Analysis of Solar Flat-plate Collectors

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Solar Thermal Collectors, 24.3.2015, Palma de Mallorca
Content of lectures today

- design tool KOLEKKTOR for theoretical analysis of flat-plate collectors
  - introduction into software
  - downloading, installation, support files, ...
- theoretical modelling of collector heat output „step by step“
  - external energy balance of absorber
  - internal energy balance of absorber
- what influences annual energy yields of collector
- examples solved with use of KOLEKKTOR and annual simulation in TRNSYS
Design Tool KOLEKTOR 2.2 for Virtual Prototyping of Solar Flat-plate Collectors

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Solar Thermal Collectors, 24.3.2015, Palma de Mallorca
Background & Motivation

Lack of software for solar thermal collector modeling:

- detailed parameters of solar collector
- detailed calculation of heat transfer
- user friendly
- general use
- freely available CoDePro (Madison), TRNSYS Type 103 (CSTB)

- wide application range for FP collectors (evacuated, building integrated, transparent cover structures, HT fluids)
- versatile for different heat transfer models (convection, sky radiation)
Model inputs and outputs

Inputs:

- detailed parameters of individual parts of solar collector geometry, thermophysical parameters, optical parameters
- climatic conditions: ambient temperature, irradiation, wind velocity
- operation conditions: input temperature $t_{in}$, mass flowrate $m$, slope $\beta$

Outputs:

- efficiency curve $\eta$ (based on aperture area $A_a$)
- output temperature from solar collector $t_{out}$
- detailed temperature distribution in collector, heat transfer coefficients
Principle of calculation

- **external balance of absorber**
  heat flow from absorber surface to ambient
  temperature distribution at main collector levels (nodes $p$, $z$, $b$)
  overall $U$-value of collector

- **internal balance of absorber**
  heat flow from abs.surface to fluid
  heat removal factor $F_R$
  absorber temperature $t_{abs}$
  heat gain $Q_u$

\[
\dot{Q}_u = A_a F_R [\tau \alpha G - U(t_{in} - t_a)]
\]
External balance of absorber

- heat transfer coefficients calculation based on temperatures of main nodes
- iteration loop for temperature of nodes (starting values based on $t_{abs}$, *U*-value calculation, reverse calculation of temperature distribution)
- starting absorber temperature is estimated from input temperature ($t_{abs} = t_{in} + 10$ K)
Internal balance of absorber

- heat transfer by fin conduction, bond conduction and forced convection in pipes
- iteration loop for mean fluid temperature (starting values based on $t_{in}$, $F_R$ calculation, reverse calculation of mean fluid temperature), absorber temperature calculation
- starting mean fluid temperature is estimated from input temperature ($t_m = t_{in} + 10 \text{ K}$)
Calculation procedure

external and internal balance iteration loops (5 loops enough)
superior iteration loop governing output / inputs
Design tool KOLEKTOR 2.2 (2009)

- model converted into program based on Visual Basic Studio
- concept of “cards” for data inputs
- selections in rolling windows
- default settings
- saving settings of parameters, loading settings
- installation pack 400 kB (zipped)
- installation for Win XP presume Microsoft.NET Framework installed
- new Windows include the Framework environment
Features and availability

- user-friendly software KOLEKТОR 2.2 is useful for analysis, design and prototyping
- several experimental validations have been performed based on commercial collectors testing
- software tool KOLEKТОR 2.2 is freely available at: [http://users.fs.cvut.cz/tomas.matuska/?page_id=194](http://users.fs.cvut.cz/tomas.matuska/?page_id=194)
- mathematical reference handbook as pdf
  - theoretical background of the tool
  - downloadable
Preparation for installation

- local settings in Windows - change to Great Britain settings
  - e.g. windows 8 – Control Panel – Language
  - change of datum, time or number format
  - further settings – using dot in decimal values

do it now ...
Download KOLEKTOŘ and install now ...

