi}{2} - \beta \right) \cos \gamma_0 \\ \sin \gamma_1 \\ \sin \left( \frac{\pi}{2} - \beta \right) \sin \gamma_0 \end{pmatrix}$$

Fig. 3 shows an example where a part of one tube is shaded and a part is exposed to beam radiation. In order to determine the size of the area exposed to beam radiation, the points $P_0$ and $P_1$ must be determined.

Since $P_0$ is located where the solar vector and the tube vector are at right angles to each other, $P_0$, described by the angle $\gamma_0$, can be determined by the scalar product of the two vectors:

$$\vec{S} \cdot \vec{N} = |\vec{S}| |\vec{N}| \cos \left( \frac{\pi}{2} \right) = 0$$

$$\Rightarrow \sin \theta \cdot \cos \gamma_1 \cdot \cos \left( \frac{\pi}{2} - \beta \right) \cos \gamma_0 + \sin \theta \cdot \sin \gamma_1 \cdot \sin \gamma_0 + \cos \theta \cdot \sin \left( \frac{\pi}{2} - \beta \right) \sin \gamma_0 = 0$$

$$\gamma_0 = -\arctan \left( \frac{\sin(\theta_1) \cdot \cos(\gamma_1) \cdot \cos \left( \frac{\pi}{2} - \beta_1 \right) + \cos(\theta_1) \cdot \sin \left( \frac{\pi}{2} - \beta_1 \right)}{\sin(\theta_1) \cdot \sin(\gamma_1)} \right)$$

**Fig. 2:** The solar vector, $\vec{S}$, and the tube vector, $\vec{N}$.

**Fig. 3:** Illustration of the shaded area and the area exposed to beam radiation.
Since the equation for $\gamma_0$ involves the tangens function, the equation will return two solutions. Based on information on the position of the sun, the correct solution is found.

The point $P_1$, described by the angle $\gamma_1$, can be determined from the following equations (15), (16) and (17). A graphical illustration of symbols used in the equations can be seen in Fig. 3 and Fig. 4.

$$P_1 = \begin{pmatrix} x_1 \\ y_1 \\ z_1 \end{pmatrix} = \begin{pmatrix} x_n \\ y_n \\ z_n \end{pmatrix} + \begin{pmatrix} \sin \theta_s \cos \gamma_s \\ \sin \theta_s \sin \gamma_s \\ \cos \theta_s \end{pmatrix} T \tag{15}$$

$$P_1 = \begin{pmatrix} x_1 \\ y_1 \\ z_1 \end{pmatrix} = \begin{pmatrix} x_n \\ 0 \\ z_n \end{pmatrix} + r_p \begin{pmatrix} \cos \left( \frac{\pi}{2} - \beta_s \right) \cos \gamma_1 \\ \sin \gamma_1 \\ \sin \left( \frac{\pi}{2} - \beta_s \right) \sin \gamma_1 \end{pmatrix} \tag{16}$$

$$x_n = -z_n \tan \left( \frac{\pi}{2} - \beta_s \right) \tag{17}$$

Equations (15) (16) and (17) together give four equations to the four unknowns: $T$, $\gamma_1$, $x_n$ and $z_n$. Solving for $\gamma_1$ gives:

$$\gamma_1 = \arctan 2 \left[ \frac{K_1 + 0.5 \frac{K_2}{K_2^2 + K_3^2} \left( -2 K_1 K_2 + 2 K_4^{0.5} \right)}{K_3} - 0.5 \frac{K_2}{K_2^2 + K_3^2} \left( -2 K_1 K_2 + 2 K_4^{0.5} \right) \right] \tag{18}$$

or

$$\gamma_1 = \arctan 2 \left[ \frac{K_1 + 0.5 \frac{K_2}{K_2^2 + K_3^2} \left( -2 K_1 K_2 - 2 K_4^{0.5} \right)}{K_3} - 0.5 \frac{K_2}{K_2^2 + K_3^2} \left( -2 K_1 K_2 - 2 K_4^{0.5} \right) \right]$$

where

$$K_1 = \frac{x_n}{\tan \left( \frac{\pi}{2} - \beta_s \right)} - \frac{y_n}{\tan \left( \frac{\pi}{2} - \beta_s \right) \tan (\gamma_s - \gamma_f) - \tan (\theta_s) \sin (\gamma_s - \gamma_f) + z_n}$$

$$K_2 = \frac{C}{\tan \left( \frac{\pi}{2} - \beta_s \right) \tan (\gamma_s - \gamma_f) + \tan (\theta_s) \sin (\gamma_s - \gamma_f)}$$

$$K_3 = C \left( \cos^2 \left( \frac{\pi}{2} - \beta_s \right) + \frac{1}{\tan (\theta_s) \sin (\gamma_s - \gamma_f)} \right)$$

$$K_4 = K_1^4 - K_1^2 K_2^2 + K_2^2 K_3^2$$

From equation (18) it appears that there are two solutions for $\gamma_1$. Based on information on the position of $\gamma_0$, the correct solution is found.
The incident angle, $\theta$, and the geometric factor, $R_b$: 

The incident angle, $\theta$, can be described as:

$$\cos(\theta) = \sin(\theta_s) \cos(\gamma_s - \gamma_t) \cos(\frac{\pi}{2} - \beta_s) \cos(\theta_{actual}) + \sin(\theta_s - \gamma_t) \sin(\gamma_{actual}) + \cos(\theta_s) \sin(\frac{\pi}{2} - \beta_s) \cos(\gamma_{actual})$$

(20)

The geometric factor, $R_b$, can be described as (Duffie J.A. and Beckman W.A. (1991)).

$$R_b = \frac{\cos(\theta)}{\cos(\theta_s)}$$

(21)

Solving the performance equation:

In order to evaluate the performance of the tubular collector on a yearly basis, the above theory is implemented into a Trnsys type. All the integrals can be solved analytically, except the integral in equation (11), which is solved by using the trapezoidal formula for solving integrals numerically. 360 integration steps are used in the numerical integration. Taking the collector capacity into account, the collector outlet temperature is evaluated by:

$$P_a = V_p C_{p} (T_{out,hot} - T_{in,cold}) + \frac{C_{p,cal}(T_{in}^{cal} - T_{in})}{\Delta t}$$

(22)

Measurements and model validation

The thermal performance of the collector described in Table 1 was measured in an outdoor test facility where the inlet temperature, the outlet temperature and the volume flow rate was measured. The temperatures were measured with copper-constantan thermocouples (Type TT) and the volume flow rate was measured with a HGQ1 flow meter. A 31% glycol/water mixture was used in the solar collector loop. Further, the global radiation and the diffuse radiation on horizontal were measured with two Kipp&Zonen CM5 pyranometers.

The collector performance was measured for two different tilts: 45° and 90° (both facing south). A period of 11 days (17/5-28/5 2003) has been selected for validating the Trnsys model for the collector at 45° and a period of 7 days (12/8-19/8 2003) has been selected for validating the Trnsys model for the collector at 90°.

The necessary data for describing the collector are shown in Table 1. The heat loss coefficient, $k_0$, was determined from efficiency measurements (Shah, L.J. & Furbo, S. (2004)) and split into two parts for the evacuated tubes and the manifold pipes respectively. $F'$ was calculated from theory (Duffie J.A. and Beckman W.A. (1991)), (Incropera F.P. and de Witt D.P. (1990)) and $(\tau a)_b$ and $a$ were calculated with a simulation program for determining optical properties (Svendsen S. and Jensen F.F. (1994)).

<table>
<thead>
<tr>
<th>No. of pipes</th>
<th>[-]</th>
<th>14</th>
</tr>
</thead>
<tbody>
<tr>
<td>L</td>
<td>[m]</td>
<td>1.47</td>
</tr>
<tr>
<td>$r_c$</td>
<td>[m]</td>
<td>0.0235</td>
</tr>
<tr>
<td>$r_p$</td>
<td>[m]</td>
<td>0.0185</td>
</tr>
<tr>
<td>C</td>
<td>[m]</td>
<td>0.067</td>
</tr>
<tr>
<td>$k_0$</td>
<td>[W/m²K]</td>
<td>2.09</td>
</tr>
<tr>
<td>$F'$</td>
<td>[-]</td>
<td>0.98</td>
</tr>
<tr>
<td>$(\tau a)_b$</td>
<td>[-]</td>
<td>0.856</td>
</tr>
<tr>
<td>a</td>
<td>[-]</td>
<td>3.8</td>
</tr>
<tr>
<td>$C_{collector}$</td>
<td>[kJ/K/tube]</td>
<td>1.9</td>
</tr>
</tbody>
</table>

Table 1: Data describing the collector in the model.
In Fig. 6 the measured and calculated collector outlet temperatures are compared. It can be seen that there is a good degree of similarity between the measured and calculated temperatures. Further Fig. 5 shows the measured and calculated collector performance for the two periods. The difference between the measured and calculated performance lies within the measuring inaccuracy of 4%.

Simulation of solar heating plants

Model description:
A model of a solar heating plant is built in TRNSYS. The collector array consists of 100 rows where the distance between the rows is assumed to be so large that the shadows between the rows have negligible influence on the collector performance. The energy consumption of a town is defined by a water mass flow rate, a return temperature and a flow temperature of 80°C.

If the temperature from the solar heat exchanger is above 80°C the temperature is mixed down to 80°C with at three-way valve. If the temperature from the solar heat exchanger is below 80°C, an auxiliary boiler plant heats up the district heating water to 80°C.

An illustration of the TRNSYS model can be seen in Fig. 7 and Fig. 8 shows the mass flow rate and a flow and return temperature throughout the year for the district heating net of the town. The annual heat consumption of the town is about 32500 MWh.

The collector performance is investigated for two locations:

Fig. 5: Measured and calculated collector performance for the test periods

Fig. 6: Measured and calculated outlet temperature for the test periods

Fig. 7: Schematic illustration of the TRNSYS model

Fig. 8: Assumed flow rate and temperatures in the district heating net.
• Uummannaq, Greenland, lat. 71°N, yearly average ambient temperature: -4.2°C. Weather data: TRY (Kragh J. et al (2002)).

Tube distance, collector tilt and collector orientation
The optimum tube centre distance, collector tilt and orientation with respect the thermal performance per tube is investigated for the two locations. The gross collector area is assumed to be constant in the solar heating plant. Consequently, there are more tubes in the collector area when the tube distance is small than when the tube distance is large. Table 2 shows how the collector orientation, the tilt and the tube distance are varied.

<table>
<thead>
<tr>
<th>Collector azimuth [°]</th>
<th>-90 (east), 75, 60, 45, 30, 15, 0, 15, 30, 45, 60, 75, 90 (west)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Collector tilt [°]</td>
<td>15, 30, 45, 60, 75, 89</td>
</tr>
<tr>
<td>Tube centre distance [m]</td>
<td>0.048, 0.077, 0.107, 0.137, 0.167, 0.197</td>
</tr>
</tbody>
</table>

Table 2: Overview of the parameter variations performed with the model.

Fig. 9 and Fig. 10 show the thermal performance per tube for Copenhagen and Uummannaq respectively. The figures clearly show how the thermal performance increases with increasing tube centre distance. The increase is mainly caused by less shadow from the adjacent tubes but also by the differences in the average temperature level of the collector. The figures also show that the optimum tilt and orientation is about 45° south for Copenhagen and about 60° south for Uummannaq.

![Fig. 9: The thermal performance per tube as a function of the tube centre distance, collector tilt and orientation (Copenhagen).](image-url)
Comparison with a flat-plate collector

Still considering the solar heating plant, the thermal performance of the evacuated tubular collector is compared to the thermal performance of the new Arcon HT collector. The collectors are facing south and tilted 45° in Copenhagen. In Ummannaq the collectors are facing south and tilted 60°.

It can be difficult to compare the thermal performances of flat-plate collectors and tubular collectors as the effective area of a flat plate collector typically is defined as the transparent area of the glass cover and the effective area of a tubular collector can be defined in many ways. In the present comparison, the tubes are placed close together so that there is no air-gap between the tubes and the outer tube cross-section area (=L·2·π·r·N) directly corresponds to the transparent area of a flat-plate collector.

Fig. 10: The thermal performance per tube as a function of the tube centre distance, collector tilt and orientation (Ummannaq).

Fig. 11: Thermal performance pr. m² collector as a function of the solar fraction of the solar heating plant.

Solar fraction = 1 - \( \frac{Q_{\text{auxil}}}{Q_{\text{town}}} \)  

(23)

First of all, the figure shows that the tubular collector has the highest thermal performance for both locations. Further, it can be seen that the Ummannaq curves decreases more
rapidly with increasing solar fractions. This is due to the lower air temperature in Uummannaq.
The figure also shows that the ARCON HT collector has a better thermal performance in Copenhagen than in Uummannaq, whereas the tubular collector performs best in Uummannaq. The main reason for the result is that there is much more solar radiation “from all directions” in Uummannaq and this radiation can better be utilized with the tubular collector.

Conclusions
A new TRNSYS collector model for evacuated tubular collectors with tubular absorbers is developed. The model is based on traditional flat plate collector theory, where the performance equations have been integrated over the whole absorber circumference. On each tube the model determines the size and position of the shadows caused by the neighbour tube as a function of the solar azimuth and zenith. This makes it possible to calculate the energy from the beam radiation.
The thermal performance of an all glass tubular collector with 14 tubes connected in parallel is investigated theoretically with the model and experimentally in an outdoor collector test facility. Calculations with the new model of the tubular collector vertically placed and tilted 45° is compared with measured results and a good degree of similarity between the measured and calculated results is found.
Further, the collector model is used in a model of a solar heating plant and a sensitivity analysis of the tube centre distance, collector tilt and orientation with respect the thermal performance per tube is investigated for the two locations Copenhagen (Denmark) and Uummannaq (Greenland). The results show that the optimum tilt and orientation is about 45° south for Copenhagen and about 60° south for Uummannaq.
Finally, the thermal performance of the evacuated tubular collector is compared to the thermal performance of the newest Arcon HT collector. Here, the results show that the tubular collector has the highest thermal performance for both Uummannaq and Copenhagen. This analysis also illustrates the differences in the thermal behaviour of the two collector types: The ARCON HT collector has a higher thermal performance in Copenhagen than in Uummannaq, whereas the tubular collector performs best in Uummannaq compared to Copenhagen. The main reason for the result is that there is much more solar radiation “from all directions” in Uummannaq and this radiation can better be utilized with the tubular collector than with the flat plate collector.

References


**Nomenclature**

**LATIN SYMBOLS:**

- \(a\): Incident angle modifier constant
- \(A_a\): Absorber area \([m^2]\)
- \(A_b\): Absorber area exposed to beam radiation \([m^2]\)
- \(C\): Tube centre distance \([m]\)
- \(C_p\): Collector fluid heat capacity \([J/(kg\cdot K)]\)
- \(C_{collector}\): Collector panel heat capacity incl. fluid \([kJ/K/Tube]\)
- \(F'\): Collector efficiency factor [-]
- \(F_{1,2}\): View factor from tube 1 to tube 2 [-]
- \(F_{c,g}\): View factor from tube to ground [-]
- \(F_{c,s}\): View factor from tube to sky [-]
- \(G_b\): Beam radiation on horizontal \([W/m^2]\)
- \(G_d\): Diffuse radiation on horizontal \([W/m^2]\)
- \(G_{gr}\): Ground reflected radiation on horizontal \([W/m^2]\)
- \(K_0\): Collector heat loss coefficient \([W/m^2K]\)
- \(K_1\): Help variable [-]
- \(K_2\): Help variable [-]
- \(K_3\): Help variable [-]
- \(K_4\): Help variable [-]
- \(K_5\): Incident angle modifier for beam radiation [-]
- \(K_{a,d}\): Incident angle modifier for diffuse radiation [-]
- \(K_{a,gr}\): Incident angle modifier for ground reflected radiation [-]
- \(\bar{N}\): Tube vector [-]
- \(N\): Number of tubes [-]
- \(L\): Pipe length \([m]\)
- \(P_t\): Energy from beam radiation on collector/tube \([W]\)
- \(P_d\): Energy from diffuse radiation on collector/tube \([W]\)
- \(P_{gr}\): Energy from ground reflected radiation on collector/tube \([W]\)
- \(P_{loss}\): Heat loss from collector/tube \([W]\)
- \(P_u\): Useful energy from collector/tube \([W]\)
- \(Q_{auxiliar}\): Energy supplied from the boiler plant \([kWh]\)
- \(Q_{town}\): Energy supplied to the town \([kWh]\)
- \(R_0\): Geometric factor; irradiance on a tilted surface divided by irradiance on a horizontal surface [-]
- \(r_0\): Outer glass tube radius \([m]\)
- \(r_p\): Absorber radius \([m]\)
- \(S\): Solar vector [-]
- \(T_a\): Ambient temperature \([°C]\)
- \(T_{fm}\): Fluid mean temperature \([°C]\)
- \(T_{in,hot}\): Hot inlet temperature \([°C]\)
- \(T_{out,cold}\): Cold outlet temperature \([°C]\)
- \(T_a\): Help parameter [-]
- \(U_L\): Heat loss coefficient based on absorber area \([W/(m^2K)]\)
- \(V\): Collector volume flow rate \([m^3/s]\)
- \(x_1\): x coordinate for \(P_1\) \([m]\)
- \(x_n\): Help length \([m]\)
- \(x_r\): y coordinate for \(P_r\) \([m]\)
- \(y_1\): y coordinate for \(P_1\) \([m]\)
- \(y_r\): z coordinate for \(P_r\) \([m]\)
- \(z_1\): z coordinate for \(P_1\) \([m]\)
- \(z_n\): Help length \([m]\)
- \(\beta_s\): Collector panel tilt \([rad]\)
- \(\gamma_s\): Solar azimuth \([rad]\)
- \(\rho\): Collector fluid density \([kg/m^3]\)
- \(\theta\): Incident angle \([rad]\)
- \(\phi\): Solar zenith \([rad]\)
- \(\tau_{e}\): Effective transmittance-absorptance product [-]
- \(\xi\): Integration variable \([rad]\)
- \(\gamma_0\): Integration border \([rad]\)
- \(\gamma_1\): Integration border \([rad]\)
- \(\gamma_1\): Collector panel azimuth \([rad]\)
- \(\gamma_{actual}\): Actual absorber azimuth \([rad]\)

**GREEK SYMBOLS:**

- \(\beta_s\): Collector panel tilt \([rad]\)
- \(\gamma_s\): Solar azimuth \([rad]\)
- \(\rho\): Collector fluid density \([kg/m^3]\)
- \(\theta\): Incident angle \([rad]\)
- \(\phi\): Solar zenith \([rad]\)
- \(\tau_{e}\): Effective transmittance-absorptance product [-]
- \(\xi\): Integration variable \([rad]\)
- \(\gamma_0\): Integration border \([rad]\)
- \(\gamma_1\): Integration border \([rad]\)
- \(\gamma_1\): Collector panel azimuth \([rad]\)
- \(\gamma_{actual}\): Actual absorber azimuth \([rad]\)
Bilag 3: Artikel optaget i det videnskabelige tidsskrift APPLIED ENERGY.
Vertical evacuated tubular-collectors utilizing
solar radiation from all directions

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Abstract

A prototype collector with parallel-connected evacuated double glass tubes is investigated theoretically and experimentally. The collector has a tubular absorber and can utilize solar radiation coming from all directions. The collector performance is measured in an outdoor test facility. Further, a theoretical model for calculating the thermal performance is developed. In the model, flat-plate collector's performance equations are integrated over the whole absorber circumference and the model determines the shading on the tubes as a function of the solar azimuth. Results from calculations with the model are compared with measured results and there is a good degree of similarity between the measured and calculated results. The model is used for theoretical investigations on vertically-placed pipes at a location in Denmark (Copenhagen, lat. 56°N) and at a location in Greenland (Uummannaq, lat. 71°N). For both locations, the results show that to achieve the highest thermal performance, the tube centre distance must be about 0.2 m and the collector azimuth must be about 45–60° towards the west. Further, the thermal performance of the evacuated solar-collector is compared to the thermal performance of the Arcon HT flat-plate solar-collector with an optimum tilt and orientation. The Arcon collector is the best performing collector under Copenhagen conditions, whereas the performance of the evacuated tubular collector is highest under the Uummannaq conditions. The reason is that the tubular collector is not optimally tilted in Copenhagen but also that there is much more solar radiation “from all directions” in Uummannaq and this radiation can be utilized with the tubular collector. It is concluded that the collector design is very promising—especially for high latitudes.

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Keywords: Evacuated tubular solar collectors; Collector modelling; Solar heating
Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
<td>Incident angle modifier constant [ ]</td>
</tr>
<tr>
<td>A_1</td>
<td>Absorber perimeter length [m]</td>
</tr>
<tr>
<td>A_a</td>
<td>Absorber area [m²]</td>
</tr>
<tr>
<td>A_b</td>
<td>Absorber area exposed to beam radiation [m²]</td>
</tr>
<tr>
<td>A_cross,tube</td>
<td>Tube’s cross-section area [m²]</td>
</tr>
<tr>
<td>A_d</td>
<td>Absorber area exposed to diffuse radiation [m²]</td>
</tr>
<tr>
<td>A_g</td>
<td>Absorber area exposed to ground reflected radiation [m²]</td>
</tr>
<tr>
<td>C</td>
<td>Cylinder centre distance [m]</td>
</tr>
<tr>
<td>C_p</td>
<td>Collector-fluid’s heat-capacity [J/(kg K)]</td>
</tr>
<tr>
<td>C_p,vol</td>
<td>Collector-panel’s heat-capacity [J/K]</td>
</tr>
<tr>
<td>F_1-2</td>
<td>View factor from tube 1 to tube 2 [ ]</td>
</tr>
<tr>
<td>F_0-2</td>
<td>View factor from tube to ground [ ]</td>
</tr>
<tr>
<td>F_0-s</td>
<td>View factor from tube to sky [ ]</td>
</tr>
<tr>
<td>F_c</td>
<td>Collector’s efficiency-factor [ ]</td>
</tr>
<tr>
<td>F_c-2</td>
<td>View factor from tube to ground without adjacent tubes [ ]</td>
</tr>
<tr>
<td>F_s-2</td>
<td>View factor from tube to sky without adjacent tubes [ ]</td>
</tr>
<tr>
<td>G_b</td>
<td>Beam radiation on horizontal [W/m²]</td>
</tr>
<tr>
<td>G_d</td>
<td>Diffuse radiation on horizontal [W/m²]</td>
</tr>
<tr>
<td>G_global</td>
<td>Global radiation [W/m²]</td>
</tr>
<tr>
<td>G_gr</td>
<td>Ground reflected radiation on horizontal [W/m²]</td>
</tr>
<tr>
<td>G_total</td>
<td>Total radiation on collector plane [W/m²]</td>
</tr>
<tr>
<td>h_0</td>
<td>Solar altitude [rad]</td>
</tr>
<tr>
<td>k_0</td>
<td>Collector’s heat-loss coefficient [W/m²K]</td>
</tr>
<tr>
<td>K_a</td>
<td>Incident angle modifier for beam radiation [ ]</td>
</tr>
<tr>
<td>K_d</td>
<td>Incident angle modifier for diffuse radiation [ ]</td>
</tr>
<tr>
<td>K_gr</td>
<td>Incident angle modifier for ground reflected radiation [ ]</td>
</tr>
<tr>
<td>L</td>
<td>Pipe length [m]</td>
</tr>
<tr>
<td>N</td>
<td>Number of pipes [ ]</td>
</tr>
<tr>
<td>O_1</td>
<td>Help length [m]</td>
</tr>
<tr>
<td>O_2</td>
<td>Help length [m]</td>
</tr>
<tr>
<td>P_b</td>
<td>Power from beam radiation on collector/tube [W]</td>
</tr>
<tr>
<td>P_d</td>
<td>Power from diffuse radiation on collector/tube [W]</td>
</tr>
<tr>
<td>P_gr</td>
<td>Power from ground-reflected radiation on collector/tube [W]</td>
</tr>
<tr>
<td>P_loss</td>
<td>Heat loss from collector/tube [W]</td>
</tr>
<tr>
<td>P_u</td>
<td>Useful power from collector/tube [W]</td>
</tr>
<tr>
<td>q_γs</td>
<td>Angle determining the area exposed to beam radiation as a function of γ_s [rad]</td>
</tr>
<tr>
<td>R_b</td>
<td>Geometric factor; irradiance on a tilted surface divided by irradiance on a horizontal surface [ ]</td>
</tr>
<tr>
<td>r_e</td>
<td>Outer glass tube radius [m]</td>
</tr>
<tr>
<td>r_p</td>
<td>Absorber radius [m]</td>
</tr>
</tbody>
</table>
1. Introduction

A new collector design, based on evacuated tubular collectors, is investigated theoretically and experimentally. The collector is based on a number of parallel-connected double glass tubes, which are open at both ends. The tubes are annular with closed ends and the outside of the inner glass wall is treated with an absorbing selective coating. The collector fluid is floating from bottom to top of the inside of the inner tube where also another closed tube is inserted with the purpose of filling a part of the tube volume so that less collector fluid is needed. Further, it ensures a high heat-transfer coefficient from the inner glass tube to the collector fluid. Fig. 1 shows the design of the evacuated tubes and Fig. 2 shows the principle of the tube connection.

For the theoretical investigation of this collector principle, traditional collector theory cannot be applied directly, as the absorbers are tubular. Therefore, to
Theoretically determine the collector performance a number of conditions must be taken into account, including:

- That solar radiation from all directions can be utilized (also from the “back” of the collector).
- Shadow effects from adjacent tubes.
- Special incident-angle modifiers.

Thermal modelling of evacuated tubular collectors has previously been addressed. Barrett et al. [1] developed an evacuated tubular collector model that included two
incidence-angle modifiers for the longitudinal and the transverse directions. This model has widely been used, among others, by Qin and Furbo [2] who investigated the performance of differently designed evacuated solar-collectors. Based on the tube geometry and panel design, the model cannot be used to find the efficiency expression and the incidence angle modifiers since the model does not take into account directly shadow effects from adjacent tubes or solar radiation received from the back of the collector.

Perez et al. [3] developed a radiation model for evacuated tubular collectors with tubular absorbers. The model was based on the theory of a single-axis tracking solar collector and did take into account solar radiation received from the back of the collector. Further, the model gave an estimation of the size of the shadowed area of the total collector panel. However, the model did not determine the position of the shadows on each tube.

Lart [4] developed a geometrical method to determine the size and position of the shadowed area on each tube. This information was used to modify the transverse incidence-angle modifier so that the effective absorber area was taken into account.

In the present work, a collector theory for the collector's performance is developed. In the model, flat-plate collector performance equations are integrated over the absorber circumference and the model determines the shading on the tubes as a function of the solar azimuth. In this way, the determination of the transverse incident-angle modifier is not necessary.

Further, the collector is investigated in an outdoor test facility in order to determine the collector's performance experimentally. The theory is compared with the results from experiments. Based on the theory, the following points are investigated:

- Distance between tubes.
- Optimal collector-azimuth.
- Expected yearly thermal performances for different climates.

Finally, a comparison between the thermal performance of a flat-plate collector and of the investigated collector is made.

2. Collector design

The solar collector panel consists of 14 evacuated-tubes placed with a centre distance of 0.067 m. The tubes are connected to two manifold pipes, which are placed in an insulated box. The tubes are 1.6 m long, however, 2 x 0.065 m is placed inside the manifold boxes. Thus only 1.47 m is exposed to the Sun. The outer diameter of the outer tube is 0.047 m and the outer diameter of the inner tube is 0.037 m. The collector panel is vertically placed on the ground and faces south. The solar collector areas are described in Table 1 and Fig. 3 shows a photo of the collector.
3. Collector performance theory

Generally, for a solar collector without reflectors and without parts of the collector reflecting solar radiation to other parts of the collector, the performance equation can be written as:

\[ P_u = P_b + P_a + P_{gr} - P_{loss} \]  \hspace{1cm} (1)

or in more detail described by Shah et al. [5]:

---

Table 1
Solar collector panel's areas

<table>
<thead>
<tr>
<th>Gross area (m²)</th>
<th>Outer glass tube cross area (m²)</th>
<th>Absorber cross area (m²)</th>
<th>Total absorber area (m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.8</td>
<td>0.97</td>
<td>0.76</td>
<td>2.79</td>
</tr>
</tbody>
</table>
\[ P_e = A_b F(\tau \alpha) K_\theta R_b G_b + A_d F(\tau \alpha) K_{\theta, d} F_{\theta} G_d \]
\[ + A_t F(\tau \alpha) K_{\theta, t} F_{\theta} G_t - A_b U_1 (T_{in} - T_b) \]  

(2)

where \( K_\theta \) is the incident angle modifier defined as:

\[ K_\theta = 1 - \tan \left( \frac{\theta}{2} \right) \]  

(3)

The incident-angle modifiers for diffuse radiation, \( K_{\theta, d} \), and ground reflected radiation, \( K_{\theta, g} \), are evaluated by Eq. (3) using \( \theta = \pi/3 \).

For tubular collectors, there are several conditions, which make Eq. (2) more difficult to evaluate. Amongst others, the following can be mentioned:

- In flat-plate collector theory, the areas \( A_b \) and \( A_d \) are typically equal and close to the transparent area. For tubular collectors, however, this is not the case as, depending on the solar azimuth and altitude, only parts of the absorber area are exposed to the beam radiation.
- In flat-plate collector theory the incident angle modifier, \( K_\theta \), is independent of the longitudinal and transverse component of the incident angle. The cylindrical geometry in tubular collectors makes it necessary to consider both components.
- In the investigated tubular collector, where the absorber covers the whole inner-tube circumference, also radiation coming from the “back” of the collector must be evaluated.

To calculate the thermal performance of the evacuated tubes, the general performance Eqs. (1) and (2) have been integrated over the whole absorber circumference. This means that the tube is divided into small “slices”, and each slice is treated as if it was a flat-plate collector. In this way, the transverse incident-angle modifier is eliminated. For describing the solar radiation on a tubular geometry, this method has previously been used by Pyrko [6].