- [http://users.fs.cvut.cz/tomas.matuska/?page_id=194](http://users.fs.cvut.cz/tomas.matuska/?page_id=194)

- download **kolektor22.zip**

- unzip it Program files – folder Kolektor will be created

- start setup.exe – start installation process

- setup also opens the program

- for Windows versions higher than XP you don’t need **Microsoft.NET Framework** should be included in ...
Design tool KOLEKTOR 2.2 (2009)

Software for calculation of liquid flat-plate solar collector efficiency

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Design tool KOLEKTO 2.2
Design tool KOLEKTOR 2.2

**Absorber parameters**
- Material: Copper
- Thermal conductivity: $\lambda_{\text{abs}} = 300$ W/mK
- Thickness: $d_{\text{abs}} = 0.2$ mm

**Pipe register parameters**
- Length of riser pipes: $L = 2$ m
- Number of riser pipes: $n_{\text{tp}} = 10$
- Distance between riser pipes (in): $W = 100$ mm
- Pipe external diameter: $D_{\text{e}} = 10$ mm
- Pipe internal diameter: $D_{\text{i}} = 8$ mm

**Heat transfer fluid**
- Fluid type: Water
- Mixing ratio: 100%
- Freezing temperature: 0°C

**Additional parameters**
- Solar absorbance: $\alpha_{\text{abs}} = 0.95$
- Front surface emissivity: $\varepsilon_{\text{abs,0}} = 0.05$
- Back surface emissivity: $\varepsilon_{\text{abs,2}} = 0.5$
- Collector mass flow rate: $M^* = 0.04$ kg/s
- Pipe mass flow rate: $M'^* = 0.004$ kg/s
Design tool KOLEKTOR 2.2

- **Glazing parameters**
  - Material: Glass
  - Thickness: 4 mm
  - Normal solar transmittance: 0.92
  - Normal solar reflectance: 0.06
  - Diffuse solar reflectance: 0.06
  - External surface emissivity: 0.85
  - Internal surface emissivity: 0.85

- **Thermal properties**
  - Thermal conductivity: \( \lambda = 0.8 \text{ W/mK} \)
  - \( \lambda_1 = 0 \text{ W/mK}^2 \)
  - \( \lambda_2 = 0 \text{ W/mK}^2 \)

- **Frame / Insulation parameters**
  - Material: Polyurethane
  - Thickness: 50 mm
  - Thermal conductivity: \( \lambda_{fr} = 0.035 \text{ W/mK} \)
  - Thermal resistance: \( R_{fr} = 1.43 \text{ mK/W} \)
  - External frame surface emissivity: 0.5
  - Internal frame surface emissivity: 0.5

- **Gas filling of collector interior**
  - Type of gas: Air
  - Gas pressure: 100 kPa

- **Optical efficiency of collector**
  - Effective \( \tau \) product: 0.874

**VZTech**
Design tool KOLEKTOR 2.2
Default example – usual collector

- download as default.kol File – Open – default.kol

http://users.fs.cvut.cz/tomas.matuska/?page_id=194 as zip file

- main default collector parameters
  - solar glazing, transmittance 91 %
  - standard flat-plate collector with selective coating:
    $\varepsilon = 5 \% \quad \alpha = 95 \%$
  - cooper absorber 0.2 mm, risers 10/8 mm, distance 112 mm
  - absorber – glazing gap 20 mm, absorber – insulation gap 5 mm
  - insulation thickness 30 mm, mineral wool
  - slope of collector 45°
Calculation possibilities in collector

- calculation for given $t_{in}$
- analysis of heat transfer at individual parts of collector
- you can play with models ... and see the impact
Calculation possibilities in collector

- **efficiency curve calculation**

- range if $t_{in}$ from 10 °C to 400 °C
to cover also high-temperature collectors

- results can be saved to file = text file, columns separated by semicolon ;

- get the efficiency coefficients $\eta_0$, $a_1$, $a_2$ by use of **evaluation.xls**


- **important** – set the dots for decimals also in Excel settings
Solar DHW simulation in TRNSYS

- **download** example SDHW – simple solar system, modified for this Solnet course
- climate data Wurzburg (in data folder)
- daily load 200 l, cold water 10 °C, hot water 55 °C
- collector area 4 m², storage tank 200 l, no pipes ... simplified
- **annual energy yields for changed construction of collector**
- reference case:
  - annual yields of collector 1887 kWh/a
  - solar fraction 50 %
Analysis of Solar Flat-plate Collectors – part 1

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how to calculate thermal output of solar collector on theoretical basis
  - from detailed data on construction
  - at given operation conditions
  - with use of different alternative models of heat transfer phenomena

to predict the influence of
  - change in construction of the collector
  - operation conditions

on efficiency curve

on annual energy yields of the collector
Solar thermal flat-plate collector

- Transparent cover
- Absorber
- Header to draw the heat off
- Insulation
- Riser pipes with heat transfer fluid
- Collector frame
- Cover glazing
- Thermal insulation
- Full plate absorber
- Frame
- Back frame
- Header
- Riser
Solar collector energy balance

- Heat loss through glazing
- Reflection from absorber
- Reflection from cover
- Solar irradiation
- Heat loss through edge and back sides of collector frame
- Useful heat removal
Solar collector energy balance

- **External energy balance of absorber**
  - heat flow from absorber surface to ambient environment
  - heat losses of the collector
  - quality of solar collector envelope

- **Internal energy balance of absorber**
  - heat flow from absorber surface into heat transfer fluid
  - ability to transfer heat and remove it from collector
  - quality of absorber construction
External energy flows balance of absorber

\[
\frac{dQ}{dt} = \dot{Q}_s - \dot{Q}_{l,o} - \dot{Q}_{l,t} - \dot{Q}_u \quad \text{general description}
\]

steady state \( \frac{dQ}{dt} = 0 \) \( \dot{Q}_u = \dot{Q}_s - \dot{Q}_{l,o} - \dot{Q}_{l,t} \)

- \( Q_s \): solar energy input [W]
- \( Q_{l,o} \): optical losses [W]
- \( Q_{l,t} \): thermal losses [W]
- \( Q_u \): useful heat removed from collector [W]

\( Q_s = G.A_c \)
\( Q_{l,o} = Q_s - Q_s(\tau \alpha)_{ef} \)
\( Q_{l,t} = U.A_c (t_{abs} - t_a) \)
\( Q = M(c(t_{out} - t_{in}) \)
External energy balance: efficiency

useful heat output:

\[ \dot{Q}_u = G A_c (\tau \alpha)_{ef} - U A_c (t_{abs} - t_a) \]

efficiency:

\[ \eta = \frac{\dot{Q}_u}{\dot{Q}_s} = \frac{\dot{Q}_u}{G A_c} = \frac{G A_c (\tau \alpha)_{ef} - U A_c (t_{abs} - t_a)}{G A_c} \]

\[ \eta = (\tau \alpha)_{ef} - U \frac{t_{abs} - t_a}{G} \]
Efficiency formulas

\[ \eta = (\tau \alpha)_{\text{ef}} - U \frac{(t_{\text{abs}} - t_a)}{G} \]

- external balance, losses

\[ \eta = F \left[ (\tau \alpha)_{\text{ef}} - U \frac{(t_m - t_a)}{G} \right] \]

- + internal balance
  - influence of absorber design
  - European standards

\[ \eta = F_R \left[ (\tau \alpha)_{\text{ef}} - U \frac{(t_{\text{in}} - t_a)}{G} \right] \]

- + influence of flowrate,
  - specific heat of fluid
  - US standards
Solar collector efficiency (based on $t_{abs}$)

\[ \eta = (\tau \alpha)_{ef} \left( U \frac{t_{abs} - t_a}{G} \right) \]

$\eta$ [-]