Integrating over the absorber area, the performance equation can be described as:

\[ P_e = \int_{-\pi/2}^{\pi/2} (P_b + P_d + P_g - P_{lax}) d\xi \]  

(4)

In the following, each part of Eq. (4) will be investigated. The investigation is based on a theoretical analysis of a single tube.

3.1. Heat loss, \( P_{loss} \)

The heat loss can be described as:
\[ P_{\text{loss}} = \int_{-\pi}^{\pi} A_p U_L (T_{\text{em}} - T_\alpha) d\xi \]
\[ = \int_{-\pi}^{\pi} L_{\rho p} U_L (T_{\text{em}} - T_\alpha) d\xi \]
\[ = 2\pi L_{\rho p} U_L (T_{\text{em}} - T_\alpha) \quad (5) \]

### 3.2. Power from diffuse radiation on collector\(\text{tube}, P_o\)

The evaluation of the power contribution from the diffuse radiation is based on an isotropic diffuse model. Thus, the circumsolar diffuse and horizontal brightening contributions are not taken into consideration in this model.

The power contribution from the diffuse radiation can be written as (Shah et al. [5]):

\[ P_o = \int_{-\pi}^{\pi} A_p F(\alpha) K_{0,\alpha} F_{\epsilon-\phi} G_\theta d\xi \]
\[ = 2\pi L_{\rho p} LF(\tau\alpha) K_{0,\alpha} G_\theta \int_{-\pi}^{\pi} F_{\epsilon-\phi} d\xi \quad (6) \]

Assuming that there are no adjacent tubes, the view factor from the tube to the sky can be described as:

\[ F_{o-t} = \int_{-\pi}^{\pi} \frac{1 + \cos(\beta)}{2} = 0.5 \quad (7) \]

In reality, there will be adjacent tubes, which will reduce the view factors to the ground and the sky respectively. This reduction must be taken into consideration.

Fig. 4 shows two adjacent tubes. The view factor \(F_{1-2}\) between the absorber of tube 1 and tube 2 can be described as:

\[ A_1 \cdot F_{1-2} = \frac{1}{2} \sum \text{(length of the crossing curves)} \]
\[ - \frac{1}{2} \sum \text{length of the non crossing curves} \]
\[ = (O_1 + z + O_2) - C\sin(\chi_\lambda) \quad (8) \]

Here \(A_1\) is the absorber perimeter of tube 1. The curves \(O_1\) and \(O_2\) between the points \(P_1, P_2\) and \(P_3, P_4\) respectively can be described by:
Fig. 4. Determination of view factors between the two tubes.

\[
O_1 = \frac{(\pi/2 - x_4) + (\pi/2 - x_3)}{2\pi} \cdot 2\pi r_p \tag{9}
\]

and

\[
O_2 = \frac{(\pi/2 - x_4) - (\pi/2 - x_3)}{2\pi} \cdot 2\pi r_c \tag{10}
\]

Here the angles \( x_1 \) and \( x_3 \) are defined by:

\[
x_1 = \cos \left( \frac{r_p + r_p}{C} \right) \tag{11}
\]

\[
x_3 = \cos \left( \frac{r_c - r_p}{C} \right) \tag{12}
\]

If the centre of tube 1 has the coordinates \((0,0)\), the coordinates of the points \(P_2\) and \(P_3\) and thus the distance, \(z\), between the two points can be found as:

\[
P_2 = [r_p \cos(x_1), r_p \sin(x_1)]
\]

\[
P_3 = [C + r_c \cos(\pi + x_1), r_c \sin(\pi + x_1)]
\]

\[
z = \sqrt{(C + r_c \cos(\pi + x_1) - r_p \cos(x_1))^2 + (r_c \sin(\pi + x_1) - r_p \sin(x_1))^2} \tag{14}
\]
By inserting Eqs. (9), (10) and (14) into Eq. (8), the view factor from tube 1 to tube 2 can be written as:

\[
F_{1-2} = \frac{1}{2\pi r_p} \left[ (x - x_1 - x_2) r_p 
+ \sqrt{(C + r_c \cos(x + x_1) - r_p \cos(x_1))^2 + (r_c \sin(x + x_1) - r_p \sin(x_1))^2}
+ (x_3 - x_1) r_c - C \sin(x_3) \right] \tag{15}
\]

The final view factor from tube to sky, including shading adjacent tubes can thus be described as:

\[
F_{v-s} = F_{v-s}^* - F_{v-2} \tag{16}
\]

3.3. Power from ground-reflected radiation on collector tube, \( P_{gr} \):

The power contribution from the ground reflected radiation is described by (Shah et al. [5]):

\[
P_{gr} = \int_{-\pi}^{\pi} A_F(x_0, x) K_e \varphi F_{e-g} G_{gr} d\xi \tag{17}
\]

\[
= 2 \pi r_p L F(\alpha_e) K_e \varphi \int_{-\pi}^{\pi} F_{e-g} d\xi
\]

with

\[
G_{gr} = \rho_{gr} (G_b + G_d) \tag{18}
\]

In a similar way as for the diffuse radiation, the view factor from the tube to the ground, assuming that there are no neighbouring tubes, can be described as:

\[
F_{v-g}^* = \int_{-\pi}^{\pi} \frac{1 - \cos(\beta)}{2} = 0.5 \tag{19}
\]

Including the adjacent shading tubes, the view factor from tube to ground becomes:

\[
F_{v-g} = F_{v-g}^* - F_{v-2} \tag{20}
\]
3.4. Power from beam radiation on the collector tube, $P_b$:

The power contribution from the beam radiation can be written as (Shah et al. [5]):

$$
P_b = \int_{\theta_{\text{beam}}}^{\theta_{\text{max}}} F(\tau x)_b G_b A_b R_b d\xi
= F(\tau x)_b G_b L_\gamma \int_{\theta_{\text{beam}}}^{\theta_{\text{max}}} K_b R_b d\xi
$$

(21)

Notice that there is now integration over only a part of the circumference. This is because only part of the absorber surface is exposed to the beam radiation.

Assuming that the tubes are placed vertically and that the collector panel azimuth is $0^\circ$, Fig. 5 shows the three critical angles, when the solar azimuth, $\gamma_s$, is between $0$ and $\pi/2$ (Lart [4]). When the solar azimuth is smaller than the angle $x_1$, there is no shading of the tubes. If the solar azimuth is larger than $x_3$, the tubes are fully shaded and if the solar azimuth is between $x_1$ and $x_3$ the tubes are partly shaded. If the solar azimuth is equal $x_2$, the tubes are half shaded.

For the analysis of the critical angles as well as the area that is exposed to beam radiation when the collector azimuth is not $0^\circ$, the tubes are divided into six parts. These parts are illustrated in Fig. 6 for a collector-panel orientation towards the east ($\gamma_c \leq 0^\circ$) and towards the west ($\gamma_c \geq 0^\circ$) and in Table 2 the critical angles are defined for all six parts of the circle.

![Figure 5](image)

Fig. 5. An illustration of the critical angles determining the exposed area, when the pipes are placed vertically and when the collector panel azimuth is $0^\circ$. 
Table 2
Critical angles determining the exposed area, when the pipes are placed vertically

<table>
<thead>
<tr>
<th>Part of the tube</th>
<th>$\gamma_c &lt; 0^\circ$</th>
<th>$\gamma_c &gt; 0^\circ$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$x_1$</td>
<td>$x_2$</td>
</tr>
<tr>
<td>1</td>
<td>$\pm \arccos \left( \frac{x_3 + x_4}{l_5} \right)$</td>
<td>$-\pi$</td>
</tr>
<tr>
<td>2</td>
<td>$\pm \arccos \left( \frac{x_3 - x_4}{l_5} \right)$</td>
<td>$-\pi$</td>
</tr>
<tr>
<td>3</td>
<td>$\arccos \left( \frac{x_3 + x_4}{l_5} \right)$</td>
<td>$-\arccos \left( \frac{x_3 - x_4}{l_5} \right)$</td>
</tr>
<tr>
<td>4</td>
<td>$\arccos \left( \frac{x_3 + x_4}{l_5} \right)$</td>
<td>$\arccos \left( \frac{x_3 - x_4}{l_5} \right)$</td>
</tr>
<tr>
<td>5</td>
<td>$\arccos \left( \frac{x_3 + x_4}{l_5} \right)$</td>
<td>$\arccos \left( \frac{x_3 - x_4}{l_5} \right)$</td>
</tr>
<tr>
<td>6</td>
<td>$-\arccos \left( \frac{x_3 + x_4}{l_5} \right)$</td>
<td>$-\arccos \left( \frac{x_3 - x_4}{l_5} \right) + \pi$</td>
</tr>
</tbody>
</table>

Fig. 6. For analysis purposes, the tubes are divided into six parts.
For a collector panel orientation towards the east, Figs. 7 12 show the angle 'qH', which represents the tube area exposed to beam radiation, for different solar azimuths. Further, the angles qH start and qH stop used in Eq. (21) are shown.

As a function of the solar azimuth, the angle, qH, and the integration borders, qH start and qH stop, are described in Table 3 for a collector-panel orientation towards the east (γH ≤ 0°) and towards the west (γH ≥ 0°).

3.5. The incident angle, θ, and the geometric factor, Rb:

In Eq. (21), the incident angle, θ, and the geometric factor, Rb, still need to be addressed. As mentioned earlier, when integrating over the absorber area, both the surface tilt and the surface azimuth change will have an impact on both Rb and θ.

The incident angle, θ, can be described as (Duffie and Beckman [7]):

$$\cos(\theta) = \sin(\delta)\sin(\phi)\cos(\beta)$$
$$- \sin(\delta)\cos(\phi)\sin(\beta)\cos(\xi)$$
$$+ \cos(\delta)\cos(\phi)\cos(\alpha)\cos(\beta)$$
$$+ \cos(\delta)\sin(\phi)\cos(\alpha)\cos(\xi)\sin(\beta)$$
$$+ \cos(\delta)\sin(\alpha)\sin(\xi)\sin(\beta)$$

The geometric factor, Rb, can be described as (Duffie and Beckman [7]):

$$R_b = \frac{\cos(\theta)}{\cos(\theta_{horizontal})}$$
$$= \frac{\cos(\theta)}{\cos(\delta)\cos(\phi)\cos(\alpha) + \sin(\delta)\sin(\phi)}$$

All information needed to calculate the thermal performance is given in the above equations and the collector's outlet temperature T_out,hot can finally be calculated from:

$$T_{out, hot} = \frac{P_u \cdot C_D \cdot (T_{in} - T_{in, hot})}{V \cdot \Delta t} + T_{in, cold}$$

3.6. Solving the performance equation:

All the parameters involved in the performance Eqs. (1) and (2) have been described in Eqs. (3)-(23). In order to evaluate the performance of the tubular collector on a yearly basis, the above theory was implemented into a numerical program. All the
Table 3
Angles determining the exposed area, when the pipes are placed vertically

<table>
<thead>
<tr>
<th>γ₀ &lt; 9°</th>
<th>( y₀ \leq y₁ )</th>
<th>( y₀ \geq y₁ )</th>
<th>γ₀ &gt; 9°</th>
<th>( y₀ \leq y₁ )</th>
<th>( y₀ \geq y₁ )</th>
</tr>
</thead>
<tbody>
<tr>
<td>γ₀ &lt; 9°</td>
<td>( \text{ac}(\text{Con}(r₀, y₀, x₀, x₀, x₀)) )</td>
<td>( \text{ac}(\text{Con}(r₀, y₀, x₀, x₀, x₀)) )</td>
<td>( \text{ac}(\text{Con}(r₀, y₀, x₀, x₀, x₀)) )</td>
<td>( \text{ac}(\text{Con}(r₀, y₀, x₀, x₀, x₀)) )</td>
<td>( \text{ac}(\text{Con}(r₀, y₀, x₀, x₀, x₀)) )</td>
</tr>
<tr>
<td>( y₀ &lt; y₁ ), ( y₀ &lt; x₀ )</td>
<td>( y₀ &lt; y₁ ), ( y₀ &lt; x₀ )</td>
<td>( y₀ &lt; y₁ ), ( y₀ &lt; x₀ )</td>
<td>( y₀ &lt; y₁ ), ( y₀ &lt; x₀ )</td>
<td>( y₀ &lt; y₁ ), ( y₀ &lt; x₀ )</td>
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<td>( y₀ &lt; x₀ ), ( y₀ &lt; x₀ )</td>
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<td>( y₀ &lt; x₀ ), ( y₀ &lt; x₀ )</td>
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<td>( y₀ &gt; x₀ ), ( y₀ &gt; x₀ )</td>
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<td>( y₀ &gt; x₀ ), ( y₀ &gt; x₀ )</td>
<td>( y₀ &gt; x₀ ), ( y₀ &gt; x₀ )</td>
</tr>
</tbody>
</table>

\( y₀ \) is in Part 1
\( y₀ \) is in Part 2
\( y₀ \) is in Part 3
\( y₀ \) is in Part 4
\( y₀ \) is in Part 5
\( y₀ \) is in Part 6
Fig. 7. Relationship between the solar azimuth, $\gamma_s$, the beam exposure angle, $\theta_{ex}$, and the integration borders $\xi_{\text{start}}$ and $\xi_{\text{stop}}$ when $\gamma_s$ is in part 1 and when the pipes are placed vertically.

Fig. 8. Relationship between the solar azimuth, $\gamma_s$, the beam exposure angle, $\theta_{ex}$, and the integration borders $\xi_{\text{start}}$ and $\xi_{\text{stop}}$ when $\gamma_s$ is in part 2 and when the pipes are placed vertically.
Fig. 9. Relationship between the solar azimuth, $\gamma_s$, the beam exposure angle, $q_{\text{ex}}$, and the integration borders $\xi_{\text{start}}$ and $\xi_{\text{stop}}$, when $\gamma_s$ is in part 3 and when the pipes are placed vertically.

Fig. 10. Relationship between the solar azimuth, $\gamma_s$, the beam exposure angle, $q_{\text{ex}}$, and the integration borders $\xi_{\text{start}}$ and $\xi_{\text{stop}}$, when $\gamma_s$ is in part 4 and when the pipes are placed vertically.
Fig. 11. Relationship between the solar azimuth, $\gamma_s$, the beam exposure angle, $q_{pe}$, and the integration borders $\xi_{start}$ and $\xi_{stop}$, when $\gamma_s$ is in part 5 and when the pipes are placed vertically.

Fig. 12. Relationship between the solar azimuth, $\gamma_s$, the beam exposure angle, $q_{pe}$, and the integration borders $\xi_{start}$ and $\xi_{stop}$, when $\gamma_s$ is in part 6 and when the pipes are placed vertically.
integrals could be solved analytically, except the integral in Eq. (21), which was solved by using the trapezoidal formula for solving integrals numerically. Three hundred and sixty integration steps were used in the numerical integration.

The program is based on weather data with hourly data for global radiation, diffuse radiation on horizontal and outdoor temperature. However, the incident angle and thus the collector performance were calculated every half hour.

4. Measurements

The performance of the collector was measured in an outdoor test facility where the inlet temperature, the outlet temperature and the volume flow-rate were measured. The temperatures were measured with copper-constantan thermocouples (Type TT) and the volume flow rate was measured with a HGQ1 flow meter. A 31% glycol/water mixture was used in the solar-collector loop. Further, the global radiation and the diffuse radiation on the horizontal were measured with two Kipp & Zonen CM11 pyranometers.

The power from the solar collector was determined from the measurements by:

$$P_v = VpC_p(T_{out,hot} - T_{in,cold}) + C_{p,coil}(T_{in} - T_{out})$$

(25)

5. Comparing the model predictions with measurements

Two periods of 6 days (25/7-30/7 2003 and 12/8-7/8 2003) have been selected for validating the computer model of the collector. In Fig. 13, the global irradiance, the ambient temperature, the inlet temperature to the collector and the volume flow rate in the collector are shown. These values are used as input data to the model.

The necessary data for describing the collector are shown in Table 4. The heat loss coefficient, $k_0$, was determined from efficiency measurements and split into two parts for the evacuated tubes and the manifold pipes respectively (Shah et al. [5]). $F$ was calculated from theory (Duffie and Beckman [7], Incropera and de Witt [8]) and $\omega_k$ and $\omega_r$ were calculated with a simulation program for determining optical properties (Svendsen and Jensen [9]). The collector's heat-capacity was calculated from the geometrical information.

<table>
<thead>
<tr>
<th>No. of pipes</th>
<th>L (m)</th>
<th>$t_e$ (m)</th>
<th>$t_p$ (m)</th>
<th>C (m)</th>
<th>$k_0$ (W/m²°C)</th>
<th>F (%)</th>
<th>$\omega_k$ (%)</th>
<th>$C_{p,coil}$ (J/K)</th>
<th>a (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>14</td>
<td>1.47</td>
<td>0.0235</td>
<td>0.0185</td>
<td>0.067</td>
<td>2.09</td>
<td>0.98</td>
<td>0.856</td>
<td>27614</td>
<td>3.8</td>
</tr>
</tbody>
</table>
In Fig. 14 the measured and calculated collector outlet temperatures are compared. It can be seen that there is a good degree of similarity between the measured and calculated temperatures.

6. Theoretical investigations

In this section, the model will be used for theoretical investigations. Only vertically-placed pipes will be analysed, as the model is yet unfit to calculate for tilted pipes. The collector’s performance is investigated for two locations:

- Copenhagen, Denmark, lat. 56°N, yearly average ambient-temperature: 7.8 °C. Weather data: DRY (Lund [10]).
- Uummannaq, Greenland, lat. 71°N, yearly average ambient-temperature: −4.2 °C. Weather data: TRY (Kragh [11]).

Fig. 15 shows the sum of the solar radiation on the front and the back of vertical planes with different orientations. For Copenhagen, there is symmetry around 0° (south) whereas for Uummannaq the minimum solar radiation on the plane is around −30° (towards the east). The reason for this asymmetrical behaviour is that there is a mountain east of Uummannaq, which reduces the radiation coming from the east.

As a function of the collector’s azimuth, Fig. 16 shows the utilized solar energy per tube in a panel assuming a tube centre distance of 0.67 m and a constant solar-collector’s fluid temperature throughout the year of 50 °C. For both Uummannaq and Copenhagen, there is an optimum at a collector azimuth of about 45°–60°. The
results are caused both by the distribution of solar radiation and by the higher afternoon temperatures.

Fig. 17 shows the utilized solar-energy per tube as a function of the tube centre distance for a collector fluid temperature of 50 °C. For both locations, the utilized energy increases for tube centre distances up to 0.2 m, which is due to the reduced shaded areas. For larger distances, the utilized energy decreases again, due to the increasing heat loss from the manifold pipes.

For the two locations, Figs. 18 and 19 show the utilized solar-energy per tube as a function of the collector azimuth and the tube centre distance for a collector fluid
mean temperature of 50 °C. Here it can be seen that the tendencies in Fig. 16 are true for tube distances in the range of 0.047–0.3 m.

Finally Figs. 20 and 21 show the utilized solar-energy per m² as a function of the collector-fluid’s mean temperature assuming a tube centre distance of 0.047 m and a collector azimuth of 50°. Further, the thermal performance of the newest (Vejen et al. [12]) Arcon HT flat-plate collector is shown. This collector represents the state of the art of collectors for solar-heating plants. The Arcon HT flat-plate collector is
Fig. 18. Uummannaq: Utilized solar-energy per tube as a function of the collector’s azimuth and the tube centre distance. Collector’s fluid mean-temperature: 50 °C.

Fig. 19. Copenhagen: Utilized solar-energy per tube as a function of the collector’s azimuth and the tube centre distance. Collector’s fluid mean-temperature: 50 °C.
facing south and tilted at 45° in Copenhagen. In Uummannaq the Arcon HT flat-plate collector is facing south and tilted at 60°.

It can be difficult to compare the thermal performances of flat-plate collectors and tubular collectors as the effective area of a flat plate collector typically is defined as the transparent area of the glass cover and the effective area of a tubular collector can be defined in many ways. In the present comparison, having a tube centre distance of 0.047 m eliminates this problem. This means that there is no air-gap between the tubes and the outer tube cross-section area (≈L-2·t·N) directly corresponds to the transparent area of a flat-plate collector.

In Figs. 20 and 21, it is interesting that the Arcon collector is the best performing collector under Copenhagen conditions, whereas the thermal performance of the evacuated tubular collector based on the outer tube cross-section area, is the highest under the Uummannaq conditions. The reason for the change in the ranking of the

---

Fig. 20. Copenhagen: Utilized solar-energy per m² as a function of the collector’s fluid mean-temperature. For the ARCON HT flat-plate collector, the area in consideration is defined as the transparent area of the cover plate and, for the tubular collector, the considered area is the outer-tube’s cross section area.

Fig. 21. Uummannaq: Utilized solar-energy per m² as a function of the collector’s fluid mean-temperature. For the ARCON HT flat-plate collector, the area in consideration is defined as the transparent area of the cover plate and, for the tubular collector, the considered area is the outer-tube’s cross section area.
collectors is that the tubular collector is not optimally tilted in Copenhagen but also that there is much more solar radiation “from all directions” in Uummannaq and this radiation can be utilized with the tubular collector.

7. Discussion and conclusion

A prototype collector with parallel-connected evacuated double glass tubes is investigated theoretically and experimentally. The collector has a tubular absorber and can thus utilize solar radiation coming from all directions. The collector performance is measured in an outdoor test facility.

Further, a theoretical model for calculating the thermal performance is developed. In the model, the flat-plate collector performance equations have been integrated over the whole absorber circumference. In this way, the transverse incident-angle modifier is eliminated. Also, the model determines the shading of the tubes as a function of the solar azimuth in order to calculate the energy from the beam radiation correctly. The calculation of the energy from the diffuse and ground-reflected radiation is based on an isotropic diffuse sky model.

The calculations with the model are compared with measured results and there is a good degree of similarity between the measured and calculated results.

The model is used for theoretical investigations on vertically-placed pipes placed in Copenhagen, Denmark and in Uummannaq, Greenland. For both locations, the results show that to achieve the highest thermal performance the tube distance should be about 0.2 m and the collector azimuth should be about 45° towards the west. The thermal performance of the evacuated solar collector is also compared to the thermal performance of the Arcon HT flat-plate solar collector. These results show that the Arcon collector is the best performing collector under Copenhagen conditions, whereas the performance of the evacuated tubular collector is the highest under the Uummannaq conditions. The reason is that the tubular collector is not optimally tilted in Copenhagen but also that there is much more solar radiation “from all directions” in Uummannaq and this radiation can be utilized with the tubular collector. It is therefore concluded that the collector design is very promising—especially for high latitudes.

The theoretical results presented in this paper are based on a collector model, which needs to be further developed. First of all, the model must be able to calculate for tilted pipes. The reflections between the pipes must be included in the model and also an anisotropic diffuse-sky model should be included. This extended model must of course be thoroughly validated with measurements.

Finally, though the collector design seems very promising, the collector reliability and durability must be examined before any final conclusions can be drawn.

Acknowledgements

The study is financed by the VILLUM KANN RASMUSSEN FOUNDATION.
References

Solvarme i Grønland

Stort potentiale for udnyttelse af solvarme i Grønland

Solenergi er den reneste og naturligste energiform, vi overhovedet har. Solindfaldet er så stort på kloden – og i Grønland – at der er mulighed for at udfytte solenergi i stort omfang.

Solenergi kan udnyttedes til at reducere brugen af fossile brændstoffer, f.eks. ved at anvende solvarmeanlæg til boliger. Solvarmeanlæg kan for eksempel benyttes til brugsvandsovarmning eller til kombineret rumoalvarmning og brugs- vandsovarmning.

Det årlige antal timer med mulighed for solskin er stort set det samme, uanset hvor på kloden vi befinder os.


Solindfaldet på en flade afhænger stærkt af fladens lokalitet, orientering og hældning. I København (breddegrad 56°) er solindfaldet stort på en sydvestlig hældende flade, mens solindfaldet i Sisimiit i Grønland (breddegrad 67°) er stort på en sydvestlig hældende flade. Solindfaldet i København og i Sisimiit er stort set ens på de optimale hældende flader, ca. 1160 kWh/m²/år.

Solvarmeanlæg

Anvendelsen af solvarmeanlæg varierer stærkt fra land til land. I Europa er Østrig og Grønland, efterfulgt af Danmark, Tyskland og Schweiz, de lande hvor der er installeret flest kvadratmeter solfanger pr. indbygger. Der er ingen entydig sammenhæng mellem disse landes (relative) succes inden for solvarmeområdet, solindfald og energiprisniveau. Der er eller har været en aktiv solvarmeindustri og politisk opbakning til solvarmeanlæg i form af støtte til forskning, udvikling og demonstrationsprojekter i alle de navnede lande.

Derudover er der, eller har der været, økonomiske støtte til opførelse af solvarmeanlæg.

Solvarmeanlægs rentabilitet afhænger stærkt af energiprisniveauet og energiprisudviklingen. I Danmark har typiske solvarmeanlæg økonomiske tilbagebeløbningstider på ca. 10 år og energimæssige tilbagebeløbningstider på ca. et år, og der er inden for en forholdvis kort tidstabels mulighed for teknologisk udvikling så den økonomiske tilbagebeløbningstid når ned på ca. fem år.

Solvarme i Grønland

Der er et alternativt forbrug af solvarme i Grønland. Blandt andet kan det nævnes at:

- energipriserne for fossilt brændsel er lavere i Grønland end i Danmark.
- der ikke er en solvarmeindustri i Grønland.
- der ikke er solvarmeanlæg i Grønland (eller kun få).
- der ikke er udviklet solvarmeanlæg, som er specielt velegnede til Grønland.

Der er dog også en række forhold, som gør, at solvarmeanlæg er mere velegnede i Grønland end i Danmark. Blandt andet kan nævnes at:

- solen reflekterer en meget stor del af solstrålingen. Denfor er solindfaldet på tagflader i perioder med sne på jorden meget stort i Grønland.
- der er rumoalvarningsbehov i sommerperioden med meget sol i Grønland.
- temperaturen af det kolde brugvand, der tilsides holder, er lavere i Grønland end i Danmark.
- den optimalesolfangerhældning fra vandet er større i Grønland end i Danmark. Det betyder, at solfangerens effektiviteten for den samme solfanger er højere i Grønland end i Danmark.
- der er mere solindfald fra "alle retninger" i Grønland end i Danmark. I denne forbindelse kan det nævnes, at de forholdvis lille kinesiske massproducenter vakuumpanser sandsynligvis er velegnede til grønlandske forhold, da de kan udnytte solstråling fra alle retninger, dvs. de kan udnytte solstrålingen i alle dagslysstimer, hvis blot rorene placeres loddret med frit udsyn til alle sider.

Vakuumpanserfanger

Vakuumpanserfanger har i mange år været markedsført i Europa og i USA. Da der er vakuumpanser, er varmety-
DTU udstiller i lufthavnen

Fra 1. april - 5. maj detager DTU i udstillingen Future & Technology i Københavns Lufthavn. Fra DTU vil man bl.a. kunne blive klogere på den seneste forskning inden for laser- og mikromodulation på IPL, tele- kommunikation på COM, ansigtsmodellering og ansigtsgenkendelse på IMM og satellitnavigation, solfanger m.v. på Center for Arktisk Teknologi/BYG/Oersted.

UD over DTU delager bl.a. Intel, Center for Advanced Technology (CAT), Symbion, Alexandra Institutet og Institut for Fremtidsoptik.

Udstillingen er delt op i tre zoner, og DTU kan opleves i Zone 1 over for den store Tax Free butik mellem Finger B og C - også kaldet Nytorv. Så for at se udstillingen kræver det altid, at man skal ud at flyve i løbet af den næste måned. Ca. 40.000 passagerer rejser hver dag igennem lufthavnen, så der er tale om en ganske pæn eksponering af nogle af DTUs forskningsresultater.

Studerende fra DTU benævner udstillingen, som er åben hver dag fra kl. 09.30 - 19.00. Læn til de studerende samt øvrige udgifter til planche og betale af lufthavnen og Intel, som sponsorerer udstillingen.

...bet fra adsorberne på grund af konvektion og varmeledning meget lille. Varmetabskefficeren for voksnosolfangere er derfor meget mindre end varmetabskefficien
ten for almindelige plane solfanger. I modtagelse til almindelige plane solfanger kan voksnosolfangere udnytte solstrålen spredt galt ud, når indfaldsvinken er stor. Ansagen til dette forhold er dels refleksionsforholdene mellem glasrenne, dels glassrenenes cylinderformede overlade, som tillader, at solstråler transmitsere gennem glasset selv ved store indfaldsvinkler på tværs af glassrenene. Voksnosolfanger udnytter solvens stråler specielt godt ved høje solfangleverter, ved lave udluftetemperaturer, ved små bestmætningstrykker og ved store indfaldsvinkler.

Billige solfanger fra Østen

For nylig har en række kinesiske firmaer startede masseproduktion af forholdsvis billige voksnosolfangere. I asiatisk har voksnosolfanger derfor nået så lavt et prisniveau og så høj en effektivitet, at det er blevet attraktivt at benytte disse højeflektive solfanger i stedet for almindelige plane solfanger.

De mest anvendte kinesiske solfanger er baseret på dobbelglasrevier med voksn i mellemrummet mellem glassrene. De udvendige overlader af de inderste glassrene har en høj absorptans og en lav emitt-

Studenteprojekter


For nylig er et eksamensprojekt ved BYG-DTU med Titel Design og Analyse af an Evacuated Tubular Collector afsat- tet, hvor en første forsvagsglassfanger base-
ret på de kinesiske solfanger blev opbygget (se foto), og i et igangværende eksamensprojekt, ”Vakuumrørslangere til Grønland”, undersøges grønlandske solfangersforhold samt forventede ydelser for voksnosolfangere under grønlandske forhold.

Yderligere er der studerende i gang med et midtvejsprojekt, ”Vakuumrørslangere og Solvarmeudlægning under Arktiske Forhold”, som de har kombineret med kurset Arktisk Teknologi. Under deres feltstudier i Grønland til sommer skal de der midtvæsstudere bl.a. undersøge installationsforhold med fokus på optimal hældning og orientering af voksnosolfangerne.