$(t_{abs} - t_e)/G$ [m$^2$.K/W]
Reference area for calculations

gross area: $A_G$

edge side area: $A_b$

aperture area: $A_a$

maximum area = $H_G \cdot L_G$

$\left(2H_G + 2L_G\right) \cdot B$

area of glazing = $H_a \cdot L_a$
Reference area - comparison

Collector testing standards relates collector efficiency to:

- aperture area $A_a$
- absorber area $A_A$
- gross area $A_G$

- aperture area: for comparison of collector properties or design
- gross area: for decision on roof production potential
  for comparison of collector with different active areas

EN 12975-2
EN ISO 9806
Reference area - comparison
External energy balance: detailed

\[
\dot{Q}_u = G A_a (\tau \alpha)_{ef} = U_p A_G (t_{abs} - t_a) - U_z A_G (t_{abs} - t_a) - U_b A_b (t_{abs} - t_a)
\]

- \( U_p \) = front \( U \)-value
- \( U_z \) = back \( U \)-value
- \( U_b \) = side \( U \)-value
Scheme of external balance
Wind convection

- Wind induced convection heat transfer from cover to ambient:
  - mix of natural and forced convection
  - turbulent environment
  - very dependent on location, terrain morphology, barriers, etc.

- Number of models existing for number of boundary conditions
  - wind tunnel measurements (low turbulence)
  - outdoor experiments (on roof, on ground, urban, countryside)

- Most of wind convection models
  - empirical linear function  \( h_w = a + bw \)
Wind convection

for list of models
see Reference handbook on web

how the largest difference in models
influence the calculation

more realistic

McAdams
Test
Sharples
Watmuff

wind tunnels

Theory
Johnson
Sparrow
Wind convection for $w = 3 \text{ m/s}$

- how can the model influence calculated output temperature?
- run KOLEKTOR programme
- open default.kol
- Design parameters wind velocity $w = 3 \text{ m/s}$
- Calculation card select Calculation for given $t_{\text{in}}$
  - select McAdams ... Calculate
  - select Watmuff ... Calculate

\begin{align*}
\text{McAdam’s correlation}\hspace{1cm}&\text{Watmuff correlation} \\
 h_w &= 16.1 \text{ W/m}^2\text{K} & h_w &= 11.3 \text{ W/m}^2\text{K} \\
t_{\text{out}} &= 55.6 \text{ °C} & t_{\text{out}} &= 55.7 \text{ °C}
\end{align*}
Wind convection for $w = 3 \text{ m/s}$

- how can the model selection influence the efficiency curve?
  - Calculation card select Efficiency curve calculation
  - select McAdams Calculate, Results export to file
  - select Watmuff Calculate, Results export to file

- open Evaluation.xls
- make copies of the sheets
- open res files in excel, mind the semicolons as separators
- compare the efficiency curves based on $(t_m - t_e)/G$
Wind convection for $w = 3 \text{ m/s}$
Wind convection for \( w = 3 \text{ m/s} \)

- how can the model selection influence the annual energy yields?
  - McAdams: \( \eta_0 = 0.789 \quad a_1 = 4.857 \quad a_2 = 0.006 \)
  - Watmuff: \( \eta_0 = 0.791 \quad a_1 = 4.664 \quad a_2 = 0.006 \)

- open studio SDHW-SOLNET in TRNSYS
- make a change of coefficients in KOLEKTOR type
- results of collector yield find under Totals.txt
- compare the annual sums for both alternatives
Wind convection for $w = 3 \text{ m/s}$

- **McAdams:** 1887 kWh/a
- **Watmuff:** 1911 kWh/a

1.2 %
Wind convection for \( w = 3 \) m/s

- selection of the wind convection models
- minor influence even if two models with largest difference selected
  - valid for usual quality flat-plate collectors (as our reference is)

- negligible difference for high quality flat-plate collectors

- very critical for unglazed collectors modelling
- less critical for PV modules modelling
  - reduced influence on electric yields via temperature coefficients
High wind velocity as test condition

Testing standard: Air velocity during the testing procedure $w > 3 \text{ m/s}$
Use of artificial wind needed!
Sky radiation