Voksnosolfangere er ikke kun interessant for arktiske forhold. Det er interessant for alle klimaforhold og for de fleste typer af solvarmeanlæg. Det er bl.a. fordi det med optimalt designede voksnosolfanger er mulighed for forbedrede solvarmeanläggs rentabilitet mærkbart. Det kræver grundig forskning, før det er muligt at udforme solfangerne på denne måde.

Der er mange andre muligheder for teknologiske forbedringer af solvarmeanlæg, og vi håber derfor på at kunne fortælle om de mange studenteprojekter og videre forskning udviklede voksnosolfangere af solvarmeanlæg.

Louise Iban Starck, forskningsudvalg
Simon Pamb, ekst. BYG-DTU
Bilag 5: Overheads til foredraget “Thermal Performance of Evacuated Tubular Collectors utilizing Solar Radiation from all Directions”.
Thermal Performance of Evacuated Tubular Collectors utilizing Solar Radiation from all directions

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The Evacuated Tubular Collector
**Investigations**

- **Theoretical:**
  - Developed collector performance theory that include:
    - Treatment of solar radiation from all directions (also from the "back" of the collector)
    - Shadow effects from neighbouring tubes
    - Treatment of tubular geometry and incident angle modifiers

- **Experimental:**
  - Thermal performance
  - Efficiency expression

---

**Theory**

The collector performance equation is integrated over the pipe circumference:

\[ P_v = \int_{-\pi}^{\pi} \left( P_b + P_d + P_{gr} - P_{loss} \right) \, \mathrm{d}\xi \]

\[ P_b = A_b \cdot F_b(\tau \alpha) \cdot K_b \cdot R_b \cdot G_b \]
\[ P_d = A_d \cdot F_d(\tau \alpha) \cdot K_d \cdot R_d \cdot F_{c+d} \cdot G_d \]
\[ P_{gr} = A_a \cdot F_{gr}(\tau \alpha) \cdot K_{gr} \cdot R_{gr} \cdot F_{c-gr} \cdot G_{gr} \]
\[ P_{loss} = A_a \cdot U_L \cdot (T_{fin} - T_a) \]
The Model

- View factor between tubes
  - depending on tube geometry and distance
- View factors to sky and ground
  - integrating over the tube circumference
  - minus view factor to adjacent tubes
- The energy from beam radiation
  - size and position of the non-shaded area as a function of solar azimuth
  - integrating over this area
- The energy from diffuse radiation
  - isotropic model
- The energy from ground reflected radiation
  - albedo
- Heat loss
  - based on a heat loss coefficient for the total absorber area

All included in a simulation program that can calculate the thermal performance of the tubular collector on a yearly basis.

Measurements

Efficiency expression:
- \( G_{int} > 800 \text{ W/m}^2 \)
- Incidence angle on collector aperture < 20°
- \( G_{sh} < 0.22 \times G_{int} \)
- Stationary conditions during at least 15 min.
- Based on outer tube cross area (0.97 m²)

\[
\eta = 0.99 - 6.17 \frac{dT}{G_{int}}
\]
\[R^2 = 0.93\]
Comparison of measured and calculated results

- **Model input:**
  - Global irradiance, diffuse irradiance and ambient temperature
  - Inlet temperature and volume flow rate
  - Geometry, $F$ and $w_d$

- **Model output:**
  - Outlet temperature

![Graph showing temperature variations over time](image)

**Comparison of measured and calculated results**

- **Problem:**
  - Systematic error in the mornings/evenings

- **Reason:**
  - Shadow model not developed for tilted pipes

![Graph showing temperature variations over time](image)
Investigations with the model

- **Vertical pipes**
- **Two locations:**
  - Ummannaq (GL, 71°N)
  - Copenhagen (DK, 56°N)
- **Expected yearly thermal performance:**
  - Optimal pipe distance
  - Optimal collector azimuth
  - Yearly thermal performance
  - Comparison with Arcon HT

Pipe distance/Collector azimuth

**Copenhagen:**

**Ummannaq:**

(Outer tube diameter=0.047 m)
Yearly thermal performance
based on the transparent area

Tubular collector:
Orientation: 50° west
Tilt: 0°

Arcon HT:
Orientation: 0° south
Tilt: 45°

Tubular collector:
Orientation: 50° west
Tilt: 0°

Arcon HT:
Orientation: 0° south
Tilt: 60°

Conclusion

- A prototype collector utilizing solar radiation from all directions has been investigated
- Vertical placed tubes in Copenhagen and Ummannaq have the highest thermal performance with:
  - a tube distance of 0.2 m.
  - a collector azimuth of 45°-50° towards west.
- Calculated yearly thermal performance:
  - Copenhagen: Arcon HT best performing.
  - Ummannaq: Tubular collector best performing.
  - Reason: The tubular collector is not optimally tilted in Copenhagen and there is much more solar radiation "from all directions" in Ummannaq.
- The collector design is very promising – especially for high latitudes.
- Further work:
  - the model must be able to calculate on tilted pipes
  - the reflections between the pipes must be included in the model
  - an anisotropic diffuse sky model should be included
  - the model must be thoroughly validated with measurements.
  - more measurements
  - durability and reliability tests
Bilag 6: Overheads til foredraget ”Vakuumrørsolfangere”.

DANVAK møde: ”Solvarmeforskning på DTU”, 18/9 2003.
Vakuumrørsolfanger som kan udnytte solstrålingen fra alle retninger

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Renafstand: 6,7 cm
Prototype solfanger
Rørets opbygning

Undersøgelser

- **Teoretisk:**
  - Udvikling af en solfanger teori som kan behandle:
    - Solstråling fra alle retninger (og så bag fra solfangeren)
    - Skyggeeffekter fra naboskere
    - Cylindrisk geometri i forbindelse med indfaldsvinkelkorrektioner

- **Eksperimentelt:**
  - Solfangerydelse
Teorien inkluderer:

- Vinkelforhold mellem nære
  - som afhænger af rør diameter og rør afstand
- Vinkelforhold til himmel og jord
  - som afhænger af rørafstanden
- Udvnyttet energi fra direkte stråling
  - som afhænger af skyggerne fra nabogerne
- Udvnyttet energi fra diffus himmelstråling
  - som afhænger af vinkel forholdet til himlen
- Udvnyttet energi fra jordreflekteret stråling
  - som afhænger af vinkel forholdet til jorden og af refleksionscoefficienten
- Varmetab
  - som afhænger af varmetab fra glasrørne, manifolderne m.m.

Al teorien sammies i et simuleringsprogram således at den årlige termiske ydelse af vakuumrørssolfangene kan bestemmes.

Ingen teori uden målinger...

- Solfangerydelsen blev bestemt i en uendeligt prøvestand hvor følgende blev målt:
  - indløbstemperatur
  - udløbstemperatur
  - volumenstrøm
  - solstråling
  - omgivelsestemperatur

\[ P_u = \dot{V}\cdot p\cdot c_p\cdot \Delta T_{uul\cdot \text{varm} - T_{uul\cdot \text{kolde}}} + C_{solfanger} \cdot \frac{(T_{\text{middel}} - T_{\text{middel}})}{\Delta t} \]
Målinger

- Forsøgsbetingelser:
  - 2 perioder (11 dage)

Sammenligning mellem målinger og beregninger

- Model input:
  - Globalstråling, diffus stråling og omgivelststemperatur
  - Indåbstemperatur og volumenstrøm
  - Geometri, \( F' \) and \( \tau_0 \)

<table>
<thead>
<tr>
<th>Antal rør</th>
<th>( L ) [m]</th>
<th>( r_c ) [cm]</th>
<th>( R_p ) [cm]</th>
<th>( C ) [cm]</th>
<th>( k_0 ) [W/m²K]</th>
<th>( F' ) [-]</th>
<th>( \tau_0 ) [-]</th>
<th>( \theta ) [-]</th>
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</thead>
<tbody>
<tr>
<td>14</td>
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<td>2.09</td>
<td>0.98</td>
<td>0.856</td>
<td>3.6</td>
</tr>
</tbody>
</table>

- Model output:
  - Udløbstemperatur
  - Ydelse
Sammenligning mellem målinger og beregninger

- Der er en god overensstemmelse mellem målt og beregnet udløbstemperatur (og dermed også mellem målte og beregnede ydelser – mindre end 2% afvigelse)

Årsberegninger med den nye teori

- Lodrette rør
  - Teorien gælder kun for lodrette rør
- To lokationer:
  - Uummannaq (GL, 71°N)
  - København (DK, 56°N)
- Forvendede årsydelser:
  - Rør afstand
  - Solfanger azimuth
  - Sammenligning med Arcon HT
Først lidt om solstråling
i København og i Uummannaq

- Figuren viser summen af solstråling på forskiden
  og bagfoden af et ladet flask for forskellige
  orienteringer.
- For København er der
  (maksimale) symmetri
  omkring 0° (syd).
- I Uummannaq er der et
  minimum omkring −80°
  (mod øst).
- Ansagen er at der er et
  højere gennemsnit for
  Uummannaq, som
  reducerer solstrålingen
  som kommer fra øst.

- Solstrålingsværdierne er
  højere i Uummannaq fordi
  der er et soligt sted,
  fordi fladen er ladet og
  fordi der er meget sne.

- København: Årlig middeltemperatur: 7.8°C
- Uummannaq: Årlig middeltemperatur: −4.2°C

Solfangerazimut

- Figuren viser ydelsen pr. rør i et solfangerpanel, hvor rø refstanden er 6.7 cm
  (~2 cm luft mellem rørene) og solfangermiddeltemperatur er 50°C.
- I Uummannaq og i København, er den optimale solfangerorientering (azimut)
  omkring 45°-60°.
- Resultaterne skyldes både solstrålingsfordelingen (jf. forrige figur) og at
  temperaturen er højst om eftermiddagen.
Rørafstand

- Figureren viser ydelsen pr. m² solfangerpanel der vender mod syd, hvor rørafstanden varieres og solfangermiddletemperaturen er 50°C.
- Ydelsen stiger når centerørstand øges op til 0.2 m (~ 15 cm luft mellem rørene), hvilket skyldes at skyggerne fra naborørene mindskes.
- For endnu større afstande falder nettoydelsen igen, hvilket skyldes at stigende varmetab fra manifolddørene.

Årsydelser sml. med Arcon HT

- Figurerne viser solfangerydelsen pr. m² transparent areal som funktion af solfanger middletemperaturen.
- Vakuumrørrene er placeret helt tæt op ad kindergarten så det transparente areal for vakuumrørselseren kan sammenlignes med det transparente areal for Arcon HT.
- Arcon HT er bedst ydende i København.
- Vakuumrørselseren er bedst ydende i Uummannaq.
- Årsagen er at vakuumrørselseren ikke er optimalt holdt i København.
- Årsagen er også at der er meget mere solstråling fra "alle rørganger" i Uummannaq og den stråling kan udnyttes af vakuumrørselseren.
Konklusion

- Undersøgelserne drejer sig om vakuumsolfangere som kan udnytte solstråling fra "alle retninger"
- Dertil har vi udviklet en teori for beregning af vakuumsolfangernes ydelse:
  - højrestand = 0,2 m
  - Solfangerns afstand = 45°-60° mod vest
- Beregnede årsværdier viste:
  - København: Årsen HT højstefyldende
  - Uummannaq: Vakuumsolfangere højstefyldende
  - Årsag: Årsagen er at vakuumsolfangeren ikke er optimalt bevægede i København, og aldrer er meget mere
    solstråling fra "alle retninger" i Uummannaq og den stråling kan udnyttes af vakuumsolfangeren
- Solfangervejledning er lovende – iser for nordlige breddegrader
- Fortsat arbejde:
  - Teorien skal udfærdes til at kunne tage ikke-lokaltre nør i beregning
  - refleksioner mellem nærens skal inkluderes i teorien
  - flere effektive strålingsmodeller skal inkluderes i teorien
  - flere design skal undersøges (trætyper, koblingsprincipper, varmevekslingsprincipper m.m.)
  - flere målinger (både i Danmark og i Grønland)
  - undersøgelser af holdbarhed og påklædning
Bilag 7: Artikel optaget i det videnskabelige tidsskrift Journal of Solar Energy Engineering, Transactions of the ASME.
MODELLING SHADOWS ON EVACUATED TUBULAR
COLLECTORS WITH CYLINDRICAL ABSORBERS

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Abstract

A new TRNSYS collector model for evacuated tubular collectors with tubular absorbers is
developed. Traditional flat plate collector performance equations have been integrated
over the whole absorber circumference. On each tube the model determines the size and
position of the shadows caused by the neighbour tube.

An all glass tubular collector with tubular absorbers with 14 tubes connected in parallel is
investigated theoretically with the model and experimentally in an outdoor collector test
facility. Performance calculations with the new model are compared with measured results
and a good degree of similarity between the measured and calculated results is found.

Further, it is illustrated how the model can be used for geometrical parameter studies both
for constant collector mean fluid temperatures and for varying temperature conditions in a
solar heating plant. These investigations are performed for two climates: Copenhagen
(Denmark) and Uummannaq (Greenland).

Introduction

Evacuated tubular collectors (ETC) have an increasing share of the collector market in the
world. Up to 2001 more than 100 million m² collectors were installed world over. Of this,
about 28% were unglazed collectors, 49% were traditional flat plate collectors and about
22% were ETC [1].

As a consequence of this market development, almost all larger German and Austrian
solar thermal companies have ETC on their product list. Due to the market development,
also the theoretical modelling and simulation development becomes more and more
important.

Of course, thermal modelling of evacuated tubular collectors has previously been
included two incidence angle modifiers for the longitudinal and the transverse direction.
This model has widely been used, among others by, Qin L. and Furbo S. [3] who
investigated the performance of differently designed evacuated solar collectors. The model
cannot be used to find the efficiency expression and the incidence angle modifiers based
on the tube geometry and panel design, since the model does not directly take into
account shadow effects from adjacent tubes or solar radiation received from the back of
the collector.

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Perez et al. [4] developed a radiation model for evacuated tubular collectors with tubular absorbers. The model was based on the theory of one-axis tracking solar collector and did take into account solar radiation received from the back of the collector. Further the model gave an estimation of the size of the shadowed area of the total collector panel. However, the model did not determine the position of the shadows on each tube.

Lart S. [5] found a geometrical method to determine the size and position of the shadowed area on each tube. This information was used to modify the transverse incidence angle modifier so that the effective absorber area was taken into account.

Shah, L.J. and Furbo, S. [6] worked further on this principle and developed a new ETC simulation model. The model was developed for ETC with tubular collectors mounted on ground in large collector fields. The advantages of the model were that shadows, the solar radiation and the incidence angle modifier for each tube were precisely determined for all solar positions, including solar positions on the “back” of the collector. However, the model could be improved further as the model was only valid for vertical pipes and as the model was not developed for a commonly used simulation program.

In the present paper, this theory is further developed so it can simulate solar collector panels of any tilt and based on the theory a new TRNSYS [7] collector type is developed. This model is validated with measurements from outdoor experiments.

Some examples of the model calculations of shadows on the tubes will be shown. Further, TRNSYS simulations of the yearly thermal performance of a solar heating plant based on the evacuated solar collectors are carried out and it is investigated how the distance between tubes and the collector tilt influences the yearly thermal performance.
Collector performance theory for tubular absorbers

In Shah, L.J. and Furbo, S [6,8], a theoretical model for calculating the thermal performance of ETC with tubular absorbers was developed. The principle in the model was that flat plate collector performance equations were integrated over the whole absorber circumference. In this way, the transverse incident angle modifier was eliminated. The model was valid only for vertically tilted pipes.

In this section, the principle of the model will shortly be summarized. Further, the newest development that improves the model to be able to also take tilted pipes into calculation will be described.

In principle, the theory is suitable for all ETC without reflectors. However, in this case, the theory development is based on a collector design with a number of parallel-connected double glass tubes, which are open in both ends. The tubes are annuluses with closed ends and the outside of the inner glass wall is equipped with an absorbing selective coating. The collector fluid is floating from bottom to top of the inside of the inner tube where also another closed tube is inserted with the purpose to fill out a part of the tube volume so that less collector fluid is needed. Further, it ensures a high heat transfer coefficient from the inner glass tube to the collector fluid. Figure 1 shows the design of the evacuated tubes and the tube connection.

Generally, for a solar collector without reflectors and without parts of the collector reflecting solar radiation to other parts of the collector, the performance equation can be written as:

\[ P_i = P_{in} + P_{out} - P_{loss} \tag{1} \]

or more detailed described:

\[ P_i = A_s F\left(\theta\right) K_s R_s G_i + A_s F\left(\theta\right) K_{s,gr} F_{s,gr} G_{s,gr} + A_s F\left(\theta\right) K_{s,dir} F_{s,dir} G_{s,dir} - A_s U_i \left(T_{in} - T_o\right) \tag{2} \]

where \( K_s \) is the incident angle modifier defined as [9]:

\[ K_s = 1 - \tan^2 \left(\frac{\theta}{2}\right) \tag{3} \]

The incident angle modifiers for diffuse radiation, \( K_{s,dir} \), and ground reflected radiation, \( K_{s,gr} \), are evaluated by Eq. 3 using \( \theta = \pi/3 \).
To calculate the thermal performance of the evacuated tubes, the general performance Eq. (1) and Eq. (2) have been integrated over the whole absorber circumference. This means that the tube is divided into small “slices”, and each slice is treated as if it was a flat plate collector. In this way, the transverse incident angle modifier is eliminated. To describe the solar radiation on a tubular geometry, this method has previously been used by Pyrko J. [10].

Integrating over the absorber area, the performance equation can be described as:

$$\tau_n = \int_{\gamma_1}^{\gamma_0} \left( r_n - \frac{P_n}{P_{in}} - \frac{P_{in}}{P_n} \right) \, d\xi$$

(4)

where,

$$P_{in} = \frac{1}{2} A_n \int_{\gamma_1}^{\gamma_0} \left( T_{in} - T_c \right) \, d\xi = \int_{\gamma_1}^{\gamma_0} L \int_{\gamma_1}^{\gamma_0} \left( T_{in} - T_c \right) \, d\xi = 2\pi L \int_{\gamma_1}^{\gamma_0} \left( T_{in} - T_c \right) \, d\xi$$

(5)

$$P_r = \int_{\gamma_1}^{\gamma_0} A_n \left[ f^r(\alpha) \right] K_n \int_{\gamma_1}^{\gamma_0} G_n \, d\xi = 2\pi \int_{\gamma_1}^{\gamma_0} f^r(\alpha) K_n \int_{\gamma_1}^{\gamma_0} G_n \, d\xi$$

(6)

$$P_p = \int_{\gamma_1}^{\gamma_0} A_n \left[ f^p(\alpha) \right] K_n \int_{\gamma_1}^{\gamma_0} G_p \, d\xi = 2\pi \int_{\gamma_1}^{\gamma_0} f^p(\alpha) K_n \int_{\gamma_1}^{\gamma_0} G_p \, d\xi$$

(7)

$$G_n = \rho_n (G_0 + G_1)$$

(8)

$$F_{in} = 0.5 - F_{in}$$

(9)

$$F_{out} = 0.5 - F_{out}$$

(10)

**Power from beam radiation on collector tube, P_r:**

The power contribution from the beam radiation can be written as:

$$0 < \gamma_1 < \gamma_0: \quad P_r = \int_{\gamma_1}^{\gamma_0} \left[ f^r(\alpha) \right] K_n \int_{\gamma_1}^{\gamma_0} G_n \, d\xi = 2\pi \int_{\gamma_1}^{\gamma_0} f^r(\alpha) K_n \int_{\gamma_1}^{\gamma_0} G_n \, d\xi$$

(11)

$$0 < \gamma_0 < \pi: \quad P_r = \int_{\gamma_0}^{\gamma_0} \left[ f^r(\alpha) \right] K_n \int_{\gamma_1}^{\gamma_0} G_n \, d\xi = 2\pi \int_{\gamma_0}^{\gamma_0} f^r(\alpha) K_n \int_{\gamma_1}^{\gamma_0} G_n \, d\xi$$

Notice that there is now integrated over only a part of the circumference. This is because only part of the absorber surface is exposed to the beam radiation due to shadows from the neighbour tube. The task is now to determine the size of this area, thus determining the size and position of the shadowed area. In vector notation, the position of the sun can be described by:
\[ \vec{S} = \begin{bmatrix} \sin \theta_i \cos \gamma_i \\ \sin \theta_i \sin \gamma_i \\ \cos \theta_i \end{bmatrix} \tag{12} \]

and a "cross section circle" (see Fig. 2) on the absorber of one tube can be described by:

\[ \vec{N} = r \begin{bmatrix} \cos \left( \frac{\pi}{2} - \beta_i \right) \cos \gamma_i \\ \sin \gamma_i \\ \sin \left( \frac{\pi}{2} - \beta_i \right) \sin \gamma_i \end{bmatrix} \tag{13} \]

**Fig. 2: The solar vector, \( \vec{S} \), and the tube vector, \( \vec{N} \).**

Figure 3 shows an example where a part of one tube is shaded and a part is exposed to beam radiation. In order to determine the size of the area exposed to beam radiation, the points \( P_0 \) and \( P_1 \) must be determined.

Since \( P_0 \) is located where the solar vector and the tube vector are at right angles to each other, \( P_0 \), described by the angle \( \gamma_0 \), can be determined by the scalar product of the two vectors:

**Fig. 3: Illustration of the shaded area and the area exposed to beam radiation.**

\[ \vec{S} \cdot \vec{N} = |\vec{S}| |\vec{N}| \cos \left( \frac{\pi}{2} \right) = 0 \Rightarrow \sin \theta_i \cos \gamma_i \cos \left( \frac{\pi}{2} - \beta_i \right) \cos \gamma_i + \sin \theta_i \sin \gamma_i + \cos \theta_i \sin \left( \frac{\pi}{2} - \beta_i \right) \sin \gamma_i = 0 \Rightarrow \]

\[ \gamma_0 = -\arctan \left( \frac{\sin(\theta_i) \cos(\gamma_i) \cos(\frac{\pi}{2} - \beta_i) + \cos(\theta_i) \sin(\gamma_i)}{\sin(\theta_i) \sin(\gamma_i)} \right) \tag{14} \]

Since the equation for \( \gamma_0 \) involves the arctan function, the equation will return two solutions. Based on information on the position of the sun, the correct solution is found.
The point $P_1$, described by the angle $\gamma_1$, can be determined from the following Eq. (15), (16) and (17). A graphical illustration of symbols used in the equations can be seen in Fig. 3 and Fig. 4.

$$
P = \begin{pmatrix}
x_1 \\
y_1 \\
z_1
\end{pmatrix} = \begin{pmatrix}
x_s \\
y_s \\
z_s
\end{pmatrix} + \begin{pmatrix}
\sin \theta_s \cos \gamma_1 \\
\sin \theta_s \sin \gamma_1 \\
\cos \theta_s
\end{pmatrix} T
$$
(15)

$$
P = \begin{pmatrix}
x_1 \\
y_1 \\
z_1
\end{pmatrix} = \begin{pmatrix}
x_s \\
y_s \\
z_s
\end{pmatrix} + \gamma_1 \begin{pmatrix}
\cos \left( \frac{\pi}{2} - \beta_s \right) \cos \gamma_1 \\
\sin \gamma_1 \\
\sin \left( \frac{\pi}{2} - \beta_s \right) \sin \gamma_1
\end{pmatrix}
$$
(16)

$$
x_s = z_s \tan \left( \frac{\pi}{2} - \beta_s \right)
$$
(17)

Equations (15) (16) and (17) together give four equations to the four unknowns: $T$, $\gamma_1$, $x_s$, and $z_s$. Solving for $\gamma_1$ gives:

$$
\gamma_1 = \arctan \left( \frac{K_1 + 0.5 \frac{K_2}{K_1^2 + K_5} \left(-2K_3K_4 + 2K_3^{*} \right)}{0.5 \frac{K_2}{K_1^2 + K_5} \left(-2K_3K_4 - 2K_3^{*} \right)} \right)
$$
(18)

or

$$
\gamma_1 = \arctan \left( \frac{K_1 + 0.5 \frac{K_2}{K_1^2 + K_5} \left(-2K_3K_4 + 2K_3^{*} \right)}{0.5 \frac{K_2}{K_1^2 + K_5} \left(-2K_3K_4 - 2K_3^{*} \right)} \right)
$$
(18)

where $\arctan(X,Y)$ is defined as $\tan(Y/X)$ and

$$
K_1 = \frac{x_s}{\tan \left( \frac{\pi}{2} - \beta_s \right)} - \frac{y_s}{\tan \left( \frac{\pi}{2} - \beta_s \right) \tan(\theta_s) \sin(\gamma_1 - \gamma_1^*)} + z_s
$$

$$
K_2 = \frac{C}{\tan \left( \frac{\pi}{2} - \beta_s \right) \tan(\gamma_1 - \gamma_1^*) + \tan(\theta_s) \sin(\gamma_1 - \gamma_1^*)}
$$

$$
K_3 = C \left( \cos \left( \frac{\pi}{2} - \beta_s \right) + \frac{1}{\tan(\theta_s) \sin(\gamma_1 - \gamma_1^*)} \right)
$$

$$
K_4 = K_1^* - K_1 K_5 + K_1^* K_5^*
$$

From Eq. (18) it appears that there are two solutions for $\gamma_1$. Based on information on the position of $\gamma_0$, the correct solution is found.

**The incident angle, $\theta$, and the geometric factor, $R_0$**

The incident angle, $\theta$, can be described as:
\[
\cos(0) = \sin(\theta) \cos(\gamma) \cos\left(\frac{\pi}{2} - \beta\right) \cos(\gamma_{\text{rad}}) + \sin(\theta) \sin(\gamma - \gamma_r) \sin(\gamma_{\text{rad}}) + \cos(\theta) \sin\left(\frac{\pi}{2} - \beta\right) \cos(\gamma_{\text{rad}})
\]

(20)

The geometric factor, \( R_0 \), can be described as [11]:

\[
R_0 = \frac{\cos(0)}{\cos(\theta)}
\]

(21)

Solving the performance equation:

In order to evaluate the performance of the tubular collector on a yearly basis, the above theory is implemented into a Trnsys type. All the integrals can be solved analytically, except the integral in Eq. (11), which is solved by using the trapezoidal formula for solving integrals numerically. 360 integration steps are used in the numerical integration. Taking the collector capacity into account, the collector outlet temperature is evaluated by:

\[
P = \dot{V} \rho C_p \left( T_{\text{in},\text{hot}} - T_{\text{out},\text{rad}} \right) + \frac{C_p \Delta T_{\text{in},\text{out}}}{\Delta t}
\]

(22)

### Measurements and model validation

The model was validated with a prototype collector based on the collector principle shown in Fig. 1. Data of the collector is given in Table 1 and a photo of the collector is shown in Fig. 5.

The thermal performance of the collector was measured in an outdoor test facility where the inlet temperature, the outlet temperature and the volume flow rate was measured. The temperatures were measured with copper-constantan thermocouples (Type TT) and the volume flow rate was measured with a HGF1 flow meter. A 31% glycol/water mixture was used in the solar collector loop. Further, the global radiation and the diffuse radiation on horizontal were measured with two Kipp&Zonen CM5 pyranometers.

The collector performance was measured for two different tilts: 45° and 90° (both facing south). A period of 11 days (17/5-28/5 2003) has been selected for validating the Trnsys model for the collector at 45° and a period of 7 days (12/8-19/8 2003) has been selected for validating the Trnsys model for the collector at 90°.

<table>
<thead>
<tr>
<th>No. of pipes</th>
<th>[-]</th>
<th>14</th>
</tr>
</thead>
<tbody>
<tr>
<td>( L )</td>
<td>[m]</td>
<td>1.47</td>
</tr>
<tr>
<td>( r_0 )</td>
<td>[m]</td>
<td>0.0235</td>
</tr>
<tr>
<td>( r_p )</td>
<td>[m]</td>
<td>0.0165</td>
</tr>
<tr>
<td>( C )</td>
<td>[m]</td>
<td>0.067</td>
</tr>
<tr>
<td>( k_{\text{tube}} )</td>
<td>[W/m²K]</td>
<td>0.8</td>
</tr>
<tr>
<td>( k_{\text{manifold}} )</td>
<td>[W/K/(m manifold)]</td>
<td>0.134</td>
</tr>
<tr>
<td>( F^* )</td>
<td>[-]</td>
<td>0.98</td>
</tr>
<tr>
<td>( (10)_e )</td>
<td>[-]</td>
<td>0.856</td>
</tr>
<tr>
<td>( a )</td>
<td>[-]</td>
<td>3.8</td>
</tr>
<tr>
<td>( C_{\text{collector}} )</td>
<td>[kJ/K/tube]</td>
<td>1.9</td>
</tr>
</tbody>
</table>

**Table 1: Data describing the collector in the model.**
The necessary data for describing the collector are shown in Table 1. The heat loss coefficient, \( k_0 \), was determined from efficiency measurements [6] and split into two parts for the evacuated tubes and the manifold pipes respectively. \( F^* \) was calculated from theory [11,12] and \( (\tau \omega) \), and \( a \) were calculated with a simulation program for determining optical properties [13].

In Fig. 6 the measured and calculated collector outlet temperatures are compared. It can be seen that there is a good degree of similarity between the measured and calculated temperatures. The difference between measured and calculated outlet temperature, \( dT \), is always close to 0 K and the maximum temperature difference is about 3 K. Further Fig. 7 shows the measured and calculated collector performance for the two periods. The difference between the measured and calculated performance lies within the measuring inaccuracy of 4%.

![Fig. 6: Measured and calculated outlet temperature for the test periods](image1)

![Fig. 7: Measured and calculated collector performance for the test periods](image2)

**Calculation of the tube area exposed to beam radiation**

One of the features of the new model is that it determines the tube area exposed to beam radiation. This area corresponds to the size of the angle of the non-grey area on the middle pipe illustrated in Fig. 3.

For illustration purpose, Fig. 8 shows the angle on the tube exposed to beam radiation as a function of the solar azimuth and zenith, when the centre tube distance is 0.087 m, the outer tube radius is 0.0235 m and the inner tube radius is 0.0185 m.