Radiation heat exchange

\[ q_{p1-o} = \varepsilon_{p1}\sigma(T_{p1}^4 - T_o^4) \quad [\text{W/m}^2] \]

Sky temperature, sky emittance

\[ T_o^4 = \varepsilon_o T_a^4 \quad [\text{K}] \]

Sky radiation heat transfer coefficient related to ambient temperature

\[ h_{s,p1-a} = \varepsilon_{p1}\sigma \frac{T_{p1}^4 - T_o^4}{T_{p1} - T_a} \quad [\text{W/m}^2\text{K}] \]
Sky temperature models

Number of models for sky temperature $T_o$ calculation based on:

- ambient temperature $T_a$
  
  - cloudy sky: $T_o = T_a$
  
  - clear sky: $T_o = 0.0552 (T_a)^{1.5}$
  
  (Swinbank, 1963)

- dew point temperature $T_{dp}$
- water vapour pressure $p_d$
- sky clearness index $K_o$

- clear sky conditions
- cloudy sky conditions

for more see Reference handbook on web
Sky temperature models

Sky temperature calculation based on:

\[ T_o = T_a \]

\[ T_o = 0.0552 \left( T_a \right)^{1.5} \]

\( e_{p1} = 0.85, \; t_{p1} = 40 \, ^\circ\text{C}, \; t_a = 20 \, ^\circ\text{C} \)

\( h_{s,p1-a} = 5.38 \, \text{W/m}^2\text{K} \)

\( t_o = 3.9 \, ^\circ\text{C} \)

\( h_{s,p1-a} = 8.97 \, \text{W/m}^2\text{K} \)

\( e_{p1} = 0.10, \; t_{p1} = 40 \, ^\circ\text{C}, \; t_a = 20 \, ^\circ\text{C} \)

\( h_{s,p1-a} = 0.63 \, \text{W/m}^2\text{K} \)

\( h_{s,p1-a} = 1.06 \, \text{W/m}^2\text{K} \)

considerable decrease of heat transfer coefficient if low-e coating does it mean also improvement of collector performance?
Coupling sky radiation + wind convection

heat transfer from glazing exterior surface: **proportions**

\[ q = q_s \text{ (sky radiation)} + q_p \text{ (wind convection)} \]

<table>
<thead>
<tr>
<th>( w ) [m/s]</th>
<th>( h_s ) [W/m(^2)K]</th>
<th>( h_p ) [W/m(^2)K]</th>
<th>( h_s + h_p ) [W/m(^2)K]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>5.1 / 0.6</td>
<td>5.3</td>
<td>10.4 / 5.9</td>
</tr>
<tr>
<td>2</td>
<td>5.0 / 0.6</td>
<td>12.5</td>
<td>17.5 / 13.1</td>
</tr>
<tr>
<td>4</td>
<td>5.0 / 0.6</td>
<td>19.7</td>
<td>24.7 / 20.3</td>
</tr>
<tr>
<td>6</td>
<td>5.0 / 0.6</td>
<td>26.2</td>
<td>31.2 / 26.8</td>
</tr>
</tbody>
</table>

convection heat transfer is dominant (if wind is present ...)
Influence of sky radiation reduction

\[ \eta (t_m - t_a)/G \text{ [W/m}^2\text{K}] \]

\[ \Delta \eta = 1 \% \]

not considered reduction of solar transmittance !!!
Influence of sky radiation reduction

- Commercially available architectural glass
- New developments for solar collector applications

**Graph:**
- Y-axis: Emissivity $\varepsilon$
- X-axis: Solar transmittance $\tau_e$
- Type 1 and Type 2

**Low-e glass design:**
- Low-e coating system (AZO or ITO)
- Type 1
- Type 2
- AR coating
Conduction through cover glazing

Heat conductance

\[ h_{gl} = h_{v,p1-p2} = \frac{\lambda_{gl}}{L_{gl}} \quad [\text{W/m}^2\text{K}] \]