Each sub-figure in Fig. 8 shows the angle exposed to beam radiation for a constant solar zenith, and each sub-figure contains six curves that represent different tube tilts from 0° to 75°. If one of these curves shows that the angle exposed to beam radiation is 180°, it means that there is no shadow from the adjacent tubes. Likewise, if one of these curves shows that the angle exposed to beam radiation is 0°, it means that there is full shadow from the adjacent tubes.
As expected, Fig. 8 shows that the smaller the zenith the larger the angle exposed to beam radiation. Also the figure shows that for small zenith angles, small tube tilts are preferable and vice versa.

Fig. 8: The angle on the tube exposed to beam radiation as a function of the solar azimuth and zenith, when the centre tube distance is 0.067 m.

To show the influence of the tube centre distance, Fig. 9 shows the angle on the tube exposed to beam radiation as a function of the solar azimuth and zenith, when the centre tube distance is now 0.047 m. This means that there is no air gap between the tubes. The outer and inner tube radiuses are still 0.0235 m and 0.0185 m.

Comparing Fig. 8 and Fig. 9 it becomes clear that the smaller the tube centre distance is the more shadow will come from the adjacent tubes. This is especially clear for the smallest zeniths. For example for a zenith of 29° there are only shadows on the tubes tilted 60° and 75° when the tube centre distance is 0.067 m whereas for a tube centre distance of 0.047 m there are shadow on all the tilted tubes.
Fig. 9: The angle on the tube exposed to beam radiation as a function of the solar azimuth and zenith, when the centre tube distance is 0.047 m.

Parameter variations
In this section, it is illustrated how the model can be used for geometrical parameter studies. In the following examples it is assumed that the ETC collector panel is operating in a solar heating plant at a constant temperature of 50°C throughout the year. The model data of the collector panel is given in Table 1. The collector performance is investigated for two climates, as summarized in Table 2. The albedo is set higher for Uummannaq during a large part of the year due to snow on the ground.
<table>
<thead>
<tr>
<th>Location</th>
<th>Latitude</th>
<th>Longitude</th>
<th>$T_{\text{average}}$</th>
<th>$G_{\text{sun}}$</th>
<th>$G_{\text{diff}}$</th>
<th>Albedo</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Copenhagen, Denmark</td>
<td>56</td>
<td>12</td>
<td>7.6</td>
<td>1002</td>
<td>510</td>
<td>0.2</td>
<td>[14]</td>
</tr>
<tr>
<td>Uummannaq, Greenland</td>
<td>71</td>
<td>52</td>
<td>-4.2</td>
<td>888</td>
<td>409</td>
<td>0.2 (15/9-14/6)</td>
<td>0.7 (15/9-14/6)</td>
</tr>
</tbody>
</table>

Table 2: Summarized data for the two locations.

The following parameters are analysed:
- Collector tilt and collector azimuth (geometry unchanged)
- Inner tube radius (outer tube radius constant)
- Tube centre distance (tilt and orientation unchanged)

Figure 10 shows the thermal performance per tube for Copenhagen and Uummannaq respectively. The figure clearly shows how the thermal performance decreases when the collector orientation deviates from south. The figure also shows that the optimum tilt is about 45° for Copenhagen and about 60° for Uummannaq.

![Yearly thermal performance](image)

Fig. 10: The thermal performance per tube, at a fluid mean temperature of 50°C, as a function of the collector tilt and orientation for Copenhagen (Left) and Uummannaq (Right).

Figure 11 shows the thermal performance per tube as a function of the inner tube radius for a collector fluid temperature of 50°C. The outer tube radius is kept constant 0.0235 m and the tube heat loss coefficient is assumed to vary linear with the absorber area. The figure shows that the thermal performance increases with the inner tube radius. The increase is not linear as the heat loss coefficient also increases with increasing inner tube radius.

For both locations, Fig. 12 shows the thermal performance per tube as a function of the tube centre distance for different collector fluid temperatures. The thermal performance increases for increasing tube centre distances up to a certain level, due to reduced shaded areas. For even larger distances the utilized energy decreases again, due to the
increasing heat loss from the manifold pipes. The optimal tube centre distance differs for different collector fluid temperatures. For a temperature of 70°C, the optimal tube centre distance is about 0.15 m and for a temperature level 30°C, the optimal tube centre distance is about 0.3 m. For higher temperature levels the heat losses from the manifold pipes becomes more important, and this explains the differences in the optimal tube centre distances.

Fig. 11: The thermal performance per tube as a function of the inner tube radius for a collector fluid mean temperature of 50°C.

Fig. 12: The thermal performance per tube as a function of the tube centre distance for different collector fluid mean temperatures.

Simulation of solar heating plants

Model description

To illustrate simulation examples when the collector fluid mean temperature is not constant throughout the year, a model of a solar heating plant is built in TRNSYS.

The collector array consists of 100 rows where the distance between the rows is assumed to be so large that the shadows between the rows have negligible influence on the collector performance. The energy consumption of a town is defined by a water mass flow rate, a return temperature and a flow temperature of 80°C. If the temperature from the solar heat exchanger is above 80°C the temperature is mixed down to 80°C with at three-way valve.

Fig. 13: Schematic illustration of the TRNSYS model.
If the temperature from the solar heat exchanger is below 80°C, an auxiliary boiler plant heats up the district heating water to 80°C.

An illustration of the TRNSYS model can be seen in Fig. 13 and Fig. 14 shows the mass flow rate and a flow and return temperature throughout the year for the district heating net of the town. The annual heat consumption of the town is about 32500 MWh.

Figure 14: Assumed flow rate and temperatures in the district heating net.

Tube distance, collector tilt and collector orientation

The optimum tube centre distance, collector tilt and orientation with respect the thermal performance per tube is investigated for the two locations. The gross collector area is assumed to be constant in the solar heating plant. Consequently, there are more tubes in the collector area when the tube distance is small than when the tube distance is large. Table 3 shows how the collector orientation, the tilt and the tube distance are varied.

<table>
<thead>
<tr>
<th>Collector azimuth [°]</th>
<th>-90 (east), 75, 60, 45, 30, 15, 0, 15, 30, 45, 60, 75, 90 (west)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Collector tilt [°]</td>
<td>15, 30, 45, 60, 75, 89</td>
</tr>
<tr>
<td>Tube centre distance [m]</td>
<td>0.048, 0.107, 0.167, 0.227, 0.287, 0.347</td>
</tr>
<tr>
<td></td>
<td>(corresponds to 1mm – 300 mm of air gap between the tubes)</td>
</tr>
</tbody>
</table>

Table 3: Overview of the parameter variations performed with the model.

Figure 15 and Fig. 16 show the thermal performance per tube for Copenhagen and Uummannaq respectively. The figures clearly show how the thermal performance increases with increasing tube centre distances up to about 0.3 m for Copenhagen and 0.27 m for Uummannaq. The increase is caused by less shadow from the adjacent tubes. Further, the temperature level in the collectors is increased with decreasing tube centre distances as there totally are more tubes in the collector field and this temperature increase also decreases the thermal output per tube.

The figures also show that the optimum tilt and orientation is about 45° south for Copenhagen and about 60° south for Uummannaq.
Conclusions

A new TRNSYS collector model for evacuated tubular collectors with tubular absorbers is developed. The model is based on traditional flat plate collector theory, where the
performance equations have been integrated over the whole absorber circumference. On each tube the model determines the size and position of the shadows caused by the neighbour tube as a function of the solar azimuth and zenith. This makes it possible to calculate the energy from the beam radiation.

The thermal performance of an all glass tubular collector with 14 tubes connected in parallel is investigated theoretically with the model and experimentally in an outdoor collector test facility. Calculations with the new model of the tubular collector vertically placed and tilted 45° is compared with measured results and a good degree of similarity between the measured and calculated results is found.

For two climates Copenhagen (Denmark) and Uummannaq (Greenland), it is illustrated how the model can be used for geometrical parameter studies.

For constant collector mean fluid temperature conditions throughout the year, the influence on the yearly thermal performance of the tube centre distance and the inner tube radius is investigated. These results show that to achieve the highest thermal performance, the tube centre distance must be about 0.2 m, and the inner tube radius should be as large as possible.

Also, the collector model is used in a model of a solar heating plant with varying temperatures and a sensitivity analysis of the tube centre distance, collector tilt and orientation with respect to the yearly thermal performance per tube is made. For the investigated load, the results show that the optimum tilt and orientation is about 45° south for Copenhagen and about 60° south for Uummannaq. These optima are not influenced by the average temperature level of the collectors.

References


Acknowledgement

This study is financed by the VILLUM KANN RASMUSSEN FOUNDATION.

Nomenclature

<table>
<thead>
<tr>
<th>LATIN SYMBOLS</th>
<th>Dimension</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
<td>[-]</td>
<td>Incident angle modifier</td>
</tr>
<tr>
<td>A_s</td>
<td>[m²]</td>
<td>Absorber area</td>
</tr>
<tr>
<td>A_b</td>
<td>[m²]</td>
<td>Absorber area exposed to beam radiation</td>
</tr>
<tr>
<td>C</td>
<td>[-]</td>
<td>Tube centre distance</td>
</tr>
<tr>
<td>C_f</td>
<td>[J/(kg K)]</td>
<td>Collector fluid heat capacity</td>
</tr>
<tr>
<td>C_p</td>
<td>[W/(m² K)]</td>
<td>Collector panel heat capacity incl. fluid</td>
</tr>
<tr>
<td>F</td>
<td>[-]</td>
<td>Collector efficiency factor</td>
</tr>
<tr>
<td>F_1–2</td>
<td>[-]</td>
<td>View factor from tube 1 to tube 2</td>
</tr>
<tr>
<td>F_s</td>
<td>[-]</td>
<td>View factor from tube to ground</td>
</tr>
<tr>
<td>F_s,t</td>
<td>[-]</td>
<td>View factor from tube to sky</td>
</tr>
<tr>
<td>G_s</td>
<td>[W/m²]</td>
<td>Beam radiation on horizontal</td>
</tr>
<tr>
<td>G_u</td>
<td>[W/m²]</td>
<td>Diffuse radiation on horizontal</td>
</tr>
<tr>
<td>G_r</td>
<td>[W/m²]</td>
<td>Ground reflected radiation on horizontal</td>
</tr>
<tr>
<td>k</td>
<td>[W/m K]</td>
<td>Heat loss coefficient per tube with respect to the absorber area</td>
</tr>
<tr>
<td>K</td>
<td>[W/m K]</td>
<td>Manifold heat loss coefficient per meter manifold pipe</td>
</tr>
<tr>
<td>K_1</td>
<td>[-]</td>
<td>Help variable</td>
</tr>
<tr>
<td>K_2</td>
<td>[-]</td>
<td>Help variable</td>
</tr>
<tr>
<td>K_3</td>
<td>[-]</td>
<td>Help variable for beam radiation</td>
</tr>
<tr>
<td>K_4</td>
<td>[-]</td>
<td>Incident angle modifier for diffuse radiation</td>
</tr>
<tr>
<td>K_5</td>
<td>[-]</td>
<td>Incident angle modifier for ground reflected radiation</td>
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<td>N</td>
<td>[-]</td>
<td>Tube vector</td>
</tr>
<tr>
<td>N</td>
<td>[-]</td>
<td>Number of tubes</td>
</tr>
<tr>
<td>L</td>
<td>[m]</td>
<td>Pipe length</td>
</tr>
<tr>
<td>P_b</td>
<td>[W]</td>
<td>Energy from beam radiation on collector/tube</td>
</tr>
<tr>
<td>P_d</td>
<td>[W]</td>
<td>Energy from diffuse radiation on collector/tube</td>
</tr>
<tr>
<td>P_a</td>
<td>[W]</td>
<td>Energy from ground reflected radiation on collector/tube</td>
</tr>
<tr>
<td>P_u</td>
<td>[W]</td>
<td>Useful energy from collector/tube</td>
</tr>
<tr>
<td>Q</td>
<td>[W]</td>
<td>Energy supplied from the boiler plant</td>
</tr>
<tr>
<td>Q_sung</td>
<td>[W]</td>
<td>Energy supplied to the town</td>
</tr>
<tr>
<td>R</td>
<td>[-]</td>
<td>Geometric factor, irradiance on a tilted surface divided by irradiance on a horizontal surface</td>
</tr>
</tbody>
</table>

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$r_o$ Outer glass tube radius [m]
$r_a$ Absorber radius [m]
$\beta$ Solar vector [-]
$T_a$ Ambient temperature [°C]
$T_m$ Fluid mean temperature [°C]
$T_{in,hot}$ Hot inlet temperature [°C]
$T_{out,cold}$ Cold outlet temperature [°C]
$T$ Help parameter [-]
$U_l$ Heat loss coefficient based on absorber area [W/(m²K)]
$y$ Collector volume flow rate [m³/s]
$x_i$ x coordinate for $P_i$ [m]
$x_0$ Help length [m]
$y_i$ y coordinate for $P_i$ [m]
$y_i$ y coordinate for $P_i$ [m]
$z_i$ z coordinate for $P_i$ [m]
$z_a$ Help length [m]
$z_i$ z coordinate for $P_i$ [m]

GREEK SYMBOLS:
$\beta_i$ Collector panel tilt [rad]
$\gamma_i$ Solar azimuth [rad]
$p$ Collector fluid density [kg/m³]
$\theta$ Incident angle [rad]
$\theta_i$ Solar zenith [rad]
$\tau_{m}$ Effective transmittance-absorptance product [-]
$\xi$ Integration variable [rad]
$\gamma_i$ Integration border [rad]
$\gamma_i$ Collector panel azimuth [rad]
$\gamma_{actual}$ Actual absorber azimuth [rad]
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Evacuated Tubular Collectors

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ABSTRACT
In this paper two research examples on evacuated tubular collectors are given. The first example concerns development of theoretical models for Heat pipe evacuated tubular collectors. In the second example heat transfer and flow structures inside All-glass evacuated tubular collectors for different operating conditions are investigated by means of Computational Fluid Dynamics (CFD).

1 INTRODUCTION
Solar energy is a clean and natural energy source. The solar radiation on earth – including at Arctic latitudes – is so large that it is possible to utilize solar energy in large scale.

The yearly number of hours with possibility of sunshine is almost independent of the latitude. However, Fig. 1 and Fig. 2 show that the distribution of the solar radiation on a monthly basis strongly depends on the latitude. At northern latitudes:

• more solar radiation occurs during the summer months
• the sun altitude is lower
• the day length variation is larger

For example, north of the Arctic Circle in the summer the sun is present 24 hours a day and solar radiation occurs from all directions during the 24 hours.

1.1 Evacuated tubular collectors
When solar collectors are developed for Arctic conditions, it is an advantage if the collector design can utilize solar radiation from all directions. Due to low ambient temperatures it is also an advantage if the collector has a low heat loss so that as little as possible of the absorbed solar radiation is lost to the surroundings. Further, the collector must be able to utilize ground reflected radiation, as the snow on the ground has a large reflection coefficient. Evacuated tubular collectors can be designed to fulfill these demands. Basically there exist two types of evacuated tubular collectors:

Fig. 1: The earth’s path around the sun.

Fig. 2: The day length as a function of latitude and the season.
Heat pipe evacuated tubular collectors consist of cylindrical evacuated glass tubes which are connected to a condenser/heat exchanger unit. Inside the evacuated tubes are the absorber fins with selective coatings on the surfaces and with a heat pipe, which contains the working fluid, for example water. The working fluid evaporates at a low temperature when the absorber is heated by the solar radiation. The evaporated fluid rises in the pipe and condenses on the condenser in the heat exchanger unit. Thus the energy is transferred to the solar collector fluid, which is pumped through the condenser/heat exchanger unit. When the working fluid in the heat pipe condenses, it drops down in the heat pipe again and the whole process is repeated if the temperature is high enough. Fig. 3 shows an evacuated glass tube with a heat pipe.

All-glass evacuated tubular collectors are designed in a different way. They are based on double glass tubes (see Fig. 4) with the evacuated space between the glass tubes. The outside of the inner glass wall is treated with an absorbing selective coating and works as the absorber. With solar irradiation on the tube, the inner glass tube gets very hot. The heat can be transferred from the inner glass tube to the solar collector fluid in different ways. Either the solar collector fluid is flowing directly inside the inner glass tube or the solar collector fluid can flow in a metal pipe, which is in good thermal contact with the inner glass tube.

Evacuated tubular collectors are suitable for Arctic conditions as:

- The collectors have a low heat loss coefficient. Due the evacuated space in both types of evacuated tubular collectors, the heat loss due to convection and conduction is very small. Therefore, the heat loss from evacuated tubular collectors is lower than the heat loss from traditional flat plate collectors.
- The collectors can utilize solar radiation from all directions. All-glass evacuated tubular collectors have cylindrical absorbers and in Heat pipe evacuated tubular collectors the absorber fin can have a curved shape, which follows the shape of the glass tube.
- The curved/cylindrical absorber shape can further utilize the ground reflected radiation better.

In order to develop optimum designed evacuated tubular collectors for Arctic conditions detailed background research is needed. In chapter 2 and 3 two research examples are given. The first example concerns development of theoretical models for Heat pipe evacuated
tubular collectors. In the second example heat transfer and flow structures inside All-glass evacuated tubular collectors for different operating conditions are investigated by means of Computational Fluid Dynamics (CFD).

2 RESEARCH EXAMPLE 1: HEAT PIPE EVACUATED TUBULAR COLLECTORS

In this example, two designs of heat pipe evacuated tubular collectors are investigated theoretically. The absorber fins in the evacuated tubes are either flat or curved and the fins have selective coating on both sides. This means that solar radiation from all directions can be utilized. An illustration of the evacuated tubes is given in Fig. 5. The tubes are connected to a heat exchanger manifold pipe where condensers for all tubes are placed.

Two new TrmSys [1] models for collectors with evacuated tubes with flat and curved fins are developed. The models take solar radiation from all directions into account. Further, due to the cylindrical tubes, depending on the position of the sun and the distance between the tubes, the tubes will be able to cast shadow on each other as illustrated in Fig. 6 and the solar irradiance can vary along the fin. For the curved fin model, the irradiance always varies along the fin as the incidence angle varies along the fin. Due to the variation in the solar irradiances along the fins, the traditional fin efficiency cannot be applied. Therefore, the heat transfer processes in the fin are solved in detail.

With the models, a parameter sensitivity analysis is carried out for the ground mounted evacuated tubular collectors installed in a solar heating plant. This analysis illuminates how the:
- different fin geometries
- collector tilt
- operating temperature
- tube distances
- distances between collector rows influence the yearly thermal performance of the collector field.

2.1 Theory and definitions

In this section, the theory of the two collector models, for flat and curved fins respectively, will be summarized. The models are developed for the TrmSys simulation program. The models include both the evacuated tubular collectors and the heat exchanger manifold pipe, as illustrated in Fig. 7.
The power, $P_u$, from this system can be written as:

$$ P_u = m_u c_u (T_{\text{manifold, outlet}} - T_{\text{manifold, inlet}}) \quad (1) $$

$$ P_u = P_{\text{ETC}} - P_{\text{loss}} - \frac{dQ_{\text{manifold}}}{dt} \quad (2) $$

The power from the heat pipes, $P_{\text{ETC}}$, the heat loss from the manifold tube, $P_{\text{loss}}$, and the energy change in the manifold tube, $dQ_{\text{manifold}}$, can be written as:

$$ P_{\text{ETC}} = UA_{\text{manifold}} (T_{\text{heat pipe}} - T_{\text{manifold}}) \quad (3) $$

$$ P_{\text{loss}} = U_{\text{manifold}} (T_{\text{manifold}} - T_{\text{sub}}) \quad (4) $$

with

$$ T_{\text{manifold}} = \frac{T_{\text{manifold, inlet}} + T_{\text{manifold, outlet}}}{2} \quad (5) $$

The power from the heat pipes, $P_{\text{ETC}}$, is larger than zero if the temperature of the heat pipe working fluid, $T_{\text{heat pipe}}$, is larger than the lowest evaporation temperature and larger than the mean temperature in the manifold, $T_{\text{manifold}}$. In other cases, $P_{\text{ETC}}$ is zero. In order to determine the temperature of the heat pipe working fluid, the fin is discretized into a number of elements as illustrated in Fig. 8. The energy balances for the elements of the fin are:

$$ i = 1: $$

$$ \frac{2S}{L} \frac{dT}{dx} \left( T_i - T_{i+1} \right) + S_i L dx - U_i L (T_i - T_{\text{sub}}) = \frac{m_{\text{fin}} c_{\text{fin}}}{dt} (T_i^{\text{new}} - T_i^{\text{old}}) \quad (6) $$

$$ 1 < i < n: $$

$$ \frac{2S}{L} \frac{dT}{dx} \left( T_i - T_{i+1} \right) = \frac{2S}{L} \frac{dT}{dx} \left( T_i - T_{i-1} \right) + S_i L dx - U_i L (T_i - T_{\text{sub}}) = \frac{m_{\text{fin}} c_{\text{fin}}}{dt} (T_i^{\text{new}} - T_i^{\text{old}}) \quad (7) $$

$$ i = n: $$

$$ \frac{2S}{L} \frac{dT}{dx} \left( T_i - T_{i-1} \right) - U A_{\text{manifold}} (T_i - T_{\text{manifold}}) + S_i L dx - U_i L (T_i - T_{\text{sub}}) = \frac{m_{\text{fin}} c_{\text{fin}}}{dt} (T_i^{\text{new}} - T_i^{\text{old}}) \quad (8) $$

Here, the solar radiation absorbed on the fin, $S_i$, is dependent on the position on the fin, due to varying incident angles (for curved fins) and due to possible shadows on the fin (for both flat and curved fins). The principles of determining the size and position on the fin of the shadows are described in [2].
The heat exchange capacity rate, $U_{\text{manifold}}$, has a constant value larger than zero if $T_b$ is larger than the lowest evaporation temperature and if $T_b$ is larger than the mean temperature in the manifold pipe. In other cases, $U_{\text{manifold}}$ is zero.

### 2.1.1 Solar radiation and view factors

The total solar radiation absorbed on the fin, $S_b$, can be written as:

$$S_b = S_{\text{dir, front}} + S_{\text{dir, back}} + S_{\text{diff, sky, front}} + S_{\text{diff, sky, back}} + S_{\text{diff, ground, front}} + S_{\text{diff, ground, back}}$$

where,

$$S_{\text{dir, front}} = (\tau_{\alpha})_d G_d K_d R_{\text{front}, R_{\text{sun}}}$$

$$S_{\text{dir, back}} = (\tau_{\alpha})_d G_d K_d R_{\text{back}, R_{\text{sun}}}$$

$$S_{\text{diff, sky, front}} = (\tau_{\alpha})_d G_d K_{\text{diff, sky}} F_{\text{sky, front}}$$

$$S_{\text{diff, sky, back}} = (\tau_{\alpha})_d G_d K_{\text{diff, sky}} F_{\text{sky, back}}$$

$$S_{\text{diff, ground, front}} = (\tau_{\alpha})_d G_d F_{\text{ground, front}} K_{\text{diff, ground}} F_{\text{ground, front}}$$

$$S_{\text{diff, ground, back}} = (\tau_{\alpha})_d G_d F_{\text{ground, back}} K_{\text{diff, ground}} F_{\text{ground, back}}$$

The incident angle modifier $K_\theta$ is calculated by:

$$K_\theta = 1 - \tan^2 \left( \frac{\theta}{2} \right)$$

Here, the incident angle, $\theta$, is defined as the incident angle on the absorber. The incident angle modifiers for diffuse radiation, $K_{\text{diff}}$, and ground reflected radiation, $K_{\text{diff, ground}}$, are evaluated by equation (14) using $\theta = \pm \alpha$ [3].

An illustration of the view factors is given in Fig. 9 and a more detailed description of the view factors including reduction of the view factors to ground and sky due to the collector rows, is given in [4].

### 2.1.2 Numerical issues

In order to evaluate the performance of the evacuated tubular collectors on an annual basis, the above theory is implemented into three TmSys types: two TmSys types for the two collector designs and an additional TmSys type for the view factor calculations. Concerning the two collector types, for each simulation time step the equations (1)-(8) are solved through an iteration loop. The iteration is stopped when the fin temperature difference from iteration to iteration is less than 0.00001 K and when the difference from iteration to iteration in power from the manifold is less than 0.01 W.
If the criteria are not reached the iteration loop stops after 100,000 iterations, and a warning is written to an output file. This seldom happens and the influence on the final result is insignificant. The fins are discretized into nine elements (18 elements in total), and a typical annual simulation of a collector array takes approximately 3 min. on a 2.8 GHz PC when a timestep of 0.05 h is used.

2.2 Parameter variations

In this section, it is illustrated how the model can be used for geometrical parameter studies. In the following examples it is assumed that the ground mounted evacuated tubular collector panels are operating in a solar heating plant at a constant operating temperature of 50°C in the heat exchanger manifold pipe throughout the year. The model data of the collector panel is given in Table 1. The collector performance is investigated for Uummannaq, Greenland, as summarized in Table 2. The albedo is set higher during a large part of the year due to snow on the ground.

Table 1: Data describing the collector in the model.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube length, $L$</td>
<td>[m]</td>
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</tr>
<tr>
<td>Glass tube radius, $r_g$</td>
<td>[m]</td>
<td>0.05</td>
</tr>
<tr>
<td>Absorber radius for curved fin, $r_a$</td>
<td>[m]</td>
<td>0.04375</td>
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<tr>
<td>Fin width for curved fin, $w_c$ (corresponding to a curved fin angle of 164°)</td>
<td>[m]</td>
<td>0.125</td>
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<tr>
<td>Fin width (for flat fin), $w_f$</td>
<td>[m]</td>
<td>0.0875</td>
</tr>
<tr>
<td>Fin thickness</td>
<td>[m]</td>
<td>0.0002</td>
</tr>
<tr>
<td>Fin conductivity, $k_{fin}$</td>
<td>[W/mK]</td>
<td>216</td>
</tr>
<tr>
<td>Tube centre distance, $C$</td>
<td>[m]</td>
<td>0.125</td>
</tr>
<tr>
<td>Tube heat loss coefficient, $h_{tube}$, based on absorber front side area</td>
<td>[W/m²K]</td>
<td>2.43</td>
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<tr>
<td>Effective transmittance absorbance product, $\alpha$</td>
<td>[-]</td>
<td>0.84</td>
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<tr>
<td>Incident angle modifier constant, $a$</td>
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<tr>
<td>Manifold heat loss coefficient, $k_{manifold}$</td>
<td>[W/Km]</td>
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</tr>
<tr>
<td>Manifold heat exchange rate per connection, $U/A_{manifold}$</td>
<td>[W/K]</td>
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</tr>
<tr>
<td>Operating temperature in manifold, $T_{manifold}$</td>
<td>[°C]</td>
<td>50</td>
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<tr>
<td>Collector heat capacity, $C_{collector}$</td>
<td>[kJ/K/tube]</td>
<td>1.9</td>
</tr>
<tr>
<td>Distance between collector rows</td>
<td>[m]</td>
<td>10</td>
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</table>

Table 2: Summarized data for Uummannaq, Greenland.

<table>
<thead>
<tr>
<th>Location</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Latitude</td>
<td>[°]</td>
<td>71</td>
</tr>
<tr>
<td>Longitude</td>
<td>[°]</td>
<td>52</td>
</tr>
<tr>
<td>$T_{average}$</td>
<td>[°C]</td>
<td>-4.2</td>
</tr>
<tr>
<td>$Q_{global}$</td>
<td>[kWh/m²]</td>
<td>888</td>
</tr>
<tr>
<td>$Q_{diffuse}$</td>
<td>[kWh/m²]</td>
<td>409</td>
</tr>
<tr>
<td>Albedo</td>
<td>[-]</td>
<td>0.2 (15/6-14/9) or 0.7 (15/9-14/6)</td>
</tr>
</tbody>
</table>
2.2.1 View factors

To get an understanding of the geometry included in the models, illustrations of some of the view factors for different conditions is given below. The view factors are calculated for a collector tilt of 60° and for infinite distance between the collector rows. For different tube centre distances Fig. 10 and Fig. 11 shows the view factor from one tube to the two neighbour tubes, respectively, and the view factors to the sky and ground from the collector’s front and back side respectively.

For the collector front side, it can be seen that the flat fin collector has a larger view factor to the sky, a smaller view factor to the ground and a smaller view factor between the tubes compared to the curved fin collector. This is because parts of the curved fin – due to the curve – face the neighbour tubes and the ground more than the flat fin. Further, as expected, the view factors between the tubes decreases with increasing tube centre distances.

For the collector back side, it can be seen that there is almost no difference between the curved and the flat fin. Here the reason is that, when calculating the view factors, the back side of the curved fin can be treated as a flat fin placed in the opening of the curved fin.

Fig. 12 shows the yearly diffuse sky and ground reflected radiation absorbed by the fin per m² front side absorber.

For the collector back side, it can be seen that, just as for the view factors, there is almost no differences between the curved and the flat fin. For the collector front side it is clear that the flat fin absorbs more diffuse radiation than the curved fin.

Finally, Fig. 13 shows the direct radiation absorbed by the front side of the fins. As expected, due to the view factors between fins and sky, the flat fin absorbs more direct radiation per m² front side absorber on a yearly basis. As it looks like the flat fin collector is more efficient, it must be pointed out that the absorbed energy quantities are given per m².
absorber area and that the curved fin absorber area is 45% larger than the flat fin absorber area.

2.2.2 Temperature profiles on the fin
Before going into thermal performance analyses, some examples of the temperature distribution on the fins are given. For a summer day (31/7), Fig. 14 and Fig. 15 show the temperature profile at solar time 9 AM, 12 PM and 3 PM for the curved and the flat fin respectively. At these three times there are no shadows from the neighbour tubes. The temperature distribution on the curved fin is symmetrical around the heat pipe at 12 PM whereas the east side of the fin is warmer during the morning and the west side of the fin is warmer during the afternoon.

This clearly illustrates the influence of the distribution of the solar radiation on the fin and it explains why the traditional fin efficiency, $F$, cannot be applied when analysing this type of collector in detail. As expected for the flat fin, there is symmetry in the temperature distribution at all three times. The differences in the temperature level for the three times are due to the weather conditions.