**Single glazing:**
- **Glass:** \( \lambda_{gl} = 0.8 \text{ W/mK} \), \( L_{gl} = 4 \text{ mm} \)
- \( h_{gl} = 200 \text{ W/m}^2\text{K} \)

**PC:**
- \( \lambda_{gl} = 0.2 \text{ W/mK} \), \( L_{gl} = 4 \text{ mm} \)
- \( h_{gl} = 50 \text{ W/m}^2\text{K} \)

Practically negligible
Conduction through cover glazing

- transparent insulation structures
  - channel structures: $h_{gl} = 2$ to $8 \text{ W/m}^2\text{K}$
  - honeycombs: $h_{gl} = 1$ to $2 \text{ W/m}^2\text{K}$
  - aerogels: $h_{gl} = \text{less than 1 W/m}^2\text{K}$

function of mean temperature
Conduction through cover glazing
Radiation between absorber and cover

Radiation heat exchange

\[ q_{s,\text{abs-p2}} = \sigma \frac{T_{\text{abs}}^4 - T_{\text{p2}}^4}{\frac{1}{\varepsilon_{\text{p2}}} + \frac{1}{\varepsilon_{\text{abs,p}}} - 1} \quad [\text{W/m}^2] \]
Radiation between absorber and cover

influence of absorber emissivity

<table>
<thead>
<tr>
<th>$t_{abs}$</th>
<th>40 °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>$t_{p2}$</td>
<td>20 °C</td>
</tr>
<tr>
<td>$\varepsilon_{abs,p}$</td>
<td>0.85</td>
</tr>
<tr>
<td>$\varepsilon_{p2}$</td>
<td>0.85</td>
</tr>
<tr>
<td>$h_{s,p1-a}$</td>
<td>4.7 W/m²K</td>
</tr>
</tbody>
</table>

Graph showing the ratio of $8x$ and $14x$. 
Spectrally selective absorber

- how can the absorber emissivity influence the thermal performance of the solar collector?
- influence on efficiency curve
- influence on the yields

- are low-emittance coating for absorbers needed?

- compare $\varepsilon_{\text{abs}} = 0.85$  $\varepsilon_{\text{abs}} = 0.10$  $\varepsilon_{\text{abs}} = 0.05$
Spectrally selective absorber

- run KOLEKTOR programme, open `default.kol`
- Absorber card change **Front surface emissivity**
- Calculation card select Efficiency curve calculation
  Calculate, Export results

- open `Evaluation.xls`
- make copies of the sheets for three alternatives
- open res files in excel, mind the semicolons as separators
- compare the efficiency curves based on \( (t_m - t_e)/G \)
Spectrally selective absorber
Spectrally selective absorber

- how can the absorber emissivity influence the annual energy yields of collector?
  - $\varepsilon_{\text{abs}} = 0.05 \quad \eta_0 = 0.789 \quad a_1 = 4.857 \quad a_2 = 0.006$
  - $\varepsilon_{\text{abs}} = 0.10 \quad \eta_0 = 0.785 \quad a_1 = 5.006 \quad a_2 = 0.008$
  - $\varepsilon_{\text{abs}} = 0.85 \quad \eta_0 = 0.744 \quad a_1 = 6.773 \quad a_2 = 0.020$

- open studio SDHW-SOLNET in TRNSYS
- make a change of coefficients in KOLEKTOR type
- results of collector yield find under Totals.txt
- compare the annual sums for both alternatives
Spectrally selective absorber

\[ \varepsilon_{\text{abs}} = 0.05: \quad 1887 \text{ kWh/a} \]

\[ \varepsilon_{\text{abs}} = 0.10: \quad 1858 \text{ kWh/a} \quad -1.5 \% \]

\[ \varepsilon_{\text{abs}} = 0.85: \quad 1590 \text{ kWh/a} \quad -16 \% \]
Spectrally selective absorber