2.2.3 Collector tilt
Fig. 16 shows the thermal performance per tube as a function of the collector tilt. The figure shows that the optimum tilt is about 45° for the flat fin collector and slightly higher for the curved fin collector. The reason is most likely related to the differences in the view factors for the two collectors. As the curved fin collector, due to the design, sees more of the ground compared to the flat fin collector, the curved fin collector can utilize the ground reflected radiation better than the flat fin collector and consequently the optimum tilt is larger for the curved fin collector than for the flat fin collector. The curved fin collector performs better than the flat fin collector as the curved fin absorber area is larger than the flat fin absorber area.

2.2.4 Operating temperature
Fig. 17 shows the thermal performance as a function of the operating temperature in the manifold tube.

![Fig. 14: The temperature distribution on the curved fin in the morning, at noon and in the afternoon of a summer day.](image)

![Fig. 15: The temperature distribution on the flat fin in the morning, at noon and in the afternoon of a summer day.](image)

![Fig. 16: The thermal performance per tube as a function of the collector tilt.](image)
As expected the thermal performance decreases with increasing temperature level due to the increasing heat loss. Further, it can be seen that the curves are about to cross at a temperature of 90°C. The reason is that the heat loss is larger for the curved fin collector compared to the flat fin collector as the heat loss coefficient (the same for the two collectors) is based on the absorber front side area and the curved absorber is larger than the flat fin absorber.

2.2.5 Tube centre distances

Fig. 18 shows the thermal performance per tube as a function of the tube centre distance. The thermal performance increases for increasing tube centre distances up to about 0.3 m, due to reduced shaded areas, reduced view factors between the tubes and thus increased view factors to sky and ground. For even larger distances the utilized energy decreases again, due to the increasing heat loss from the manifold pipes.

2.2.6 Collector rows distances

Finally, Fig. 19 shows the thermal performance per tube as a function of the distance between the collector rows. It can be seen that thermal performance increases rapidly when the row distance increases from 1 m to 10 m. Further increase in the row distance has less impact on the thermal performance. The reason for the increase in thermal performance is that with increasing distances the shadows from neighbour rows on the collector (see Fig. 9 bottom) decreases and the view factor from the collector to ground and sky increases. It must be noticed that the heat loss in the pipes connecting the collector rows is not included in this analysis.

2.3 Summary

Two designs of ground mounted heat pipe evacuated tubular collectors operating in a solar heating plant are investigated theoretically. The absorber fins inside the evacuated tubes are either flat or curved and the surfaces of the fins have selective coating on both sides. Two new TrnSys models for evacuated tubular collectors are developed. The models calculate in detail the heat transfer processes of the absorber fins.

It is illustrated how the model can be used for geometrical parameter studies. For example, it is investigated how fin geometry, collector tilt, operating temperature, tube distances and distances between collector rows influences the yearly thermal collector performance.
3 RESEARCH EXAMPLE 2: ALL-GLASS EVACUATED TUBULAR COLLECTORS

All-glass evacuated tubular collectors are based on double glass tubes where the outside of the inner glass wall is treated with an absorbing selective coating and the evacuated space is between the tubes as illustrated in Fig. 20.

A collector design based on horizontal tubes connected to a manifold pipe is especially popular due to its low cost. An illustration of the collector design is shown in Fig. 21.

The collector fluid enters the bottom of the square manifold channel and leaves at the top of the manifold channel. The intended flow inside the glass tubes is indicated with the arrows. The flow is primarily naturally driven, as the walls of the tubes are hot due to the solar radiation. However it is unclear, how the operating conditions and the collector geometry influence the flow structures in the tubes and thus the collector performance.

The objective of this work is to investigate the heat transfer and the flow structures inside the tubes for different flow rates and collector geometries by means of Computational Fluid Dynamics (CFD).

3.1 Numerical investigations

To solve the flow and energy equations in the glass tubes, a simulation model of the flow in the tubes is developed using the CFD code Fluent 6.1 [6]. As illustrated in Fig. 22, only one section of the collector with two horizontal tubes placed in a vertical plane is investigated.

Steady state numerical solutions are obtained for laminar flow with the Boussinesq approximation for buoyancy modelling. The velocity-pressure coupling is treated by using the SIMPLE algorithm and the First Order Upwind scheme is used for the momentum and energy terms.

3.1.1 Geometry

The model consists of the inner boundaries of the geometry. The outer glass tube, the evacuated space between the two glass tubes and the wall thickness of the inner glass tube are
not included in the model. Also the outer casing and the insulation material of the manifold channel are not included in the model. This means that no solids are simulated – only the fluid is included in the model. The conduction in the inner glass wall is however included in the model. The geometry is summarized in Table 3.

The computational mesh is constructed in the pre-processing program Gambit 2.0.4 [7]. The number of computational cells depends on the length of the tubes and is given in Table 3. Fig. 23 shows a close up of the mesh near the manifold channel.

### 3.1.2 Boundary conditions

The solar irradiance is simulated as a distributed heat flux on the tube wall. The flux varies from 0 W/m² to 1150 W/m² as shown in Fig. 24. The average flux on the tube wall is 566 W/m².

The heat loss from the tubes is modelled with a heat loss coefficient of 0.85 W/m²K [8] and a constant ambient temperature of 293 K. The heat loss coefficient is related to the absorber area. The manifold channel is assumed to have zero heat loss.

Five different inlet mass flow rates of respectively 0.05 kg/min, 0.4 kg/min, 1 kg/min, 3 kg/min and 10 kg/min have been computed.

The inlet velocity profile has been found by first making a computation with a uniform inlet velocity profile. The outlet velocity profile from this simulation has then been used as the inlet velocity profile for the final simulation.

The inlet temperature has been 333 K during all computations. A 40% propylene-glycol/water mixture has been used as working fluid.

### 3.2 Results

The presented results will include illustrations of flow patterns in the vertical centre plane of the model near the manifold channel and in the manifold outlet plane. Further, some overall analyses of the “collector” performance as a function of tube lengths and mass flow rates will be presented.

---

**Table 3: Geometry of the Numerical Models**

<table>
<thead>
<tr>
<th>Side length of manifold channel:</th>
<th>0.06 m</th>
</tr>
</thead>
<tbody>
<tr>
<td>Glass tube inner diameter:</td>
<td>0.037 m</td>
</tr>
<tr>
<td>Glass tube outer diameter (used in manifold channel):</td>
<td>0.047 m</td>
</tr>
<tr>
<td>Length of glass tube exposed to solar radiation:</td>
<td>0.59 m, 1.17 m, 1.47 m</td>
</tr>
<tr>
<td>Number of computational cells:</td>
<td>428682, 738708, 948490</td>
</tr>
</tbody>
</table>

Illustration of geometry:

Fig. 23: The mesh near the square manifold channel.

Fig. 24: Distribution of the heat flux on the inner glass tubes.
3.2.1 Flow distribution

For a tube length of 1.17 m, the left column in Fig. 25 shows velocity vectors in the vertical tube centre plane near the manifold channel for the five investigated inlet mass flow rates.

At the smallest mass flow rate (top-left picture) it can be seen how the fluid flows directly from the inlet in the manifold channel out in the two horizontal tubes. The fluid returns to the manifold channel along the top of the tubes. The two flows from the two tubes exits at the outlet with a profile, which is clearly formed by the flows in the tubes.

The flow patterns for the next two flow rates look similar to the flow pattern for the lowest flow rate; however, there is one significant difference. Due to the larger inlet velocities, the flow rises higher in the manifold channel before it, due to buoyancy forces, turns down to the tube bottom wall and flows out in the tubes. The differences in the forced- and buoyancy driven flows are evident just by seeing how far up in the manifold tube the flow rises.

For the two highest flow rates it is clear that some of the flow passes directly through the manifold channel without entering the tubes. Therefore, the outlet velocity profiles look different for the highest flow rates compared to the outlet velocity profiles for the lowest flow rates.

The right column in Fig. 25 shows the velocity contours in the outlet plane. It can be seen how the velocity pattern changes from being dominated by the two flows from the tubes for the lowest inlet flow rates to, for the highest flow rates, being dominated mainly by the inlet flow.

<table>
<thead>
<tr>
<th>Velocity vectors in the vertical tube centre plane:</th>
<th>Velocity contours in the outlet plane:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow=0.05 kg/min.</td>
<td>Flow=0.05 kg/min.</td>
</tr>
<tr>
<td></td>
<td>Flow=0.4 kg/min.</td>
</tr>
<tr>
<td></td>
<td>Flow=1 kg/min.</td>
</tr>
<tr>
<td></td>
<td>Flow=3 kg/min.</td>
</tr>
<tr>
<td></td>
<td>Flow=10 kg/min.</td>
</tr>
<tr>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Fig. 25: Velocity patterns in the vertical tube centre plane (left column) and in the outlet plane (right column). Tube length: L=1.17 m.
3.2.2 Thermal performance

The thermal performance is investigated by calculating efficiencies for the different combinations of flow rates and tube lengths. The efficiency, \( \eta \), is defined as the ratio between the power out of the collector, \( P_{\text{collector}} \), and the distributed heat flux absorbed by the collector, \( P_{\text{absor}} \):

\[
\eta = \frac{P_{\text{collector}}}{P_{\text{absor}}}
\]

Notice that this efficiency cannot be compared with the traditional way of defining the collector efficiency as optical losses are not included in the efficiency used here.

Fig. 26 shows the efficiency as a function of the mass flow rate. The efficiency is highest for flow rates around 0.4 kg/min – 1 kg/min. The explanation for this result can be found in Fig. 27, which shows the mean temperature in the collector as a function of the mass flow rate. For the largest inlet flow rates (3 kg/min – 10 kg/min) a large part of the fluid flows directly through the manifold channel leaving only a smaller part flowing out in the tubes. Therefore, the average temperature in the whole collector rises. This leads to a higher heat loss and thus a lower efficiency. For the lowest flow rate (0.05 kg/min) almost all the inlet flow goes out in the tubes, but now the flow is so small that this alone leads to an increased average temperature in the whole collector.

That the efficiency in fact decreases with increasing average temperatures in the whole collector is very clear in Fig. 28. Here each dot represents an efficiency found at a given inlet flow rate. All the dots together form an almost straight line with a tilt that shows how the efficiency decreases with increasing heat loss caused by increasing average temperatures in the whole collector. The collector with the shortest tube has the highest efficiency and vice versa.

Finally, Fig. 29 shows the efficiency as a function of the average of the inlet- and outlet temperature. As in Fig. 26 and in Fig. 28 it can be seen that the highest efficiency is achieved for the shortest tube length.

The results further show that the inlet mass flow rate has a relatively small influence (1-2 %) on the resulting efficiencies. This might seem strange considering the large differences in the flow patterns near the manifold channel (Fig. 25).
The main reason is that the flow in the tubes at a distance from the manifold channel is relatively unaffected by the inlet flow.

For a tube length of \( L = 1.17 \text{ m} \), Fig. 30 shows the maximum velocity magnitude as a function of the distance from the manifold centre. For the varying inlet flows, the figure shows that close to the manifold channel there are differences in the maximum velocity magnitude but further out in the tubes there are almost no differences in the maximum velocity magnitude. This explains why the absolute differences in the collector efficiencies are so small for the varying inlet flow conditions. Fig. 30 also shows that the flows out in the tubes are:
- largest for an inlet flow of 0.4 kg/min
- 2\text{nd} largest for an inlet flow of 1 kg/min
- 3\text{rd} largest for an inlet flow of 3 kg/min
- 2\text{nd} smallest for an inlet flow of 0.05 kg/min and
- smallest for an inlet flow of 10 kg/min

This corresponds nicely with the results presented in Fig. 26 where the order of efficiencies is the same.

Finally, it should be mentioned that another reason for the small differences in the calculated efficiencies is that the heat loss coefficient for the tubes is very small. Due this low heat loss coefficient, the differences in the mean temperature in the whole collector have an only minor influence on the final efficiency.

3.3 Summary

Heat transfer and flow structures inside All-glass evacuated tubular collectors for different operating conditions are investigated by means of Computational Fluid Dynamics (CFD). The investigations are based on a collector design with horizontal tubes connected to a vertical manifold channel. Three different tube lengths varying from 0.59 m to 1.47 m have been modelled with five different inlet mass flow rates varying from 0.05 kg/min to 10 kg/min with a constant inlet temperature of 333 K. Under these operating conditions the results showed that:
- the collector with the shortest tube length achieved the highest efficiency
- the optimal inlet flow rate was around 0.4-1 kg/min
- the flow structures in the glass tubes were relatively uninfluenced by the inlet flow rate

Generally, the results showed only small variations in the efficiencies. This indicates that the collector design is well working for most operating conditions.
4 CONCLUSION AND OUTLOOK

In this paper two research examples on evacuated tubular collectors are given. The first example concerns development of theoretical models for Heat pipe evacuated tubular collectors. In the second example heat transfer and flow structures inside All-glass evacuated tubular collectors for different operating conditions are investigated by means of Computational Fluid Dynamics (CFD).

For the heat pipe investigations, ground mounted Heat pipe evacuated tubular collectors operating in a solar heating plant are investigated theoretically. It is illustrated how the developed model can be used for geometrical parameter studies. For example, it is investigated how fin geometry, collector tilt, operating temperature, tube distances and distances between collector rows influences the yearly thermal collector performance.

The All-glass investigations were based on a collector design with horizontal tubes connected to a vertical manifold channel. With different glass tube lengths and for varying flow rates it was shown that the collector with the shortest tube length achieved the highest efficiency and the optimal inlet flow rate was around 0.4-1 kg/min. However, generally all the All-glass results showed only small variations in the efficiencies. This indicates that the collector design is well working for most operating conditions.

Further work:

Parallel to the theoretical work, the investigated Heat pipe and All-glass evacuated tubular collectors will be tested side-by-side in an outdoor test facility. Among other things, the measured performances will be used to a final validation of the theoretical models and the thermal performance of differently designed evacuated tubular collectors will be compared. Based on the findings and on economy considerations, optimum designed evacuated tubular collectors for Arctic latitudes will be recommended.

5 ACKNOWLEDGEMENTS

This study is financed by the VILLUM KANN RASMUSSEN FOUNDATION.

6 REFERENCES

7 NOMENCLATURE

\( a \) Incident angle modifier constant, -
\( C_{c} \) Tube centre distance, m
\( C_{\text{collector}} \) Collector heat capacity, J/K/tube
\( c_{\text{fin}} \) Heat capacity of fin material, J/kgK
\( c_{\text{m,air}} \) Average heat capacity of fin material and working fluid in heat pipe, J/kgK
\( dQ_{\text{manifold}} \) Energy change in the manifold pipes, J
\( dt \) Time step, s
\( dx \) Width of discretization element, m
\( F_{\text{col,dir,ground,from}} \) View factor from the collector back side to the part of the ground with diffuse solar radiation, -
\( F_{\text{col,dir,ground,from}} \) View factor from the collector front side to the part of the ground with diffuse solar radiation, -
\( F_{\text{col,dir,ground,back}} \) View factor from the collector back side to the part of the ground with direct solar radiation, -
\( F_{\text{col,dir,ground,front}} \) View factor from the collector front side to the part of the ground with direct solar radiation, -
\( F_{\text{col,sky,back}} \) View factor from the collector back side to the sky, -
\( F_{\text{col,sky,from}} \) View factor from the collector front side to the sky, -
\( F_{\text{ground,sky}} \) View factor from the ground to the sky, -
\( F_{\text{tube,from}} \) View factor from one tube to the neighbour tubes, -
\( G_{b} \) Beam radiation on horizontal, W/m²
\( G_{d} \) Diffuse radiation on horizontal, W/m²
\( G_{\text{diffuse}} \) Yearly diffuse radiation on horizontal, kWh/m²
\( G_{\text{global}} \) Yearly global radiation, kWh/m²
\( K_{\text{a}} \) Incident angle modifier for direct radiation, -
\( K_{\text{dir}} \) Incident angle modifier for diffuse radiation, -
\( K_{\text{hr}} \) Incident angle modifier for ground reflected radiation, -
\( L \) Tube length, m
\( m_{\text{fin}} \) Mass of discretized fin element, kg
\( m_{\text{fin,air}} \) Average mass of fin material and working fluid in heat pipe, kg
\( P_{\text{EPC}} \) Power from heat pipes, W
\( P_{\text{h}} \) Heat loss from manifold pipe, W
\( P_{\text{u}} \) Power from collector, W
\( P_{\text{collector}} \) Power from collector, W
\( P_{\text{sun}} \) Solar radiation absorbed by absorber, W
\( R_{\text{e,front}} \) Geometric factor, -
\( R_{\text{e}} \) Glass tube radius, m
\( R_{\text{f}} \) Absorber radius for curved fin, m
\( R_{\text{u}} \) Part of tube irradiated by direct radiation due to row shadows, -
\( S_{i} \) Solar radiation absorbed on element i, W/m²
\( S_{\text{sky,from}} \) Diffuse sky radiation absorbed on the absorber front side, W/m²
\( S_{\text{sky,back}} \) Ground reflected diffuse radiation absorbed on the absorber back side, W/m²
\( S_{\text{sky,from}} \) Direct radiation absorbed on the absorber front side, W/m²
\( S_{\text{sky,back}} \) Direct radiation absorbed on the absorber back side, W/m²
\( S_{\text{hr,front}} \) Ground reflected direct radiation absorbed on the absorber front side, W/m²
\( S_{\text{hr,back}} \) Ground reflected direct radiation absorbed on the absorber back side, W/m²
\( T_{\text{av,average}} \) Yearly average ambient temperature, °C
\( T_{\text{amb}} \) Ambient temperature, °C
\( T_{\text{amb}} \) Ambient temperature, °C
\( T_{\text{pipe}} \) Temperature of the heat pipe working fluid, °C
\( T_{i} \) Temperature of element i, °C
\( T_{i+1} \) Temperature of element i+1, °C
\( T_{i-1} \) Temperature of element i-1, °C
\( T_{i,\text{new}} \) New temperature of element i, °C
\( T_{i,\text{old}} \) Old temperature of element i, °C
\( T_{\text{manifold}} \) Mean temperature in manifold, °C
\( T_{\text{manifold,from}} \) Inlet temperature to manifold, °C
\( T_{\text{manifold,out}} \) Outlet temp. from manifold, °C
\( U_{\text{manifold}} \) Manifold heat exchange rate, W/K
\( U_{\text{f}} \) Tube heat loss coefficient, W/m²K
\( U_{\text{f,loss}} \) Manifold heat loss coeff., W/K
\( W_{i} \) Fin width for curved fin, m
\( W_{f} \) Fin width for flat fin, m
\( \delta \) Fin thickness, m
\( \lambda \) Fin thermal conductivity, W/mK
\( p \) Ground albedo, -
\( \tau_{\text{col}} \) Transmittance absorbance prod., -
Bilag 9: Artikel optaget i Installationsnyt - Specialhæfte nr. 46/2005. TechMedia A/S.
Energi i kulden

Vakuumrørsfanger til Arktis

Af Louise Jivan Shah, Associated Research Professor, PH.D., DTU


Solstråling ved nordlige breddegrader

Solenergi er den reneste og naturligste energiform, vi overhovedet har. Solindfaldet er så stort på kilde - og i artiklikne egne - at det er mulighed for at undreg solenergi i stort omfang. Det årlige antal timer med mulighed for solene er stort set det samme, uanset hvor på kloden vi befinder os. Figur 1 og figur 2 viser til gengæld, at fordelen af solindfaldning over årets måneder afhænger stærkt af breddegraden. Jo højere mod nord vi befinder os, des større del af solindfaldet finder sted i sommermånedene.


Vakuumrørsfanger

Når et solfangerdesign skal udvikles til artiklikne breddegrader, må det således være en fordel, at designet bedst muligt udnytter solens bane. På grund af meget lav temperaturer er det også en fordel, hvis solfangeren har en lav varmeabsorptionskoefficient, således at mindst muligt af den absorberede solenergi tabes til omgivelserne. Endelig må solfangeren være velbygget til at opfange reflekteret solstilling, hvilket er på grund af sine er meget af på de artiklike breddegrader.

Vakuumrørsfangerne kan designes, så de opfylder disse krav og de kan udformes på forskellige måder.

Heat Pipe vakuumrørsfanger

Heat Pipe vakuumrørsfanger består af en række cylinderformede glasrør, som svæver eller er koblet til en kondensator/varmeanlægs- enhed. Inde i disse glasrørne med vakuum er placeret absorberende med selektivt begrensning og et væske, der indeholder en væske transportørende medium, f.eks. vand.

Det væske transportørende medium fordamer ved et lavt temperaturniveau, så absorberen opvarmes af solens stråler.

Dampen stiger opad i røret til kondensatoren, hvor dampen kondenserer og derved afgiver varme til solfangeren, som pumpes gennem kondensator/varmeanlægs-enhed.

I kondensatoren kondenserer det varme transportørende medium...
All-glass vakuumrørsolfangere

All-glass vakuumrørsolfangere er udviklet i en anden og lidt mere simpel måde end heat-pipe vakuumrørsolfangere. De er baseret på dobbeltglasere (se figur 4) med vacuum i mellemrummet mellem glasere.

De udvendige overlader af de indre glasere har en høj absorptværdi, og der er et lavt emittance, og virker som solfangers rent absorber.

Tabellen viser data (tavle) for København og Uummannaq.

<table>
<thead>
<tr>
<th>Klima</th>
<th>Solopvæsning</th>
<th>Global solenergi på vind</th>
<th>Udenventil</th>
<th>Absorption</th>
<th>Omgivelsernes refleksionskoefficient</th>
</tr>
</thead>
<tbody>
<tr>
<td>København</td>
<td>66</td>
<td>22</td>
<td>7.8</td>
<td>1062</td>
<td>510</td>
</tr>
<tr>
<td>Uummannaq</td>
<td>71</td>
<td>17</td>
<td>-4.2</td>
<td>809</td>
<td>409</td>
</tr>
</tbody>
</table>

Figur 2.

Når solen skinner på glasereet, bliver det indvendige glasere absorberende på grund af konvektion og varmeledning meget lille. Varmeabsorptiviteten for all-glass rørsolfangere er derfor meget mindre end varmeabsorptiviteten for almindelige plane solfangere.

- Solfangersignaler kan udnytte solens base om sommeren, hvor solen kommer fra alle retninger gennem døgnet. All-glass vakuump unfoldervængere har cylinderformet absorberende, og i heat-pipe vakuumrørsolfangere kan absorberende høj, så de følger gennem slutning.

- En rørsolfangere der gør glas, at solfangere form af glas.

Tabellen viser data (tavle) for København og Uummannaq.

Teoretiske modeller

I forskningsprojekterne tænkes der at udvikle modeller til beregning af termiske ydelser for vakuumrørsolfangere, der udnytter solenergi fra solenhug. Traditionelle solfangerskovler af litteratur er udviklet med henblik på almindelige plane solfangere med plane absorberende. Disse teorier har ikke direkte knude, der kan anvendes i forbindelse med vakuump unfoldervængere, da absorberende er cylinderformede.

Derfor er der udviklet to nye teoretiske solfangermodeller til vakuumrørsolfangere med cylinderformede absorberende.

Modellerne er udviklet til hhv. All-glass og Heat-pipe vakuumrørsolfangere. Baggrundstavlen kan studeres i [1, 2, 3]. Nedenfor er et eksempel på
anvendelsen af All-glass sollangermodellen.

**Modellering af All-glass vakuumrørsollarer**

Modellen tager udgangspunkt i den traditionelle plane sollanger teori, som integreres over den cylinderformede absorber.

Modellen kan præcist bestemme skyggeeffekterne fra rør til rør (se figur 5), ligesom den kan regne på, hvordan sollangeren udfylder solstrålingen fra alle kompassret retninger.

Beregninger med den teoretiske sollangermodel er sammenholdt med målinger på en prototype sollanger og det viser sig, at modellen gengiver 'virkeligheden' med stor nøjagtighed.

Modellen er heretter videreudviklet, så den nu kan indgå i simuleringsprogrammet TRNSYS [4]. Dette amerikanske simuleringsprogram er et komponentbaseret program, som er det mest anvendte og anerkendte simuleringsprogram til solvarmeanlæg.

**Geometriske parametre**

Nedenfor illustreres det, hvordan All-glass modellen kan benyttes til f.eks. geometriske parameterundersegelser. Vakuumrørsollareren anuges at være installeret i en solvarmezentral, hvor den opererer ved en konstant middeltemperatur på 50°C. Sollangererheden er understøttet for to klimaer, hhv. for København og Uummannaq (Grønland).

Årlige gennemsnitshovede for klimaerne er vist i tabel 1. Omgivelsernes refleksionsstofhovede er for en stor del af året højere i Uummannaq end i København på grund af sneen.

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**Figur 4.**

**Figur 5.**

**Figur 6.**

*Installationen ret nr. 48 - 2005*
Følgende parametre er analyseret:

- Solfangerorientering og solfangerradius (solfangergiometri omvendt).
- Radius af indervæske glasrer (radius af yderste glasrer omvendt).
- Afstand mellem glasrørne (solfangerorientering og solfangerradius omvendt).

Hældning

Figur 6 viser solfangerydelser på glasrer som funktion af solfangerradius og solfangereorientering for hhv. Københavnen og Uummannaq.

Figur viser, at ydelsen falder, når solfangeren orienteres væk fra syd.

Det kan også ses, at den optimale solfangereorientering er ca. 45° for Københavnen og ca. 60° for Uummannaq.

Endelig ses det, at ydelsene er nøjagtige i Uummannaq, idet i Københavnen.

Det skyldes isens høje reflektionskoeficient.

Radius

Figur 7 viser solfangerydelser på glasrer ved en solfangermiddeltemperatur på 50°C som funktion af indervæske glasreradius.

Den ydre glasreradius er 0.0235 m. Figur viser, at solfangerydelsen stiger med stigende radius.

Stigningen er ikke linier, fordi også variabelt fra glasrørene stiger med stigende indre glasreradius.

Afstand

Endelig viser figur 8 solfangerydelser pr. glasrer ved forskellige solfangermiddeltemperatur, som funktion af afstanden (fra centrum til centrum) mellem glasrørene.

Solfangerydelsen stiger ved større radstandindstil en uforstådelig for forhold, når den større rabstanden er.

Endnu større radstandsefarder solfangerydelsen igen. Det skyldes, at yderst glasrer, som forbinder glasrørene, bliver endnu større end den øverste yders, der henter på grund af reducerede skygger på glasrørene.

Den optimale radstand afhænger af solfangermiddeltemperatur. Ved en solfangermiddeltemperatur på 70°C er den optimale afstand ca. 0.15 m og for en solfangermiddeltemperatur på 50°C er den optimale afstand ca. 0.3 m.

Generelt gælder, at de højre solfangermiddeltemperatur er de mere afgørende bliver varmetabnet fra manifakturerene.

Rentabilitet kan forbedres

Som nævnt er forskningsprojektet ca. halvveje i fuldført.

Det fortsatte arbejde vil inkludere eksperimentelle arbejde, hvor der bl.a. skal undersøges, hvilket solfangerkoncept (All-glass eller Heat-Pipe) der er mest attraktivt til aktive

forhold.

Det skal i øvrigt nævnes, at vakuumrør- eller -liner ikke kun er interessante for aktive forhold, det er interessant for alle klimaforhold og for de fleste typer af solvarmeanlæg.

Det er bl.a. fordi der med optimalt designede vakuumrør- og -liner kan være mulighed for at forbedre solvarme

anlægs rentabilitet mærkbart.

Ydeforce information om arbejdet udført i forskningsprojektets første år kan findes i [7].

References


Bilag 10: Overheads til foredraget “Evacuated Tubular Collectors”.
Evacuated Tubular Collectors

Louise Jivan Shah

Background

- 3-year research project
- Part of the research program "Sustainable Arctic Building Technology for the 21st century"
- Funded by the VILLUM KANN RASMUSSEN FOUNDATION
- 2/3 through the project
Solar energy at northern latitudes

- Large season variations
- Solar radiation from all directions
- High ground reflections due to snow
- Low ambient temperatures

Available solar radiation at northern latitudes
Solar collectors for northern latitudes

- When solar collectors are developed for northern latitudes, it is an advantage if the collectors:
  
  😊 Can utilize solar radiation from all directions
  
  😊 Have a low heat loss (due to low ambient temperatures)
  
  😊 Utilize ground reflected radiation well (due to large ground reflection coefficient)

Evacuated tubular collectors

😊 Low heat loss coefficient
  - Vacuum insulation

😊 High efficiency
  - Due to low heat loss

😊 Utilize solar radiation from all directions
  - Absorber design
Different principles of evacuated tubular collectors

- Heat pipe evacuated tubular collectors
- All-glass evacuated tubular collectors
- Cheap, mass-produced in China

Two research examples

- Heat pipe evacuated tubular collectors
  - Two different designs in a solar heating plant

- All-glass evacuated tubular collectors
  - Flow and temperature patterns inside the tubes
Heat pipe
evacuated tubular collectors

Heat pipe principle

Manifold heat-exchanger solar collector fluid (propylene glycol/water)

Heat pipe with working fluid (water)
Heat pipe evacuated tubular collectors with flat or curved fins

- Research work:
  - New collector theory is developed
  - Calculation yearly thermal performance is now possible
  - Detailed optimization of collector design is now possible

Detailed modeling of shadows temperature distribution

Depending on the position of the sun and the distance between the tubes, the tubes cast shadow on each other.

Therefore, the solar irradiance varies along the fin and the fin temperature must be calculated in detail.
Collector rows and shadows

![Diagram of collector rows and shadows]

Using the new theory
Collector array in a solar heating plant

- Weather data
  - Uummannaq, Greenland

- Parameter investigations:
  - Tube centre distance
  - Collector row distances

- Operation data
  - Operating temperature: 50°C

- Collector data
  - Tube length: 2 m
  - Tube diameter: 0.1 m
  - Tilt: 60°, Orientation: South
Energy production from a solar collector field

- Collector row distance: 5 m
- Tube centre distance: 0.125 m
- 8000 tubes on a ground array with the size of a football court
- Yearly energy production: 1100 MWh/year
- District heating production in Sisimiut: 25000 MWh/year

Summary of research example

- New theory for heat pipe evacuated tubular collectors developed
- The theory includes in detail heat transfer processes of the absorber fins
- Detailed optimization of collector design now possible
All-glass evacuated tubular collectors

Investigation of flow and temperature patterns inside the tubes

- All-glass collector
- Heat transfer
- Flow structures
All-glass working principle

Glass tubes with collector fluid (propylene glycol/water)

Manifold tube

Computational Fluid Dynamics (CFD)

- With CFD, temperatures and flow patterns are determined
- A grid is made of the geometry and governing equations of fluid flow are solved numerically
CFD investigations

Calculations:
- 5 different inlet flow rates:
  - 0.05 kg/min – 10 kg/min.
- Inlet temperature: 60 °C
- Typical solar radiation

Results:
- Flow structures
- Influence on collector efficiency

Flow inside the All-glass tube

We see two horizontal All-glass tubes connected to the manifold pipe in the centre

All-glass tube  Manifold pipe  All-glass tube
Temperatures inside the All-glass tube

Calculated efficiency based on CFD calculations
Results for different inlet flow rates

Observation:
- The manifold inlet flow rate has only small (<2%) influence on collector efficiency

Explanation:
- Small variations in the flow patterns in the glass tubes
- Self adjusting flow in glass tubes

Summary

- Example of how Computational Fluid Dynamics gives detailed useful results
- Optimal manifold inlet flow rate is around 0.4-1 kg/min.
- Flow structures in the glass tubes were relatively uninfluenced by the manifold inlet flow rate
- This indicates that the collector design is well working for most operating conditions.
Further work

- Measurements
  - New test facility
  - Test of 5 differently designed evacuated tubular collectors
  - Direct performance comparison
  - Final validation of theoretical models

- Optimization work

- Design
  - Based on the findings and on economy considerations, optimum designed evacuated tubular collectors for Arctic latitudes will be recommended

Outlook

- Today, solar heating is one of the largest renewable energy technologies

- Maybe, well designed cheap evacuated tubular collectors, will create a shining future for solar heating in Arctic regions
Thank you for your attention!
Bilag 11: Overheads/Poster til foredraget ”New Trnsys Model of Evacuated Tubular Collectors with Cylindrical Absorbers”.

NEW TRNSYS MODEL OF EVACUATED TUBULAR COLLECTOR WITH CYLINDRICAL ABSORBER

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Introduction

- A new TRNSYS collector model for evacuated tubular collectors with tubular absorbers is developed.
- On each tube the model determines the size and position of the shadows caused by the neighbour tube.
- The model takes into account solar radiation from all directions (also from the “back” of the collector).
- The model is validated with measurements.
- The model makes it possible to make a theoretical design optimization.
I look forward to meet you at the poster where you can see some calculation examples
NEW TRNSYS MODEL OF EVACUATED TUBULAR COLLECTOR WITH CYLINDRICAL ABSORBER

L.J. Shah & S. Furbo, BYG.DTU, Technical University of Denmark, Email: ljs@byg.dtu.dk

Summary
A new TRNSYS collector model for evacuated tubular collectors with tubular absorbers is developed.

The model is based on traditional flat plate collector theory, where the performance equations have been integrated over the whole absorber circumference.

On each tube the model determines the size and position of the shadows caused by the neighbour tube as a function of the solar azimuth and zenith.

The model takes into account solar radiation from all directions (also from the “back” of the collector).

Calculations with the new model of a tubular collector are compared with measured results and a good degree of similarity between the measured and calculated results is found.

The collector model is used in a model of a solar heating plant and a sensitivity analysis of the tube centre distance, collector tilt and orientation with respect the thermal performance per tube is investigated for the two locations Copenhagen (Denmark) and Uummannaq (Greenland).

Prototype panel and model validation

Simulation of solar heating plants.
A model of a solar heating plant is built in TRNSYS. The collector array consists of 100 rows – 50 m long.

The thermal performance per tube as a function of the tube centre distance, collector tilt and orientation (Copenhagen).

The thermal performance per tube as a function of the tube centre distance, collector tilt and orientation (Uummannaq).
Bilag 12: NorthSun 2005 paper: Utilization of solar radiation at high latitudes with evacuated tubular collectors
UTILIZATION OF SOLAR RADIATION AT HIGH LATITUDES WITH EVACUATED TUBULAR COLLECTORS

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Abstract – Theoretical investigations of how two differently designed ground mounted heat pipe evacuated tubular collectors utilize the solar radiation at high latitudes have been carried out. The absorber fins in the evacuated tubes are either flat or curved and the fins have selective coating on both sides so that solar radiation from all directions can be utilized. The yearly thermal performance of heat pipe evacuated tubular collectors has been investigated for three different Nordic climates: Uummannaq (Greenland), Sisimiut (Greenland) and Copenhagen (Denmark). Further, the thermal performance of the evacuated tubular collectors has been compared with the thermal performance of a high efficient flat plate collector. Calculations of the yearly thermal performance show that the heat pipe evacuated tubular collectors are relatively better performing in Uummannaq than in Copenhagen compared to the flat plate collector. Further calculations assuming that no solar radiation is absorbed on the back side of the absorbers in the evacuated tubular collectors have been carried out and the performance improvement by having double-sided absorbers has been studied. Here the results for both the flat strip evacuated tubular collector and the curved strip evacuated tubular collector show that the influence of the double-sided absorber increases with increasing latitudes.

1. INTRODUCTION

This work presents theoretical investigations of how heat pipe evacuated tubular collectors can utilize the solar radiation at high latitudes. Two designs of heat pipe evacuated tubular collectors are investigated theoretically. The absorber fins in the evacuated tubes are either flat or curved and the fins have selective coating on both sides. This means that solar radiation from all directions can be utilized. The tubes are connected to a heat exchanger manifold pipe where condensers for all tubes are placed. An illustration of the evacuated tubes is given in Fig. 1.

![Fig. 1: The investigated evacuated tubular heat pipes.](image)

Two new TreSys (Klein, S.A. et al. (1996)) models for collectors with evacuated tubes with flat and curved fins have been developed. The models take solar radiation from all directions into account. Due to the cylindrical tubes, depending on the position of the sun and the distance between the tubes, the tubes will be able to cast shadow on each other as illustrated in Fig. 2. Thus, the solar irradiance can vary along the fin. Of course, for the curved fin, the irradiance always varies along the fin as the incidence angle varies along the fin. Due to this variation in the solar irradiances along the fins, the traditional fin efficiency cannot be applied and the heat transfer processes in the fin must be solved in detail.

![Fig. 2: The irradiated part of the fin for a given position of the sun.](image)
The yearly thermal performance of heat pipe evacuated tubular collectors placed in a collector field on the ground will be investigated. Therefore, shadows on the ground caused by the collector rows, and view factors including reduction of the view factors to ground and sky due to the collector rows must also be calculated in detail. This theory is included in a third Trnsys model, which works together with the two collector models.

The yearly thermal performance of heat pipe evacuated tubular collectors will be investigated for three different Nordic climates: Uummannaq (GL), Sisimiut (GL) and Copenhagen (DK). Especially, for the most Nordic locations, Uummannaq and Sisimiut, during summertime solar radiation will be present almost 24 hours a day. This means that the evacuated tubular collectors with selective coating on both sides of the absorber might have an extra advantage compared to traditional collectors as they can utilize solar radiation from all directions.

For different collector operation conditions and different evacuated tubular collector geometries, the investigation will elucidate the influence of the double-sided absorbers on the thermal performance. Further, the performance of the evacuated tubular collectors will be compared with the performance of a high efficiency flat plate collector.

2. BACKGROUND THEORY

2.1 Heat pipe evacuated tubular collector models

The theory behind the heat pipe evacuated tubular collector models will not be described here, as it has previously been described in Shah, L.J. & Furbo, S. (2005a) and Shah, L.J. & Furbo, S. (2005b).

2.2 Collector row model

The purpose of the collector row model is to get information of the influence of the distance between collector rows on the thermal performance. The model calculates in detail:

- Shadows on the ground caused by the collector rows
- Shadows on the collectors caused by the collector rows
- Reduction of view factors from collector to ground and sky due to the collector rows
- Reduction of view factors from ground to sky due to the collector rows

In the following, the theory in the model will be described.

**Shadows on the ground**:

The shadows on the ground depend on the length, L, azimuth, θ_0, and tilt, β, of the collector, and the solar zenith angle, θ_s, and solar azimuth, γ_s (see Fig. 3). The aim is to find the position of the solar part on the ground (ground area irradiated with direct solar radiation) and thus to find the view factor to the solar part on the ground from the collector front side (F_{col,groundfront}) and back side(F_{col,groundback}). These view factors are important in order to correctly calculate the ground reflected direct solar radiation.

In vector notation, the position of the sun can be described by:

\[
\textbf{S} = \begin{pmatrix}
\sin \theta_s \cdot \cos (\gamma_s - \gamma_t) \\
\sin \theta_s \cdot \sin (\gamma_s - \gamma_t) \\
\cos \theta_s
\end{pmatrix}
\]

The point P_1 and P_2 (see Fig. 4 and Fig. 5) are given by:

\[
P_1 = \begin{pmatrix}
\cos (\beta) \\
\sin (\gamma - \gamma_t) \\
\sin (\gamma - \gamma_t)
\end{pmatrix}
\]

\[
P_2 = \begin{pmatrix}
\sin (\gamma - \gamma_t) \\
\cos (\beta) \\
\sin (\gamma - \gamma_t)
\end{pmatrix}
\]

Knowing that x_2 is equal to 0 gives a solution for X and thus for x_0:

\[
x_0 = x_0 - \tan (\theta_s) \cos (\gamma_s - \gamma_t) z_0
\]

Fig. 3: Illustration of angles used in the theory (from Duffie J.A. and Beckman W.A. (1980))

If \(\gamma_0 - \gamma_1\) is less than or equal to \(\pi/2\), that is if the sun shines on the front side of the collector, the view factor to the solar part on the ground from the collector front and back side is found by using the one-dimensional "string-method" (Hadvig S. (1980)).
\[
F_{\text{vol. be ground.frt}} = \frac{\text{dist} + z_1 - x - z_0}{2L} \quad (5)
\]
\[
F_{\text{vol. be ground.back}} = \frac{L + x - z_0}{2L} \quad (6)
\]

Here "dist" is the distance between the collector rows, and \( z_1, z_0 \) are given by (see Fig. 4):
\[
\alpha = \sqrt{\alpha_1^2 + L^2 - 2\alpha_1 L \cos(\beta)} \quad (7)
\]
\[
z_1 = \sqrt{\text{dist}^2 + L^2 - 2\text{dist} L \cos(\beta)} \quad (8)
\]
\[
z_0 = \sqrt{\alpha_0^2 + L^2 - 2\alpha_0 L \cos(\pi - \beta)} \quad (9)
\]

Fig. 4: The geometry used in the derivation of the view factor to the solar part on the ground from the collector when \( \gamma_c - \gamma_l \) is less than or equal to \( \pi/2 \).

If \( \gamma_c - \gamma_l \) is greater than \( \pi/2 \), that is if the sun shines on the back side of the collector, the view factor to the solar part on the ground from the collector front and back side is found in the following way (see Fig. 5):
\[
F_{\text{vol. be ground.frt}} = \frac{\text{dist} + z_1 - x - z_0}{2L} \quad (10)
\]
\[
F_{\text{vol. be ground.back}} = \frac{L + x - z_0}{2L} \quad (11)
\]

Here \( z_3, z_4, z_0 \) are given by:
\[
z_3 = \sqrt{\alpha_3^2 + L^2 - 2\alpha_3 L \cos(\beta)} \quad (12)
\]
\[
z_4 = \sqrt{\alpha_4^2 + L^2 - 2\alpha_4 L \cos(\pi - \beta)} \quad (13)
\]
\[
z_0 = \sqrt{\text{dist}^2 + L^2 - 2\text{dist} L \cos(\pi - \beta)} \quad (14)
\]

Fig. 5: The geometry used in the derivation of the view factor to the solar part on the ground from the collector when \( \gamma_c - \gamma_l \) is greater than \( \pi/2 \).

Shadows due to rows on the collectors:

Furthermore, shadows on the collectors due to neighbour rows must be treated. These shadows occur when "dist" is greater than "dist". Determination of these shadows is important in order to correctly calculate the direct solar radiation on the collectors.

If \( \gamma_c - \gamma_l \) is less than or equal to \( \pi/2 \), that is if the sun shines on the front side of the collector, the length of the shaded part of the collector, \( L_{\text{shad}} \), can be described as (see Fig. 6):
\[
L_{\text{shad}} = v \frac{\sin\left(\frac{\pi}{2} - \theta_1\right)}{\sin\left(\frac{\pi}{2} + \theta_1 - \beta_1\right)} \quad (15)
\]

If \( \gamma_c - \gamma_l \) is greater than \( \pi/2 \), that is if the sun shines on the back side of the collector, the length of the shaded part of the collector, \( L_{\text{shad}} \), can be described as (see Fig. 7):
\[
L_{\text{shad}} = v \frac{\sin\left(\frac{\pi}{2} - \theta_1\right)}{\sin\left(\beta_1 + \theta_1 - \frac{\pi}{2}\right)} \quad (16)
\]

Finally, the shade reduction factor due to shadow on the collectors, can be described as:
\[
F_{\text{shad.corr}} = 1 - \frac{L_{\text{shad}}}{L} \quad (17)
\]
Fig. 6: The geometry used in the derivation of the shadows on the collectors when $\gamma - \gamma_s$ is less than or equal to $\pi/2$.

Fig. 7: The geometry used in the derivation of the shadows on the collectors when $\gamma - \gamma_s$ is greater than $\pi/2$.

Reduction due to rows of view factors for diffuse radiation:

The view factors to the ground and to the sky from the collector will be reduced due to the collector rows, compared to if there were no neighbour rows. Therefore, reduction factors for the view factors are important in order to correctly calculate the ground reflected direct solar radiation. These reduction factors can generally be described as:

$$ F_{\text{reduce}} = \frac{F_{\text{collector surrounding with rows}}}{F_{\text{collector surrounding without rows}}} \quad (18) $$

The actual reduction factors can be described as (see Fig. 4 and Fig. 5):

$$ F_{\text{red},i,	ext{-},l,	ext{-},\text{front}} = \frac{0.5(L + \text{dist} - z_i)}{\frac{L}{1 + \cos(\beta)}} = \frac{L + \text{dist} - z_i}{L(1 + \cos(\beta))} \quad (20) $$

$$ F_{\text{red},i,	ext{-},l,	ext{-},\text{back}} = \frac{0.5(L + \text{dist} - z_i)}{\frac{L}{1 + \cos(\beta)}} = \frac{L + \text{dist} - z_i}{L(1 + \cos(\beta))} \quad (21) $$

$$ F_{\text{red},i,	ext{-},l,	ext{-},\text{both}} = \frac{0.5(L + \text{dist} - z_i)}{\frac{L}{1 + \cos(\beta)}} = \frac{L + \text{dist} - z_i}{L(1 + \cos(\beta))} \quad (22) $$

Reduction due to rows of view factor from ground to sky:

Finally, the view factor from ground to sky will also be reduced due to the collector rows, and this will influence the diffuse radiation on horizontal. The view factor from ground to sky is found by:

$$ F_{\text{g}-s} = \frac{z_i + x_s - 2L}{2 \text{dist}} \quad (23) $$

2.3 TmSys model

The theory described above is programmed into a TmSys type, which works together with the heat pipe evacuated tubular collector models. The actual values used by the collector models are given in equations (5) – (6), (17) and (19) – (23).

3. CALCULATIONS

In the following calculation examples it is assumed that the ground mounted heat pipe evacuated tubular collector panels are operating in a solar heating plant at a constant operating temperature in the heat exchanger manifold pipe throughout the year.

3.1 Weather data

The yearly thermal performance of heat pipe evacuated tubular collectors will be investigated for three different Nordic climates: Uummannaq (Greenland), Sisimiut (Greenland) and Copenhagen (Denmark) as summarized in Table 1. For Uummannaq and Sisimiut, the albedo is set higher during a large part of the year due to snow on the ground.
Table 1: Summarized data for Uummannaq (UMA), Sisimiut (SIS) and Copenhagen (CPH).

<table>
<thead>
<tr>
<th>Location</th>
<th>UMA (Knudsen et al. (2005))</th>
<th>SIS (Unpublished)</th>
<th>CPH (Land H. (1995))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Latitude</td>
<td>[°F]</td>
<td>71</td>
<td>67</td>
</tr>
<tr>
<td>Longitude</td>
<td>[°F]</td>
<td>52</td>
<td>34</td>
</tr>
<tr>
<td>$T_t$</td>
<td>[°C]</td>
<td>-1.2</td>
<td>-3.7</td>
</tr>
<tr>
<td>$G_{global}$</td>
<td>[kWh/m²]</td>
<td>888</td>
<td>827</td>
</tr>
<tr>
<td>$G_{diffuse}$</td>
<td>[kWh/m²]</td>
<td>499</td>
<td>352</td>
</tr>
<tr>
<td>$All-sky$</td>
<td>[-]</td>
<td>0.2</td>
<td>0.2</td>
</tr>
</tbody>
</table>

Table 2: Data describing the heat pipe collector in the model.

<table>
<thead>
<tr>
<th>Component</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube length, L</td>
<td>m</td>
</tr>
<tr>
<td>Glass tube outer radius</td>
<td>m</td>
</tr>
<tr>
<td>Absorber radius for curved fin</td>
<td>m</td>
</tr>
<tr>
<td>Fin width for curved fin (corresponding to a curved fin angle of 16°)</td>
<td>m</td>
</tr>
<tr>
<td>Fin width (for flat fin)</td>
<td>m</td>
</tr>
<tr>
<td>Flat fin absorber area</td>
<td>m²</td>
</tr>
<tr>
<td>Fin thickness</td>
<td>m</td>
</tr>
<tr>
<td>Fin conductivity [W/mK]</td>
<td>W/m²K</td>
</tr>
<tr>
<td>Effective transmittance absorptance product</td>
<td>[-]</td>
</tr>
<tr>
<td>Incident angle modifier constant</td>
<td>[-]</td>
</tr>
<tr>
<td>Manifold heat loss coefficient [W/Km]</td>
<td>1.34</td>
</tr>
<tr>
<td>Manifold heat exchange rate per connection [W/K]</td>
<td>10</td>
</tr>
<tr>
<td>Operating temperature in manifold [°C]</td>
<td>50</td>
</tr>
<tr>
<td>Collector heat capacity [kW/Ktube]</td>
<td>1.9</td>
</tr>
<tr>
<td>Distance between collector rows</td>
<td>m</td>
</tr>
</tbody>
</table>

Especially the direct radiation on horizontal is higher in Uummannaq during April to July. From Fig. 10 it can be seen that the monthly average ambient temperatures typically are 10 K - 15 K lower in Uummannaq and Sisimiut compared to Copenhagen.

3.2 Collector input

As mentioned earlier it is assumed that the ground mounted evacuated tubular collector panels are operating in a solar heating plant at a constant operating temperature in the heat exchanger manifold pipe throughout the year. The model data of the collector panels are given in Table 2.

The performance of the heat pipe evacuated tubular collectors is compared with the performance of a high efficiency flat plate collector. The characteristic of this collector is given in Table 3.
<table>
<thead>
<tr>
<th>η  [-]</th>
<th>a₀ [W/m²K]</th>
<th>a₁ [W/m²K²]</th>
<th>K₀ [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.82</td>
<td>2.44</td>
<td>0.005</td>
<td>1-tan(0.2°)³⁰⁶</td>
</tr>
</tbody>
</table>

Table 3: Characteristics for the flat plate collector.

5.5 Results

The first calculations are made in order to find the optimal collector tilt for the three locations. For the flat strip heat pipe evacuated tubular collector, Fig. 11 shows the thermal performance per m² transparent area as a function of the collector tilt. For the evacuated tubular collectors, the transparent area is defined as the cross sectional area of the glass tubes. The calculations are carried out with a collector row distance of 10 m and for an average temperature of 50°C in the heat exchanger manifold pipe. It can be seen that the optimal tilt is about:

- 45° in Copenhagen
- 50° in Sisimiut
- 60° in Uummannaq

The same optimal tilts are found also to be valid for the curved strip heat pipe evacuated tubular collector and for the flat plate collector.

![Fig. 11: Thermal performance for the heat pipe evacuated tubular collector with the flat strip as a function of the collector tilt.](image)

The tendencies shown in Fig. 12 are the same for the analysis with the curved strip heat pipe evacuated tubular collector and for the flat plate collector.

![Fig. 12: Thermal performance for the heat pipe evacuated tubular collector with the flat strip as a function of the collector row distance.](image)

For Copenhagen, Sisimiut and Uummannaq, Fig. 13, Fig. 14 and Fig. 15 show the thermal performance as a function of the operating temperature. For the flat plate collector, the operating temperature is the collector fluid mean temperature. For the heat pipe evacuated tubular collectors, the operating temperature is the collector fluid mean temperature in the heat exchanger manifold pipe.

First of all, the figures show that the thermal performance decreases with increasing temperature level, obviously due to the increasing heat loss. Comparing the heat pipe evacuated tubular collectors with the flat and curved strips respectively, it appears that the flat strip evacuated tubular collector performs better than the curved strip evacuated tubular collector for lower temperatures. The reason is that the heat loss is larger for the curved strip collector compared to the flat strip collector as the heat loss coefficient (the same for the two collectors) is based on the absorber front side area and the curved absorber is larger than the flat fin absorber.

Comparisons of the “collector ranking” for the three locations show that:

- Copenhagen (Fig. 13) the flat plate collector performs best for operating temperatures up to 80°C. The flat strip evacuated tubular collector performs second best for temperatures up to 80°C and the curved strip evacuated tubular collector performs third best.
- Sisimiut (Fig. 14) the flat plate collector performs best for operating temperatures up to 60°C and the flat strip evacuated tubular collector is best performing for operating temperatures over 60°C. The difference in thermal performance between the flat plate collector and the curved strip evacuated tubular collector is smaller under Sisimiut weather data compared to Copenhagen weather data.
- Uummannaq (Fig. 15) the flat strip evacuated tubular collector is best performing for almost all operating temperatures and the performances of the
flat plate collector and the curved strip evacuated tubular collector are more or less identical.

- The flat plate collector (Fig. 18) has a lower thermal performance in Uummannaq than in Copenhagen.
- All three collectors have the lowest thermal performance in Sisimiut, which is due to the less solar radiation in Sisimiut compared to the two other locations.

Fig. 13: The thermal performance per m² transparent area in Copenhagen as a function of the operating temperature.

Fig. 14: The thermal performance per m² transparent area in Sisimiut as a function of the operating temperature.

Fig. 15: The thermal performance per m² transparent area in Uummannaq as a function of the operating temperature.

In Fig. 16, Fig. 17 and Fig. 18 the thermal performances are compared in a different way. The performance of each collector type is shown as a function of the operating temperature and the location. Now it can be seen that:

- The flat strip (Fig. 16) and the curved strip (Fig. 17) evacuated tubular collectors have more or less the same thermal performances in Copenhagen and in Uummannaq.

Fig. 16: Thermal performance per m² transparent area for the flat strip evacuated tubular collector as a function of the operating temperature.

Fig. 17: Thermal performance per m² transparent area for the curved strip evacuated tubular collector as a function of the operating temperature.

Fig. 18: Thermal performance per m² transparent area for the flat plate collector as a function of the operating temperature.

The results show that the heat pipe evacuated tubular collectors are relatively better performing in Uummannaq than in Copenhagen compared to the flat plate collector. Does this mean that the evacuated tubular collectors with
selective coating on both sides of the absorbers do have an extra advantage compared to traditional collectors due to the possibility of utilization of solar radiation from all directions? This will be investigated in the following sections.

Utilization of solar radiation in Uummannaq.

As an example, Fig. 19 show the power from the three collectors in Uummannaq during the 1st of July. The weather conditions on this day are shown in Fig. 20. Here it can be seen that the solar zenith is below 50° during all 24 hours and the global radiation never goes to zero. The ambient temperature is in the range of 7°C to 13°C.

In Fig. 19 it can be seen that the period of operation for the evacuated tubular collectors is longer than the period of operation for the flat plate collector. The flat plate collector is in operation during 13 hours of the day, whereas the curved strip and the flat strip evacuated tubular collectors are in operation during respectively 18 hours and 19 hours of the day. This shows that the double-sided collectors do have an influence.

The figure also shows that the thermal performance of the flat plate collector is superior to the evacuated tubular collectors during mid-day. Further, it can be seen that the curved strip evacuated tubular collector, which has the lowest performance during mid-day, has the highest performance in the morning and in the afternoon. The reason for this is mainly related to the incidence angle.

For the investigated day, the accumulated thermal performances are 4.1 kWh/m² transparent area for the curved strip evacuated tubular collector; 4.1 kWh/m² transparent area for the flat strip evacuated tubular collector and 4.2 kWh/m² transparent area for the flat flat evacuated tubular collector.

Influence of the double-sided absorbers:

To further investigate the influence of the double-sided absorbers, calculations assuming that no solar radiation is absorbed on the back side of the absorbers in the evacuated tubular collectors have been carried out. Fig. 21, Fig. 22 and Fig. 23 show the thermal performance as a function of the operating temperature for the three locations. The figures can be compared to Fig. 13, Fig. 14 and Fig. 15. The only difference in the calculations is that the evacuated tubular collectors with both flat and curved strips do not absorb solar radiation on the strip back side.

Comparisons the "collector ranking" for the three locations now show that the flat plate collector is best performing, the flat strip evacuated tubular collector is second best performing and the curved strip evacuated tubular collector has the lowest thermal performance. Thus, it is clearly illustrated that the solar radiation on the back side of the absorbers has a significant influence on the thermal performance.

![Fig. 19: The power from the three collectors in Uummannaq during the 1st of July.](image1)

![Fig. 20: Weather conditions on the 1st of July in Uummannaq.](image2)

![Fig. 21: The thermal performance per m² transparent area in Copenhagen as a function of the operating temperature. Here the evacuated tubular collectors with both flat and curved strips do NOT absorb solar radiation on the strip back side.](image3)

![Fig. 22: The thermal performance per m² transparent area in Sisimiut as a function of the operating temperature. Here the evacuated tubular collectors with both flat and curved strips do NOT absorb solar radiation on the strip back side.](image4)
The quantitative influence of the double-sided absorbers is shown in Fig. 24 and Fig. 25 where the relative thermal performances for respectively the flat and the curved strip evacuated tubular collector are shown as a function of the operating temperature. Here the relative thermal performance, \( \psi \), is defined as the thermal performance of the double-sided evacuated tubular collector divided by the thermal performance of the single-sided evacuated tubular collector:

\[
\psi_{\text{double-sided absorber}} = \frac{\text{Performance}_{\text{double-sided absorber}}}{\text{Performance}_{\text{single-sided collector}}} \tag{24}
\]

For both the flat strip evacuated tubular collector (Fig. 24) and the curved strip evacuated tubular collector (Fig. 25), it can be seen that the influence of the double-sided absorber increases with increasing latitudes. For example, in Uummannaq for a collector operating temperature of 60°C the extra thermal performance is about 22% for the flat strip evacuated tubular collector and about 19% for the curved strip evacuated tubular collector. In Copenhagen, at the same temperature level, the extra thermal performances are about 15% for the flat strip evacuated tubular collector and about 14% for the curved strip evacuated tubular collector.

4. CONCLUSIONS

Theoretical investigations of how two differently designed ground mounted heat pipe evacuated tubular collectors utilize the solar radiation at high latitudes have been carried out. The absorber fins in the evacuated tubes are either flat or curved and the fins have selective coating on both sides so that solar radiation from all directions can be utilized.

The yearly thermal performance of heat pipe evacuated tubular collectors has been investigated for three different Nordic climates: Uummannaq (Greenland), Sisimiut (Greenland) and Copenhagen (Denmark). Further, the performance of the evacuated tubular collectors has been compared with the performance of a high efficient flat plate collector.

Calculations of the yearly thermal performance show that the heat pipe evacuated tubular collectors are relatively better performing in Uummannaq than in Copenhagen compared to the flat plate collector.

Further, a detailed analysis of the collector operation on a summer day in Uummannaq show that the period of operation for the evacuated tubular collectors is longer than the period of operation for the flat plate collector. Assuming a operating temperature of 50°C, the flat plate collector is in operation during 13 hours of the day, whereas the curved strip and the flat strip evacuated tubular collectors are in operation during 18 hours and 19 hours of the day, respectively.

To further investigate the influence of the double-sided absorbers, calculations assuming that no solar radiation is absorbed on the back side of the absorbers in the evacuated tubular collectors have been carried out and the performance improvement by having double-sided absorbers has been studied.

Here the results for both the flat strip evacuated tubular collector and the curved strip evacuated tubular collector show that the influence of the double-sided absorber increases with increasing latitudes. Thus, in Uummannaq for a collector operating temperature of 60°C the extra thermal performance is about 22% for the flat strip evacuated tubular collector and about 19% for the curved strip evacuated tubular collector. In Copenhagen at the
same temperature the extra thermal performances are
about 15% for the flat strip evacuated tubular collector
and about 14% for the curved strip evacuated tubular
collector.

Further work:
Parallel to the theoretical work, the investigated heat
pipe evacuated tubular collectors will be tested side-by-
side in an outdoor test facility. The measured
performances will be used to verify the theoretical
models.

ACKNOWLEDGEMENTS
This study is financed by the VILUM KANN
RASMUSSEN FOUNDATION.

NOMENCLATURE

\[ \text{dist} \]
Distance between collector rows, m

\[ F_{\text{col,ground,back}} \]
View factor from the collector back side
to the part of the ground with direct
solar radiation,

\[ F_{\text{col,ground,front}} \]
View factor from the collector front side
to the part of the ground with direct
solar radiation,

\[ F_{\text{ground,sky}} \]
View factor from the ground to the sky,

\[ F_{\text{red,c,s,back}} \]
Reduction factor from the collector back
side to the part of the ground with
diffuse solar radiation,

\[ F_{\text{red,c,s,front}} \]
Reduction factor from the collector
front side to the part of the ground with
diffuse solar radiation,

\[ F_{\text{red,c,s,back}} \]
Reduction factor from the collector back
side to the sky,

\[ F_{\text{red,c,s,front}} \]
Reduction factor from the collector
front side to the sky,

\[ F_{\text{red,shade}} \]
Shade reduction factor due to shades on
the collectors,

\[ F_{\text{red,c}} \]
Reduction factor,

\[ G_{\text{diffuse}} \]
Yearly diffuse radiation on horizontal,
kWh/m²

\[ G_{\text{global}} \]
Yearly global radiation, kWh/m²

\[ L \]
Tube length, m

\[ L_{\text{shade}} \]
Length of shade on collector,

\[ P_b \]
Point on top of collector row,

\[ P_r \]
Point on ground,

\[ r_c \]
Glass tube radius, m

\[ r_p \]
Absorber radius for curved fin, m

\[ S \]
Solar vector,

\[ T_a \]
Yearly average ambient temperature, °C

\[ v \]
Help variable, m

\[ w_c \]
Fin width for curved fin, m

\[ w_f \]
Fin width for flat fin, m

\[ X \]
Help variable, m

\[ x_0 \]
x coordinate of point P_b, m

\[ y_0 \]
y coordinate of point P_b, m

\[ z_0 \]
z coordinate of point P_b, m

\[ z_\text{col} \]
Help variable, m

\[ z_\text{sky} \]
Help variable, m

\[ z_\text{shade} \]
Help variable, m

\[ z_\text{global} \]
Help variable, m

\[ \beta \]
Collector row tilt, °

\[ \gamma_r \]
Collector azimuth, °

\[ \gamma_s \]
Solar azimuth, °

\[ \theta_s \]
Solar zenith, °

\[ \nu \]
Relative thermal performance, -

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Insulation Laboratory. Technical University of Denmark.
THEORETICAL INVESTIGATIONS OF DIFFERENTLY DESIGNED HEAT PIPE EVACUATED TUBULAR COLLECTORS

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ABSTRACT
Two designs of ground mounted heat pipe evacuated tubular collectors operating in a solar heating plant are investigated theoretically. The absorber fins inside the evacuated tubes are either flat or curved and the surfaces of the fins have selective coating on both sides. Two new TrmSys models for evacuated tubular collectors are developed. The models calculate in detail the heat transfer processes of the absorber fins. It is illustrated how the model can be used for geometrical parameter studies. For example, it is investigated how fin geometry, collector tilt, operating temperature, tube distances and distances between collector rows influence the yearly thermal collector performance.

1 INTRODUCTION

Evacuated tubular collectors have an increasing share of the collector market in the world. Up to 2001 more than 100 million m² collectors were installed worldwide. Of this, about 28% were unglazed collectors, 40% were traditional flat plate collectors and about 22% were evacuated tubular collectors. On the world largest solar thermal market, China, evacuated tubular collectors have today a market share greater than 80% [1].

Due to the market development, new collector designs as well as development of theoretical models for these designs becomes more and more important. In this paper, two designs of heat pipe evacuated tubular collectors are investigated theoretically. The absorber fins in the evacuated tubes are either flat or curved and the fins have selective coating on both sides. This means that solar radiation from all directions can be utilized.

An illustration of the evacuated tubes is given in Fig. 1. The tubes are connected to a heat exchanger manifold pipe where condensers for all tubes are placed. Two new TrmSys [2] models for collectors with evacuated tubes with flat and curved fins are developed. The models take solar radiation from all directions into account. Further, due to the cylindrical tubes, depending on the position of the sun and the distance between the tubes, the tubes will be able to cast shadow on each other as illustrated in Fig. 2 and the solar irradiance can vary along the fin. For the curved fin model, the irradiance always varies along the fin as the incidence angle varies along the fin. Due to the variation in the solar irradiances along the fins, the traditional fin efficiency cannot be applied. Therefore, the heat transfer processes in the fin are solved in detail.

Fig. 1: The investigated evacuated tubular heat pipes.

Fig. 2: The irradiated part of the fin for a given position of the sun.

With the models, a parameter sensitivity analysis is carried out for the evacuated tubular collectors installed in a solar heating plant. This analysis illuminates how the:
- different fin geometries
- collector tilt
- operating temperature
- tube distances
- distances between collector rows influence the yearly thermal performance of the collector field.
2 THEORY AND DEFINITIONS

In this section, the theory of the two collector models, for flat and curved fins respectively, will be summarized. The models are developed for the TmSys simulation program. The models include both the evacuated tubular collectors and the heat exchanger manifold pipe, as illustrated in Fig. 3.

Fig. 3: The collector models include both the evacuated tubular collectors and the heat exchanger manifold pipe.

The power, $P_n$, from this system can be written as:

$$P_n = m \cdot c_p \cdot (T_{\text{manifold, inlet}} - T_{\text{manifold, outlet}})$$  \hspace{1cm} (1)

$$P_c = P_{BTC} - P_{loss} - dQ_{\text{manifold}}$$  \hspace{1cm} (2)

The power from the heat pipes, $P_{BTC}$, the heat loss from the manifold tube, $P_{loss}$, and the energy change in the manifold tube, $dQ_{\text{manifold}}$, can be written as:

$$P_{BTC} = U_{A_{\text{manifold}}} \cdot (T_{\text{manifold}} - T_{\text{manifold, inlet}})$$  \hspace{1cm} (3)

$$P_{loss} = U_{loss} \cdot (T_{\text{manifold}} - T_{\text{loss}})$$  \hspace{1cm} (4)

with

$$T_{\text{manifold}} = \frac{T_{\text{manifold, inlet}} + T_{\text{manifold, outlet}}}{2}$$  \hspace{1cm} (5)

The power from the heat pipes, $P_{BTC}$, is larger than zero if the temperature of the heat pipe working fluid, $T_{\text{manifold}}$, is larger than the lowest evaporation temperature and larger than the mean temperature in the manifold pipe, $T_{\text{manifold, inlet}}$. In other cases, $P_{BTC}$ is zero.

In order to determine the temperature of the heat pipe working fluid, the fin is discretized into a number of elements as illustrated in Fig. 4. The energy balances for the elements of the fin are:

$$\frac{\lambda \cdot \delta}{dx} \cdot (T_i - T_{in}) + S_i \cdot L \cdot dx - U_i \cdot (T_i - T_{in}) \cdot L \cdot dx = \frac{m_i \cdot c_{p,i}}{dt} \cdot (T_{i+1}^{\text{new}} - T_{i}^{\text{old}})$$  \hspace{1cm} (6)

$$\begin{align*}
1 < i < n: \quad \\
\frac{\lambda \cdot \delta}{dx} \cdot (T_i - T_{in}) - \frac{\lambda \cdot \delta}{dx} \cdot (T_{i-1} - T_{in}) + S_i \cdot L \cdot dx - U_i \cdot (T_i - T_{in}) \cdot L \cdot dx = \frac{m_i \cdot c_{p,i}}{dt} \cdot (T_{i+1}^{\text{new}} - T_{i}^{\text{old}})
\end{align*}$$  \hspace{1cm} (7)

Fig. 4: The fins are discretized into a number of elements.

The total solar radiation absorbed on the fin, $S_0$, is dependent on the position on the fin, due to varying incident angles (for curved fins) and due to possible shadows on the fin (for both flat and curved fins). The principles of determining the size and position on the fin of the shadows are described in [3].

The heat exchange capacity rate, $U_{A_{\text{manifold}}}$, has a constant value larger than zero if $T_{\text{manifold}}$ is larger than the lowest evaporation temperature and if $T_{\text{loss}}$ is larger than the mean temperature in the manifold pipe. In other cases, $U_{A_{\text{manifold}}}$ is zero.

2.1 Solar radiation and view factors

The total solar radiation absorbed on the fin, $S_0$, can be written as:

$$S_0 = S_{\text{sky, front}} + S_{\text{sky, back}}$$

$$+ S_{\text{del, sky, front}} + S_{\text{del, sky, back}}$$

$$+ S_{\text{del, ground, front}} + S_{\text{del, ground, back}}$$

$$+ S_{\text{del, grass, front}} + S_{\text{del, grass, back}}$$

where

$$S_{\text{sky, front}} = (\alpha_0) \cdot c_0 \cdot K_r \cdot R_{\text{sky, front}} \cdot R_{\text{max}}$$

$$S_{\text{sky, back}} = (\alpha_0) \cdot c_0 \cdot K_r \cdot R_{\text{sky, back}} \cdot R_{\text{max}}$$

$$S_{\text{del, sky, front}} = (\alpha_0) \cdot c_0 \cdot K_{R, \text{del, sky, front}} \cdot F_{\text{sky, front}}$$

$$S_{\text{del, sky, back}} = (\alpha_0) \cdot c_0 \cdot K_{R, \text{del, sky, back}} \cdot F_{\text{sky, back}}$$

$$S_{\text{del, ground, front}} = (\alpha_0) \cdot c_0 \cdot K_{R, \text{del, ground, front}} \cdot F_{\text{ground, front}}$$

$$S_{\text{del, ground, back}} = (\alpha_0) \cdot c_0 \cdot K_{R, \text{del, ground, back}} \cdot F_{\text{ground, back}}$$

The incident angle modifier $K_r$ is calculated by:

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Here, the incident angle, \( \theta \), is defined as the incident angle on the absorber. The incident angle modifier for diffuse radiation, \( K_d \), and ground reflected radiation, \( K_g \), are evaluated by equation (14) using \( \theta = \pi / 2 \) [4].

An illustration of the view factors is given in Fig. 5 and a more detailed description of the view factors including reduction of the view factors to ground and sky due to the collector rows is given in [5].

\[
K_c = 1 - \sin^2(\theta) \quad (14)
\]

2.2 Numerical issues

In order to evaluate the performance of the evacuated tubular collectors on an annual basis, the above theory is implemented into three TMsys types; two TMsys types for the two collector designs and an additional TMsys type for the view factor calculations. Concerning the two collector types, for each simulation time step the equations (1)–(8) are solved through an iteration loop. The iteration is stopped when the difference in temperature difference from iteration to iteration is less than 0.00001 K and when the change from iteration to iteration in power from the manifold is less than 0.01 W. If the criteria are not reached the iteration loop stops after 100,000 iterations, and a warning is written to an output file. This seldom happens and the influence on the final result is insignificant. The fins are discretized into nine elements (18 elements in total), and a typical annual simulation of a collector array takes approximately 3 min. on a 2.8 GHz PC when a timestep of 0.05 h is used.

3 Parameter variations

In this section, it is illustrated how the model can be used for geometrical parameter studies. In the following examples it is assumed that the ground mounted evacuated tubular collector panels are operating in a solar heating plant at a constant operating temperature of 50°C in the heat exchanger manifold pipe throughout the year. The model data of the collector panel is given in Table 1. The collector performance is investigated for Nuussuaq, Greenland, as summarized in Table 2. The albedo is set higher during a large part of the year due to snow on the ground.

<table>
<thead>
<tr>
<th>Table 1: Data describing the collector in the model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube length, L</td>
</tr>
<tr>
<td>Glass tube radius, ( r_g )</td>
</tr>
<tr>
<td>Absorber radius for curved fin, ( r_a )</td>
</tr>
<tr>
<td>Fin width for curved fin, ( w_c ) (corresponding to a curved fin angle of 164°)</td>
</tr>
<tr>
<td>Fin width for flat fin, ( w_f )</td>
</tr>
<tr>
<td>Flat fin absorber area</td>
</tr>
<tr>
<td>Curved fin absorber area</td>
</tr>
<tr>
<td>Fin thickness</td>
</tr>
<tr>
<td>Fin conductivity, ( k_a )</td>
</tr>
<tr>
<td>Tube centre distance, C</td>
</tr>
</tbody>
</table>

| Tube heat loss coefficient, \( k_{tube} \) based on absorber front side area | [W/mK] | 2.43 |
| Effective transmittance absorber coating, \( \tau_{eff} \) | [-] | 0.84 |
| Incident angle modifier constant, a | [-] | 3.8 |
| Manifold heat loss coefficient, \( k_{manifold} \) | [W/mK] | 0.134 |
| Manifold heat exchange rate per connection, \( U_{A,manifold} \) | [W/K] | 19 |
| Operating temperature in manifold | [°C] | 50 |
| Collector heat capacity, \( C_{collector} \) | [kJ/K-tube] | 1.9 |
| Distance between collector rows | [m] | 10 |

<table>
<thead>
<tr>
<th>Table 2: Summarized data for Nuussuaq, Greenland</th>
</tr>
</thead>
<tbody>
<tr>
<td>Location</td>
</tr>
<tr>
<td>Latitude</td>
</tr>
<tr>
<td>Longitude</td>
</tr>
<tr>
<td>Temperature</td>
</tr>
<tr>
<td>( U_{global} )</td>
</tr>
<tr>
<td>( C_{heater} )</td>
</tr>
</tbody>
</table>

3.1 View factors

To get an understanding of the geometry included in the models, illustrations of some of the view factors for different conditions is given below. The view factors are calculated for a collector tilt of 60° and for infinite distance between the collector rows.

For different tube centre distances, Fig. 6 and Fig. 7 shows the view factor from one tube to the two neighbour tubes, \( F_{tube,tube} \) and the view factors to the sky.
and ground from the collector’s front and back side respectively. For the collector front side, it can be seen that the flat fin collector has a larger view factor to the sky, a smaller view factor to the ground and a smaller view factor between the tubes compared to the curved fin collector. This is because parts of the curved fin – due to the curve – face the neighbour tubes and the ground more than the flat fin. Further, as expected, the view factors between the tubes decreases with increasing tube centre distances. For the collector back side, it can be seen that the curves for the sky and ground radiation are parallel. The reason for this is that, when calculating the view factors, the back side of the curved fin can be treated as a flat fin placed in the opening of the curved fin. Thus the actual differences in the values are directly related to the area of the opening compared to the area of curved fin absorber. The opening area equals 69% of the curved fin absorber area. Fig. 8 shows the yearly diffuse sky and ground reflected radiation absorbed by the fins. For the collector back side, it can be seen that, there is almost no differences between the curved and the flat fin. For the collector front side it is clear that the curved fin absorbs more diffuse radiation than the flat fin even though the view factor between fin and sky is largest for the flat fin. The reason is that the area of the curved fin is larger than the area of the flat fin. Finally, Fig. 9 shows the direct radiation absorbed by the front side of the fins. Even though the view factor between fin and sky is largest for the flat fin, the curved fin absorbs more direct radiation per tube than the flat fin absorbs. This is still due to the larger curved fin absorber area.

3.2 Temperature profiles on the fin

Before going into thermal performance analyses, some examples of the temperature distribution on the fins are given. For a summer day (31/7), Fig. 10 and Fig. 11 show the temperature profile at solar time 9 AM, 12 PM and 3 PM for the curved and the flat fin respectively. At these three times there are no shadows from the neighbour tubes. The temperature distribution on the curved fin is symmetrical around the heat pipe at 12 PM whereas the east side of the fin is warmer during the morning and the west side of the fin is warmer during the afternoon. This clearly indicates the influence of the distribution of the solar radiation on the fin and it explains why the traditional fin efficiency, F, cannot be applied when analyzing this type of collector in detail. As expected for the flat fin, there is symmetry in the temperature distribution at all three times. The differences in the temperature level for the three times are due to the weather conditions.

Fig. 6: View factors for the collector front side.

Fig. 7: View factors for the collector back side.

Fig. 8: Absorbed diffuse sky and ground reflected radiation.

Fig. 9: Absorbed direct radiation.

Fig. 10: The temperature distribution on the curved fin in the morning, at noon and in the afternoon of a summer day.
3.3 Collector tilt

Fig. 12 shows the thermal performance per tube as a function of the collector tilt. The figure shows that the optimum tilt is about 60° for both the flat fin collector and the curved fin collector. The flat fin collector performs slightly better than the curved fin collector even though the curved fin absorber area is larger than the flat fin absorber area. The reason is that the heat loss is larger for the curved fin collector compared to the flat fin collector as the heat loss coefficient (the same for the two collectors) is based on the absorber front side area and the curved absorber is larger than the flat fin absorber.

Fig. 12: The thermal performance per tube as a function of the collector tilt.

3.4 Operating temperature

Fig. 13 shows the thermal performance as a function of the operating temperature in the manifold tube. As expected the thermal performance decreases with increasing temperature level due to the increasing heat loss.

Fig. 13: The thermal performance per tube as a function of the operating temperature in the manifold tube.

Further, it can be seen that the curves cross at a temperature of approximately 30°C. Again, the reason is that the curved fin collector has a higher heat loss compared to the flat fin collector. So, for lower temperatures the extra absorber area is an advantage for the curved fin collector and for higher temperatures the extra absorber area is a disadvantage for the curved fin collector.

3.5 Tube centre distances

Fig. 14 shows the thermal performance per tube as a function of the tube centre distance. The thermal performance increases for increasing tube centre distances up to about 0.3 m, due to reduced shaded areas, reduced view factors between the tubes and thus increased view factors to sky and ground. For even larger distances the utilized energy decreases again, due to the increasing heat loss from the manifold pipes. Further, the curved fin collector is more sensitive to the tube centre distance compared to the flat fin collector. The reason is directly related to the differences in the view factors for the two collectors as the curved fin collector, due to the design, “sees” more of the neighbour tubes (see Fig. 6) for smaller tube centre distances.

Fig. 14: The thermal performance per tube as a function of the tube centre distance.

3.6 Collector rows distances

Finally, Fig. 15 shows the thermal performance per tube as a function of the distance between the collector rows. It can be seen that thermal performance increases rapidly when the row distance increases from 1 m to 10 m. Further increase in the row distance has less impact on the thermal performance. The reason for the increase in thermal performance is that with increasing distances the shadows from neighbour rows on the collector (see Fig. 5 bottom) decreases and the view factor from the collector to ground and sky increases. It must be noticed that the heat loss in the pipes connecting the collector rows is not included in this analysis.

Fig. 15: The thermal performance per tube as a function of the distance between the collector rows.

4 Conclusion and Outlook

Two heat pipe evacuated tubular collectors operating in a solar heating plant are investigated theoretically. The
absorber fins in the evacuated tubes are either flat or curved and the fins have selective coating on both sides. Two new TmSy models for collectors with the tubes with the flat and curved fins are developed. The models are able to take solar radiation from all directions into account. On each tube the model determines the size and position of the shadows caused by the neighbouring tube as a function of the solar azimuth and zenith, and the temperature profiles on the fins are calculated in detail. Further, based on the tube geometry, the collector tilt and the distances between collector rows in the collector array of a solar heating plant, view factors between the tubes and from the collector to ground and sky are calculated in detail.

It is illustrated how the model can be used for geometrical parameter studies, where the evacuated tubular collectors are operating in a solar heating plant. For example, it is investigated how the fin geometry, collector tilt, operating temperature, tube distances and distances between collector rows influences the yearly thermal collector performance.

Further work:
Parallel to the theoretical work, the investigated heat pipe evacuated tubular collectors will be tested side-by-side in an outdoor test facility. The measured performances will be used to verify the TmSy models.

5 NOMENCLATURE

- Incident angle modifier constant, \( \alpha \)
- Tube centre distance, \( C \)
- Collector heat capacity, \( C_{\text{heat}} \)
- Average heat capacity of fin material, \( C_{\text{fin}} \)
- Heat capacity of fin material, \( C_{\text{fin}} \)
- Energy change in the manifold pipe, \( \delta E \)
- Time step, \( t \)
- Width of discretization element, \( m \)
- View factor from the collector back side to the part of the ground with diffuse solar radiation, \( F_{\text{col,ground}} \)
- View factor from the collector front side to the part of the ground with diffuse solar radiation, \( F_{\text{col,ground}} \)
- View factor from the collector back side to the part of the ground with direct solar radiation, \( F_{\text{col,dir}} \)
- View factor from the collector front side to the part of the ground with direct solar radiation, \( F_{\text{col,dir}} \)
- Beam radiation on horizontal, \( W/m^2 \)
- Diffuse radiation on horizontal, \( W/m^2 \)
- Yearly diffuse radiation on horizontal, \( W/m^2 \)
- Yearly global radiation, \( W/m^2 \)
- Incident angle modifier for direct radiation, \( \alpha_{\text{dir}} \)
- Incident angle modifier for diffuse radiation, \( \alpha_{\text{dir}} \)
- Mass of discretized fin element, \( m \)
- Mass average of fin material and working fluid in heat pipe, \( kg \)
- Power from heat pipes, \( W \)
- Heat loss from the manifold pipe, \( W \)
- Power from collector, \( W \)
- Geometric factor for back side beam radiation, \( \Omega_{\text{back}} \)
- Geometric factor for front side beam radiation, \( \Omega_{\text{front}} \)
- Glass tube radius, \( m \)
- Absorber radius for curved fin, \( m \)

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7 REFERENCES

Bilag 14: Solar World Congress 2005 paper: Numerical investigations of an all glass evacuated tubular collector.
NUMERICAL INVESTIGATIONS OF AN ALL GLASS EVACUATED TUBULAR COLLECTOR

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ABSTRACT

Heat transfer and flow structures inside all glass evacuated tubular collectors for different operating conditions are investigated by means of Computational Fluid Dynamics (CFD). The investigations are based on a collector design with horizontal tubes connected to a vertical manifold channel.

Three different tube lengths varying from 0.59 m to 1.47 m have been modelled with five different inlet mass flow rates varying from 0.05 kg/min to 10 kg/min with a constant inlet temperature of 333 K. Under these operating conditions, the results showed that:

- the collector with the shortest tube length achieved the highest efficiency
- the optimal inlet flow rate was around 0.4–1 kg/min
- the flow structures in the glass tubes were relatively uninfluenced by the inlet flow rate

Generally, the results showed only small variations in the efficiencies. This indicates that the collector design is well working for most operating conditions.

1 INTRODUCTION

All glass evacuated tubular collectors are widely used on the world market. All glass evacuated tubular collectors are based on double glass tubes where the outside of the inner glass wall is treated with an absorbing selective coating and the evacuated space is between the tubes as illustrated in Fig. 1.

Fig. 1: Design of an all glass evacuated tube.

A collector design based on horizontal tubes connected to a manifold pipe is especially popular due to its low cost. An illustration of the collector design is shown in Fig. 2.

Fig. 2: Illustration of an all glass evacuated tubular collector with horizontal tubes.

The collector fluid enters the bottom of the square manifold channel and leaves at the top of the manifold channel. The intended flow inside the glass tubes is indicated with the arrows. The flow is primarily naturally driven, as the walls of the tubes are hot due to
the solar radiation. However it is unclear, how the operating conditions and the collector geometry influence the flow structures in the tubes and thus the collector performance.

The objective of this work is to investigate the heat transfer and the flow structures inside the tubes for different flow rates and collector geometries by means of Computational Fluid Dynamics (CFD).

2 NUMERICAL INVESTIGATIONS

To solve the flow and energy equations in the glass tubes, a simulation model of the flow in the tubes is developed using the CFD code Fluent 6.1 (1). As illustrated in Fig. 3, only one section of the collector with two horizontal tubes placed in a vertical plane is investigated.

Steady state numerical solutions are obtained for laminar flow with the Boussinesq approximation for buoyancy modelling. The velocity-pressure coupling is treated by using the SIMPLE algorithm and the First Order Upwind scheme is used for the momentum and energy terms.

2.1 Geometry

The model consists of the inner boundaries of the geometry. The outer glass tube, the evacuated space between the two glass tubes and the wall thickness of the inner glass tube are not included in the model. Also the outer casing and the insulation material of the manifold channel are not included in the model. This means that no solids are simulated — only the fluid is included in the model. The conduction in the inner glass wall is however included in the model. The geometry is summarized in Table 1.

The computational mesh is constructed in the pre-processing program Gambit 2.0.4 (2). The number of computational cells depends on the length of the tubes and is given in Table 1.

Fig. 4 shows a close up of the mesh near the manifold channel.

### Table 1: Geometry of the Numerical Models

| Side length of manifold channel: | 0.06 m |
| Glass tube inner diameter:      | 0.037 m |
| Glass tube outer diameter (used in manifold channel): | 0.047 m |
| Length of glass tube exposed to solar radiation: | 0.59 m 1.17 m 1.47 m |
| Number of computational cells:  | 428082 738708 948499 |
| Illustration of geometry:       | ![Illustration of geometry](image) |

Fig. 4: The mesh near the square manifold channel.

2.2 Boundary conditions

The solar irradiance is simulated as a distributed heat flux on the tube wall. The flux varies from 0 W/m² to 1150 W/m² as shown in Fig. 5. The average flux on the tube wall is 566 W/m².

![Distribution of the heat flux on the inner glass tubes](image)

Fig. 5: Distribution of the heat flux on the inner glass tubes.
The heat loss from the tubes is modelled with a heat loss coefficient of 0.85 W/m²K (3) and a constant ambient temperature of 293 K. The heat loss coefficient is related to the absorber area. The manifold channel is assumed to have zero heat loss.

Five different inlet mass flow rates of respectively 0.05 kg/min, 0.4 kg/min, 1 kg/min, 3 kg/min and 10 kg/min have been computed.

The inlet velocity profile has been found by first making a computation with a uniform inlet velocity profile. The outlet velocity profile from this simulation has then been used as the inlet velocity profile for the final simulation.

The inlet temperature has been 333 K during all computations. A 40% propylene-glycol/water mixture has been used as working fluid.

3 RESULTS

The presented results will include illustrations of flow patterns in the vertical centre plane of the model near the manifold channel and in the manifold outlet plane. Further, some overall analyses of the "collector" performance as a function of tube lengths and mass flow rates will be presented.

3.1 Flow distribution

For a tube length of 1.17 m, the left column in Fig. 6 shows velocity vectors in the vertical tube centre plane near the manifold channel for the five investigated inlet mass flow rates.

At the smallest mass flow rate (top-left picture) it can be seen how the fluid flows directly from the inlet in the manifold channel out to the two horizontal tubes. The fluid returns to the manifold channel along the top of the tubes. The two flows from the two tubes exits at the outlet with a profile, which is clearly formed by the flows in the tubes.

The flow patterns for the next two flow rates look similar to the flow pattern for the lowest flow rate; however, there is one significant difference. Due to the larger inlet velocities, the flow rises higher in the manifold channel before it, due to buoyancy forces, turns down to the tube bottom wall and flows out in the tubes. The differences in the forced- and buoyancy driven flows are evident just by seeing how far up in the manifold tube the flow rises.

For the two highest flow rates it is clear that some of the flow passes directly through the manifold channel without entering the tubes. Therefore, the outlet velocity profiles look different for the highest flow rates compared to the outlet velocity profiles for the lowest flow rates.

<table>
<thead>
<tr>
<th>Velocity vectors in the vertical tube centre plane:</th>
<th>Velocity contours in the outlet plane:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow=0.05 kg/min.</td>
<td>Flow=0.05 kg/min.</td>
</tr>
<tr>
<td>Flow=0.4 kg/min.</td>
<td>Flow=0.4 kg/min.</td>
</tr>
<tr>
<td>Flow=1 kg/min.</td>
<td>Flow=1 kg/min.</td>
</tr>
<tr>
<td>Flow=3 kg/min.</td>
<td>Flow=3 kg/min.</td>
</tr>
<tr>
<td>Flow=10 kg/min.</td>
<td>Flow=10 kg/min.</td>
</tr>
</tbody>
</table>

Fig. 6: Velocity patterns in the vertical tube centre plane (left column) and in the outlet plane (right column). Tube length: L=1.17 m.
The right column in Fig. 6 shows the velocity contours in the outlet plane. It can be seen how the velocity pattern changes from being dominated by the two flows from the tubes for the lowest inlet flow rates to, for the highest flow rates, being dominated mainly by the inlet flow.

3.2 Thermal performance

The thermal performance is investigated by calculating efficiencies for the different combinations of flow rates and tube lengths. The efficiency, \( \eta \), is defined as the ratio between the power out of the collector, \( P_{\text{outlet}} \), and the distributed heat flux absorbed by the collector, \( P_{\text{absorber}} \):

\[
\eta = \frac{P_{\text{outlet}}}{P_{\text{absorber}}}
\]

Notice that this efficiency cannot be compared with the traditional way of defining the collector efficiency as optical losses are not included in the efficiency used here.

Fig. 7 shows the efficiency as a function of the mass flow rate. The efficiency is highest for flow rates around 0.4 kg/min – 1 kg/min. The explanation for this result can be found in Fig. 8, which shows the mean temperature in the collector as a function of the mass flow rate. For the largest inlet flow rates (3 kg/min – 10 kg/min) a large part of the fluid flows directly through the manifold channel leaving only a smaller part flowing out in the tubes. Therefore, the average temperature in the whole collector rises. This leads to a higher heat loss and thus a lower efficiency. For the lowest flow rate (0.05 kg/min) almost all the inlet flow goes out in the tubes, but now the flow is so small that this alone leads to an increased average temperature in the whole collector.

That the efficiency in fact decreases with increasing average temperatures in the whole collector is very clear in Fig. 9. Here each dot represents an efficiency found at a given inlet flow rate. All the dots together form an almost straight line with a tilt that shows how the efficiency decreases with increasing heat loss caused by increasing average temperatures in the whole collector. The collector with the shortest tube has the highest efficiency and vice versa.

Finally, Fig. 10 shows the efficiency as a function of the average of the inlet- and outlet temperature. As in Fig. 7 and in Fig. 9 it can be seen that the highest efficiency is achieved for the shortest tube length.
The results further show that the inlet mass flow rate has a relatively small influence (1-2%) on the resulting efficiencies. This might seem strange considering the large differences in the flow patterns near the manifold channel (Fig. 6). The main reason is that the flow in the tubes at a distance from the manifold channel is relatively unaffected by the inlet flow.

For a tube length of L=1.17 m, Fig. 11 shows the maximum velocity magnitude as a function of the distance from the manifold centre. For the varying inlet flows, the figure shows that close to the manifold channel there are differences in the maximum velocity magnitude but further out in the tubes there are almost no differences in the maximum velocity magnitude. This explains why the absolute differences in the collector efficiencies are so small for the varying inlet flow conditions. Fig. 11 also shows that the flows out in the tubes are:

- Largest for an inlet flow of 0.4 kg/min
- 2nd largest for an inlet flow of 1 kg/min
- 3rd largest for an inlet flow of 3 kg/min
- 4th smallest for an inlet flow of 0.05 kg/min
- Smallest for an inlet flow of 10 kg/min

This corresponds nicely with the results presented in Fig. 7 where the order of efficiencies is the same.

![Graph showing maximum velocity magnitude in the tubes.](image)

**Fig. 11.** Maximum velocity magnitude in the tubes (outside the manifold channel) in the vertical centre plane.

Finally, it should be mentioned that another reason for the small differences in the calculated efficiencies is that the heat loss coefficient for the tubes is very small. Due to this low heat loss coefficient, the differences in the mean temperature in the whole collector have an only minor influence on the final efficiency.

4 CONCLUSIONS

The objective of this work was to investigate the heat transfer and the flow structures inside all glass evacuated tubular collectors for different operating conditions by means of CFD.

A collector design based on horizontal tubes connected to a manifold channel has been investigated. The working principle of the collector is that the collector fluid enters the bottom of the square manifold channel, circulates in the glass tubes and leaves at the top of the manifold channel.

Three different tube lengths of 0.59 m, 1.17 m and 1.47 m have been modelled with five different inlet mass flow rates of respectively 0.05 kg/min, 0.4 kg/min, 1 kg/min, 3 kg/min and 10 kg/min. The inlet temperature has been 333 K during all computations and a 40% propylene-glycol/water mixture has been used as the collector fluid. The average solar radiation absorbed on the tube wall has been 565 W/m².

Under these operating conditions the results showed that:

- the collector with the shortest tube length achieved the highest efficiency
- the optimal inlet flow rate was around 0.4-1 kg/min
- the flow structures in the glass tubes were relatively uninfluenced by the inlet flow rate

All the results were directly related to the resulting average temperature in the whole collector. The shortest tube length gave the lowest average temperature. The optimal inlet flow rate was around 0.4-1 kg/min as larger inlet flow rates (3 kg/min – 10 kg/min) gave a larger part of the fluid flowing directly through the manifold channel leaving only a smaller part flowing out in the tubes. Therefore, the average temperature in the whole collector rose. For the lowest flow rate (0.05 kg/min) almost all the inlet flow went out in the tubes, but here the flow was so small that this alone led to an increased average temperature in the whole collector.

Generally, the results showed only small variations in the efficiencies. This indicates that the collector design is well working for most operating conditions.

5 NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>η</td>
<td>collector efficiency [%]</td>
</tr>
<tr>
<td>( P_{\text{exh}} )</td>
<td>Power from collector [W]</td>
</tr>
<tr>
<td>( P_{\text{abs}} )</td>
<td>Radiation absorbed by the collector [W]</td>
</tr>
<tr>
<td>L</td>
<td>Tube length [m]</td>
</tr>
</tbody>
</table>

6 ACKNOWLEDGEMENTS

This study is financed by the VILUM KANN RASMUSSEN FOUNDATION.
7 REFERENCES


Utilization of Solar Radiation at High Latitudes with Evacuated Tubular Collectors

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Solar energy at high latitudes

- Large season variations
- Solar radiation from all directions
- High ground reflections due to snow
- Low ambient temperatures
Available solar radiation at high latitudes

Solar Radiation on horizontal [kWh/m²/year]

Solar collectors for northern latitudes

- When solar collectors are developed for high latitudes, it is an advantage if the collectors:
  - Can utilize solar radiation from all directions
  - Have a low heat loss (due to low ambient temperatures)
  - Utilize ground reflected radiation well (due to large ground reflection coefficient)
Evacuated tubular collectors

- Low heat loss coefficient
  - Vacuum insulation

- High efficiency
  - Due to low heat loss

- Utilize solar radiation from all directions
  - Double-sided absorbers with coating on both sides

- Mass produced in China
  - Low cost
Heat pipe evacuated tubes in a collector panel

Glass tubes

Manifold heat exchanger pipe

Inlet →

Outlet

Heat pipes

Fins

Heat pipe principle

Manifold heat exchanger with solar collector fluid (propylene glycol/water)

Heat pipe with working fluid (water)
Heat pipe evacuated tubular collectors with \textit{double-sided} absorbers

- \textbf{Research work}:
  - New collector theory is developed - TrnSys
  - Calculation yearly thermal performance is now possible
  - Detailed optimization of collector design is now possible

\textbf{Collector rows and shadows on collector and ground}

- $F_{\text{col}}$, $F_{\text{col,ground,back}}$, $F_{\text{col,ground,front}}$
- $R_{\text{shad}}$, $1-R_{\text{shad}}$, $F_{\text{ground,sky}}$
Collector array in a solar heating plant

- Weather data
  - Uummannaq, Greenland, Latitude 71°
  - Sisimiut, Greenland, Latitude 67°
  - Copenhagen, Denmark, Latitude 56°

Weather data
Global radiation

Global radiation [KWh/m²/month]

Jan Feb Mar Apr May Jun Jul Aug Sep Oct Nov Dec
Weather data
Ambient temperature

Example: Influence of collector tilt

(Transparent area = Glass tube cross section area)
Example: Influence of collector row distance

Collector performance in Copenhagen, Denmark
Collector performance in Uummannaq, Greenland

Observations – Collector performance

- Evacuated tubular collectors are relatively better performing in Uummannaq than in Copenhagen, compared to the flat plate collector.
- Does the evacuated tubular collectors with double-sided absorbers have an extra advantage compared to flat plate collectors at high latitudes due to the possibility of utilization of solar radiation from all directions?
Utilization of solar radiation in Uummannaq (1st of July)

- Flat plate collector: In operation during 13 h of the day
- Flat strip evacuated tubular collector: In operation during 19 h of the day
- Curved strip evacuated tubular collector: In operation during 18 h of the day
- The double sided absorbers do have an influence

Influence of double-sided absorbers

Relative performance = \( \frac{\text{Performance}_{\text{double-sided absorber}}}{\text{Performance}_{\text{single-sided absorber}}} \)

![Diagram showing relative thermal performance vs. collector operating temperature for different absorber configurations.](Diagram)

- Flat Strip (Tilt=45°, Row dist.=10m, Copenhagen)
- Flat Strip (Tilt=50°, Row dist.=10m, Sisimiut)
- Flat Strip (Tilt=60°, Row dist.=10m, Uummannaq)
Conclusions

- Compared to a flat plate collector, the heat pipe evacuated tubular collectors are relatively better performing in Uummannaq than in Copenhagen

- The influence of the double-sided absorber increases with increasing latitudes.

Further work

- Measurements
  - New test facility
  - Test of 5 differently designed evacuated tubular collectors
  - Direct performance comparison
  - Final validation of theoretical models

- Optimization work

- Design
  - Based on the findings and on economy considerations, well designed evacuated tubular collectors for Northern latitudes will be recommended
Thank you for your attention!
Bilag 16: ISES Solar World Congress præsentation: Theoretical investigations of differently designed heat pipe evacuated tubular collectors.
Theoretical Investigations of differently designed Heat Pipe Evacuated Tubular Collectors

Louise Jivan Shah and Simon Furbo
Department of Civil Engineering
Technical University of Denmark
E-mail: lis@byg.dtu.dk
Heat pipe evacuated tubes in a collector panel

Heat pipe with working fluid (water)

Manifold heat exchanger solar collector fluid (propylene glycol/water)

Heat pipe evacuated tubular collectors with flat or curved fins

- Vacuum tube
- Heat pipe Selective coating on both sides of fins

- Vacuum tube
- Heat pipe Curved fins Selective coating on both sides of fins

• Research work:
  - Two new TrmSys collector models are developed
  - Ground mounted collectors
  - Model input = Collector design
  - Detailed optimization of collector design is now possible
Detailed modeling of shadows and temperature distribution

Depending on the position of the sun and the distance between the tubes, the tubes cast shadow on each other.

The solar irradiance varies along the fin and the fin temperature must be calculated in detail.

Importance of detailed fin temperature modeling
Flat fin – No shadows – Constant incidence angle

Observation: Symmetry at all three times
Importance of detailed fin temperature modeling
Curved fin – No shadows – Varying incidence angle

Observation: Only symmetry at noon!

Other features for ground mounted collectors
Influence of collector rows

\[ R_{\text{ground,lay}} \]

\[ F_{\text{col,deg,shad}} \]

\[ F_{\text{col,deg,sun}} \]
Model input:
Collector geometry and material data

Example – Using the new theory
Collector array in a solar heating plant

- Weather data
  - Uummannaq, Greenland

- Parameter investigations:
  - Tube centre distance
  - Collector row distances

- Operation data
  - Operating temperature: 50°C

- Collector data
  - Tube length: 2 m
  - Tube diameter: 0.1 m
  - Tilt: 60°, Orientation: South
Tube centre distance:

Row distance:
Summary

- Developed two new Trnsys collector types for heat pipes with double sided absorbers
- Detailed view factor modelling
- Detailed shadow modelling
- Detailed fin temperature modelling
- Model input = Exact collector design
- Detailed optimization of collector design is now possible

Further work

- Measurements
  - New test facility
  - Test of 4 differently designed heat pipe evacuated tubular collectors
  - Direct performance comparison
  - Final validation of theoretical models
Thank you for your attention 😊
Bilag 17: ISES Solar World Congress presentation: Theoretical investigations of an all glass evacuated tubular collector.
Numerical Investigations of an All Glass Evacuated Tubular Collector

Louise Jivan Shah and Simon Furbo
Department of Civil Engineering
Technical University of Denmark
E-mail:
lls@byg.dtu.dk

All-glass evacuated tubular collectors
Investigation of flow and temperature patterns inside the tubes

• All-glass collector
• Heat transfer
• Flow structures
All-glass evacuated tubular collectors

All-glass working principle

Glass tubes with collector fluid (propylene glycol/water)

Manifold tube
Computational Fluid Dynamics (CFD)

- A grid is made of the geometry and governing equations of fluid flow are solved numerically.
- With CFD temperatures and flow patterns are determined.

CFD investigations:

Calculations:
- 5 different inlet flow rates:
  - 0.05 kg/min – 10 kg/min.
  - Realistic inlet velocity pattern
- Inlet temperature: 60 °C
- Uneven solar radiation

Results:
- Flow structures
- Influence on collector efficiency
Flow structures near the manifold pipe

- Large differences in flow patterns:
  - Buoyancy flow
  - Forced flow

Flow inside the All-glass tube

The horizontal All-glass tubes connected to the manifold pipe in the centre

Inlet flow = 0.05 kg/min
Inlet flow = 1 kg/min
Inlet flow = 3 kg/min
Calculated efficiency based on CFD calculations

Results for different inlet flow rates

Observation:
• The manifold inlet flow rate has only small (<2%) influence on collector efficiency

Explanation:
• Small variations in the flow patterns in the glass tubes
• Self adjusting flow in glass tubes
Summary

- Example of how Computational Fluid Dynamics gives detailed useful results

- Optimal manifold inlet flow rate is around 0.4-1 kg/min.

- Flow structures in the glass tubes were relatively uninfluenced by the manifold inlet flow rate

- This indicates that the collector design is well working for most operating conditions.

Thank you for your attention 🌞
Bilag 18: Præsentation ved Solar seminar ved Beijing Solar Energy Institute, Beijing, Kina: Side-by-side tests of Seido collectors
Side-by-side tests of Seido collectors

Louise Jivan Shah
Department of Civil Engineering
Technical University of Denmark
E-mail: ljs@byg.dtu.dk

Overview

• Test facility
• Measurements
• Simulation models
• Parameter studies
• Conclusion
Test facility
Side-by-side tests of evacuated tubular collectors
Double-sided absorbers

- Absorber coating on both sides of the fin
Collectors with double sided absorbers can utilize solar radiation from all directions.

Measurement principle

- Side-by-side tests:
- The collectors are tested under the same operating conditions:
  - Solar radiation
  - Ambient temperature
  - Collector inlet temperature
  - Flow rate
- Direct performance comparison
Measurement period

• 28. august – now (Late summer – Autumn)

Collector operating temperature
Performance difference
27/8 – 19/10 2005

Power from collector
Summer: 1/9 2005
Power from collector

Autumn: 12/10 2005

Position of sun

Solar Azimuth Angle [°]

Solar Time

COPENHAGEN
Manifold heat loss coefficient

<table>
<thead>
<tr>
<th>Seido 5-8</th>
<th>0.4 W/K</th>
</tr>
</thead>
<tbody>
<tr>
<td>Seido 1-8</td>
<td>0.6 W/K</td>
</tr>
<tr>
<td>Seido 10-20</td>
<td>0.8 W/K</td>
</tr>
<tr>
<td>Seido 10-20</td>
<td>1.1 W/K</td>
</tr>
</tbody>
</table>

Preliminary conclusions from measurements

- All collectors were easily installed and operate without any problems
- The flat fin collectors perform better than the curved fin collectors
  - Less significant during summertime
  - Main reason is due to different effective incidence angles
- There are significant differences in the manifold heat loss coefficients
  - The manifold pipes for the flat fin collectors have higher heat losses than the manifold pipes for the curved fin collectors
Theoretical work

- Research work:
  - Two new TmSys collector models are developed
  - Model input = Collector design
  - Detailed optimization of collector design is now possible

Detailed modeling of shadows and temperature distribution

Depending on the position of the sun and the distance between the tubes, the tubes cast shadow on each other

The solar irradiance varies along the fin and the fin temperature must be calculated in detail
Importance of detailed fin temperature modeling

Flat fin – No shadows – Constant incidence angle

Observation: Symmetry at all three times

Importance of detailed fin temperature modeling

Curved fin – No shadows – Varying incidence angle

Observation: Only symmetry at noon!
Model input:
Collector geometry and material data

Calculation examples

- Copenhagen weather data
- Simulation of yearly thermal performance
- Glass tube diameter = 0.1 m
- Collector operating temperature = 30°C
- Lowest evaporation temperature = 25°C
- Flat fin heat pipe evacuated tubular collector
Conductivity of fin material

Fin width
Manifold heat exchange rate

Average manifold temperature
Preliminary conclusions from theoretical work

- Developed two new Trnsys collector types for heat pipes with double sided absorbers
- Detailed view factor modelling
- Detailed shadow modelling
- Detailed fin temperature modelling
- Model input = Exact collector design
- Detailed optimization of collector design is now possible
Further work

- Measurements
- Final validation of theoretical models
- Parameter studies
  - Absorber materials
  - Collector design
- Cost performance analyses

Thank you for your attention
Bilag 19: Theoretical flow investigations of an all glass evacuated tubular collector
THEORETICAL FLOW INVESTIGATIONS OF AN ALL GLASS EVACUATED TUBULAR COLLECTOR

Louise Jivan Shah and Simon Furbo
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ABSTRACT

Heat transfer and flow structures inside all glass evacuated tubular collectors for different operating conditions are investigated by means of Computational Fluid Dynamics (CFD). The investigations are based on a collector design with horizontal tubes connected to a vertical manifold channel. Three different tube lengths varying from 0.59 m to 1.47 m have been modelled with five different inlet mass flow rates varying from 0.05 kg/min to 10 kg/min with a constant inlet temperature of 333 K. Under these operating conditions the results showed that:

• the collector with the shortest tube length achieved the highest efficiency
• the optimal inlet flow rate was around 0.4-1 kg/min
• the flow structures in the glass tubes were relatively uninfluenced by the inlet flow rate

Generally, the results showed only small variations in the efficiencies. This indicates that the collector design is well working for most operating conditions.

KEYWORDS

All glass evacuated tubular collector, CFD calculations, flow structure

introduction

All glass evacuated tubular collectors are widely used on the world market. All glass evacuated tubular collectors are based on double glass tubes where the outside of the inner glass wall is treated with an absorbing selective coating and the evacuated space is between the tubes as illustrated in Fig. 12. A collector design based on horizontal tubes connected to a manifold pipe is especially popular due to its low cost. An illustration of the collector design is shown in Fig. 13. The collector fluid enters the bottom of the square manifold channel and leaves at the top of the manifold channel. The intended flow inside the glass tubes is indicated with the arrows. Obviously, the flow patterns inside the tubes are of big importance for the thermal performance of the collector tubes. Therefore it is of interest how for example the
operating conditions and the collector geometry influence the flow patterns in the tubes and thus the collector performance.

Flow patterns inside all-glass evacuated tubular collectors have previously been addressed. Zhiqiang and Harding (1984) made measurements for a range of tube inclinations, manifold flow rates and inlet temperatures. The results showed that for a wide range of operating conditions, buoyancy effects alone resulted in efficient heat transfer. With dye experiments, Wang et al. (1987) visualized the flow patterns inside horizontal tubes connected to a vertical manifold pipe. Here the solar radiation was imitated by a translucent electrical heating film deposited on the outer side of the glass tube. The visualization clearly showed buoyancy effects. Wang et al. (1989) continued their investigations and studied the natural convection heat transfer process inside the tubes. They developed non-dimensional heat transfer formulas and it was found that there exists an optimal Re-number in the manifold pipe; for this Re-number the heat transfer in the tube reached maximum.

With laser Doppler anemometry, Gaa et al. (1996) investigated flow inside an inclined cylindrical open thermosyphon corresponding to a glass tube, where the wall could be heated to either a uniform temperature or a differentiated temperature where the upper part of the wall was heated to a higher temperature than the lower part of the wall. Based on the measurements they performed correlations between the velocity and the control parameters (Ra-number, aspect ratio and heating mode). Gaa et al. (1998) continued their investigations on 45° inclined open thermosyphon experimentally and numerically. For the differential temperature boundary conditions, the results showed that the flow was typically bifilamental. For the uniform temperature boundary condition, however, it was shown that the flow started out by being bifilamental at low Ra-numbers ($10^3$), similar to the flow for differential wall heating, but at higher Ra-numbers ($10^5$) an upward flow was developed from the bottom surface making the flow annular.

Morrison et al. (2004) conducted a numerical study of a 45° inclined evacuated tube. Among other things they found the possible presence of a stagnant region in the bottom of very long tubes. Morrison et al. (2005) continued their investigations on the tubes connected to a water tank. The investigations were carried out both experimentally with Particle Image Velocimetry and numerically with Computational Fluid Dynamics. It was found that the natural convection flow rate in the tube was high enough to disturb the tank’s stratification. Also it was found that the tank temperature strongly affects the circulation flow rate through the tubes. Circumferential heat distribution was found to be an important parameter influencing the flow structure and circulation rate through the tube.

The objective of the present work is to investigate the heat transfer and the flow structures inside horizontal tubes connected to a vertical manifold pipe. The investigations will be carried out for different flow rates and tube geometries by means of Computational Fluid Dynamics (CFD).

**Numerical investigations**

To solve the flow and energy equations in the glass tubes, a simulation model of the flow in the tubes is developed using the CFD code Fluent 6.1 (Fluent 6.1 User’s Guide, 2003). As illustrated in Fig. 14, only one section of the collector with two horizontal tubes placed in a vertical plane is investigated.
Steady state numerical solutions are obtained for laminar flow with the Boussinesq approximation for buoyancy modelling. The velocity-pressure coupling is treated by using the SIMPLE algorithm and the First Order Upwind scheme is used for the momentum and energy terms.

**Geometry**

The model consists of the inner boundaries of the geometry. The outer glass tube, the evacuated space between the two glass tubes and the wall thickness of the inner glass tube are not included in the model. Also the outer casing and the insulation material of the manifold channel are not included in the model. This means that no solids are simulated – only the fluid is included in the model. The conduction in the inner glass wall is however included in the model. The geometry is summarized in Table 1. The computational mesh is constructed in the pre-processing program Gambit 2.0.4 (Gambit 2 User’s Guide, 2001). The number of computational cells depends on the length of the tubes and is given in Table 1. Fig. 15 shows a close up of the mesh near the manifold channel.

**Boundary conditions**

The solar irradiance is simulated as a distributed heat flux on the tube wall. The flux varies from 0 W/m² to 1150 W/m² as shown in Fig. 16. The average flux on the tube wall is 566 W/m².

The heat loss from the tubes is modelled with a heat loss coefficient of 0.85 W/m²K (Qin L. and Furbo S., 1999) and a constant ambient temperature of 293 K. The heat loss coefficient is related to the absorber area. The manifold channel is assumed to have zero heat loss.

Five different inlet mass flow rates of respectively 0.05 kg/min, 0.4 kg/min, 1 kg/min, 3 kg/min and 10 kg/min have been computed.

The inlet velocity profile has been found by first making a computation with a uniform inlet velocity profile. The outlet velocity profile from this simulation has then been used as the inlet velocity profile for the final simulation.

The inlet temperature has been 333 K during all computations. A 40% propylene-glycol/water mixture has been used as working fluid.

**Results**

The presented results will include illustrations of flow patterns in the vertical centre plane of the model near the manifold channel and in the manifold outlet plane. Further, some overall analyses of the “collector” performance as a function of tube lengths and mass flow rates will be presented.

**Flow distribution**

For a tube length of 1.17 m, the left column in Fig. 17 shows velocity vectors in the vertical tube centre plane near the manifold channel for the five investigated inlet mass flow rates.
At the smallest mass flow rate (top-left picture) it can be seen how the fluid flows directly from the inlet in the manifold channel out in the two horizontal tubes. The fluid returns to the manifold channel along the top of the tubes. The two flows from the two tubes exits at the outlet with a profile, which is clearly formed by the flows in the tubes.

The flow patterns for the next two flow rates look similar to the flow pattern for the lowest flow rate; however, there is one significant difference. Due to the larger inlet velocities, the flow rises higher in the manifold channel before it, due to buoyancy forces, turns down to the tube bottom wall and flows out in the tubes. The differences in the forced- and buoyancy driven flows are evident just by seeing how far up in the manifold tube the flow rises.

For the two highest flow rates it is clear that some of the flow passes directly through the manifold channel without entering the tubes. Therefore, the outlet velocity profiles look different for the highest flow rates compared to the outlet velocity profiles for the lowest flow rates.

The right column in Fig. 17 shows the velocity contours in the outlet plane. It can be seen how the velocity pattern changes from being dominated by the two flows from the tubes for the lowest inlet flow rates to, for the highest flow rates, being dominated mainly by the inlet flow.

**Thermal performance**

The thermal performance is investigated by calculating efficiencies for the different combinations of flow rates and tube lengths. The efficiency, \( \eta \), is defined as the ratio between the power out of the collector, \( P_{\text{collector}} \), and the distributed heat flux absorbed by the collector, \( P_{\text{solar}} \):

\[
\eta = \frac{P_{\text{collector}}}{P_{\text{solar}}}
\]

Notice that this efficiency cannot be compared with the traditional way of defining the collector efficiency as optical losses are not included in the efficiency used here.

Fig. 18 shows the efficiency as a function of the mass flow rate. The efficiency is highest for flow rates around 0.4 kg/min – 1 kg/min. The explanation for this result can be found in Fig. 19, which shows the mean solar collector fluid temperature in the tubes as a function of the mass flow rate. For the largest inlet flow rates (3 kg/min – 10 kg/min) a large part of the fluid flows directly through the manifold channel leaving only a smaller part flowing out in the tubes. Therefore, the average temperature in the tubes rises. This leads to a higher heat loss and thus a lower efficiency. For the lowest flow rate (0.05 kg/min) almost all the inlet flow goes out in the tubes, but now the flow is so small that this alone leads to an increased average temperature in the tubes.

That the efficiency in fact decreases with increasing average solar collector fluid temperatures in the tubes is very clear in Fig. 20. The average solar collector fluid temperature in the tubes is determined by means of volume weighted temperatures. Here each dot represents an efficiency found at a given inlet flow rate. All the dots together form an almost straight line with a tilt that shows how the efficiency decreases with increasing heat loss caused by increasing average temperatures. The collector with the shortest tube has the highest efficiency and vice versa.

Finally, Fig. 21 shows the efficiency as a function of the average of the inlet- and outlet temperature. As in Fig. 18 and in Fig. 20 it can be seen that the highest efficiency is
achieved for the shortest tube length. The efficiency drop observed in Fig. 10 for decreasing solar collector fluid temperature determined as average of the inlet and outlet temperature is caused by the high flow rate resulting in an increased mean solar collector fluid temperature in the tubes.

The results further show that the inlet mass flow rate has a relatively small influence (1-2 %) on the resulting efficiencies. This might seem strange considering the large differences in the flow patterns near the manifold channel (Fig. 17). The main reason is that the flow in the tubes at a distance from the manifold channel is relatively unaffected by the inlet flow. For a tube length of L=1.17 m, Error! Reference source not found. shows the maximum velocity magnitude as a function of the distance from the manifold centre. For the varying inlet flows, the figure shows that close to the manifold channel there are differences in the maximum velocity magnitude but further out in the tubes there are almost no differences in the maximum velocity magnitude. This explains why the absolute differences in the collector efficiencies are so small for the varying inlet flow conditions. Error! Reference source not found. also shows that the flows out in the tubes are:

- largest for an inlet flow of 0.4 kg/min
- 2nd largest for an inlet flow of 1 kg/min
- 3rd largest for an inlet flow of 3 kg/min
- 2nd smallest for an inlet flow of 0.05 kg/min and
- smallest for an inlet flow of 10 kg/min

This corresponds nicely with the results presented in Fig. 18 where the order of efficiencies is the same.

Finally, it should be mentioned that another reason for the small differences in the calculated efficiencies is that the heat loss coefficient for the tubes is very small. Due to this low heat loss coefficient, the differences in the mean temperature in the whole collector have an only minor influence on the final efficiency.

conclusions

The objective of this work was to investigate the heat transfer and the flow structures inside all glass evacuated tubular collectors for different operating conditions by means of CFD.

A collector design based on horizontal tubes connected to a manifold channel has been investigated. The working principle of the collector is that the collector fluid enters the bottom of the square manifold channel, circulates in the glass tubes and leaves at the top of the manifold channel.

Three different tube lengths of 0.59 m, 1.17 m and 1.47 m have been modelled with five different inlet mass flow rates of respectively 0.05 kg/min, 0.4 kg/min, 1 kg/min, 3 kg/min and 10 kg/min. The inlet temperature has been 333 K during all computations and a 40% propylene-glycol/water mixture has been used as the collector fluid. The average solar radiation absorbed on the tube wall has been 566 W/m².
Under these operating conditions the results showed that:

- the collector with the shortest tube length achieved the highest efficiency

- the optimal inlet flow rate was around 0.4-1 kg/min

- the flow structures in the glass tubes were relatively uninfluenced by the inlet flow rate

All the results were directly related to the resulting average temperature in the whole collector. The shortest tube length gave the lowest average temperature. The optimal inlet flow rate was around 0.4-1 kg/min as larger inlet flow rates (3 kg/min – 10 kg/min) gave a larger part of the fluid flowing directly through the manifold channel leaving only a smaller part flowing out in the tubes. Therefore, the average temperature in the whole collector rose. For the lowest flow rate (0.05 kg/min) almost all the inlet flow went out in the tubes, but here the flow was so small that this alone lead to an increased average temperature in the whole collector.

Generally, the results showed only small variations in the efficiencies. This indicates that the collector design is well working for most operating conditions.

**nomenclature**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tbody>
<tr>
<td>η</td>
<td>collector efficiency</td>
<td>[-]</td>
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<tr>
<td>( P_{\text{collector}} )</td>
<td>Power from collector</td>
<td>[W]</td>
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<tr>
<td>( P_{\text{solar}} )</td>
<td>Radiation absorbed by the collector</td>
<td>[W]</td>
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<tr>
<td>L</td>
<td>Tube length</td>
<td>[m]</td>
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**acknowledgements**

This study is financed by the VILLUM KANN RASMUSSEN FOUNDATION.

**references**


Huo Z., Yan X., Zhang L., 1991. Experimental Study on N-S Orientation All Glass


<table>
<thead>
<tr>
<th>Description</th>
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<tr>
<td>Side length of manifold channel:</td>
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<tr>
<td>Glass tube inner diameter:</td>
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<tr>
<td>Glass tube outer diameter (used in manifold channel):</td>
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<tr>
<td>Length of glass tube exposed to solar radiation:</td>
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</tr>
<tr>
<td></td>
<td>0.59 m</td>
</tr>
<tr>
<td></td>
<td>1.17 m</td>
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<td>1.47 m</td>
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Figure captions

Fig. 1: Design of an all glass evacuated tube.

Fig. 2: Illustration of an all glass evacuated tubular collector with horizontal tubes.

Fig. 3: Illustration of an all glass evacuated tubular collector with horizontal tubes.

Fig. 4: The mesh near the square manifold channel.

Fig. 5: Distribution of the heat flux on the inner glass tubes.

Fig. 6: Velocity patterns in the vertical tube centre plane (left column) and in the outlet plane (right column). Tube length: L=1.17 m.

Fig. 7: The efficiency as a function of the mass flow rate.

Fig. 8: The mean solar collector fluid temperature in the tubes as a function of the mass flow rate.

Fig. 9: The efficiency as a function of the solar collector fluid temperature in the tubes.

Fig. 10: The efficiency as a function of the average of the inlet- and outlet temperature.

Fig. 11: Maximum velocity magnitude in the tubes (outside the manifold channel) in the vertical centre plane.
Fig. 12: Design of an all glass evacuated tube.
Fig. 13: Illustration of an all glass evacuated tubular collector with horizontal tubes.
Fig. 14: Illustration of an all glass evacuated tubular collector with horizontal tubes.
Fig. 15: The mesh near the square manifold channel.
Maximum irradiance: 1150 W/m²
Minimum irradiance: 0 W/m²

Fig. 16: Distribution of the heat flux on the inner glass tubes.
<table>
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<tr>
<th>Velocity vectors in the vertical tube centre plane:</th>
<th>Velocity contours in the outlet plane:</th>
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Scale:

0 0.043 0.087 0.13 m/s
Fig. 17: Velocity patterns in the vertical tube centre plane (left column) and in the outlet plane (right column). Tube length: $L=1.17$ m.
Fig. 18: The efficiency as a function of the mass flow rate.
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Fig. 20: The efficiency as a function of the mean solar collector fluid temperature in the tubes.
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**Fig. 11:** Maximum velocity magnitude in the tubes (outside the manifold channel) in the vertical centre plane.