- new **cermet coatings** with extremely low-emittance are about 1.5 % better than old-fashioned **galvanic coatings**
- there could be no fear about nonuniformity of coating emittance at the whole absorber area
  - it does not matter for annual effectivity
  - facts above valid for **domestic hot water systems**
- **significant difference** for high emittance coatings
- **very critical** for high temperature operation – much higher impact on the annual yields
Natural convection in the air gap

coupled convection and conduction heat transfer in closed gas layer

\[ h_{p+v} = \text{Nu}_L \frac{\lambda_g}{L} \] [W/m²K]

**Characteristic dimension:**
- thickness of gas layer \( L \)

**Nusselt number:**
- characterization of conduction heat transfer \( \rightarrow \lambda_g/L \)
- enhancement by fluid convection (driven by buoyancy) \( \rightarrow \text{Nu}_L \)
- definition: ratio of the total heat transfer to conductive heat transfer
Convection in the air gap - criteria

Nusselt number: \( \text{Nu}_L = f (\text{Gr}_L, \text{Pr}) \)

Grashof number: \( \text{Gr}_L = \beta \frac{g L^3 \Delta t}{\nu^2} = \frac{2}{(T_h + T_c)} \cdot \frac{g L^3 (t_h - t_c)}{\nu^2} \)

ratio of the buoyancy to viscous force acting on a fluid

Prandtl number: \( \text{Pr} = \frac{\nu}{\alpha} = \frac{\nu \rho c}{\lambda} \)

express conformity of velocity and temperature fields,
ratio of momentum diffusivity (kinematic viscosity) to thermal diffusivity
Convection in the air gap - criteria

Rayleigh number: \( Ra = Gr \cdot Pr = \frac{\beta g L^3 (t_h - t_c)}{\nu a} \)

ratio of buoyancy forces to thermal and momentum diffusivities of fluid

for standard conditions

\( 400 < Gr < 60000 \)
\( 0.72 < Pr < 0.73 \)
\( 300 < Ra < 44000 \)
Structure of the convection models

Hollands:

\[ \text{Nu}_L = 1 + 1.44 \left[ 1 - \frac{1708}{R\text{a}_L \cos \phi} \right]^+ \left( 1 - \frac{(\sin 1.8\phi)^{1.6} 1708}{R\text{a}_L \cos \phi} \right) + \left( \frac{R\text{a}_L \cos \phi}{5830} \right)^{1/3} - 1 \]

Buchberg:

\[ \text{Nu}_L = 1 + 1.446 \left( 1 - \frac{1708}{R\text{a}_L \cos \phi} \right)^+ \]

\[ \text{Nu}_L = 0.157 (R\text{a}_L \cos \phi)^{0.285} \]

Randall:

\[ \text{Nu}_L = 0.118 \left[ R\text{a}_L \cos^2 (\phi - 45) \right]^{0.29} \]

Niemann:

\[ \text{Nu}_L = 1 + \frac{m (R\text{a}_L)^K}{R\text{a}_L + n} \]

for more see Reference handbook on web
Convection in the air gap – optimum $L$

slope 45°, absorber 60 °C, glass 20 °C

application region

impractical region
Slope dependence

absorber 60 °C, glass 20 °C

Hollands
applicable range 0 to 60°
Slope dependence $0^\circ$

absorber $60^\circ\text{C}$, glass $20^\circ\text{C}$
Slope dependence 45°

absorber 60 °C, glass 20 °C

$h_{p,\text{abs}_p}[\text{W/m}^2\text{K}]$

$L [\text{mm}]$
Slope dependence 90°

absorber 60 °C, glass 20 °C

\[ h_{\text{abs-p}} \text{[W/m}^2\text{K]} \]

\[ L \text{ [mm]} \]

- Randall
- Schinkel
- Niemann
- A1
Convection in inclined air layer

\[ \text{Ra}_L = 10^4 \]

\[ \text{Ra}_L = 10^5 \]

\[ \text{Ra}_L = 10^6 \]
Slope impact on collector efficiency

- Increase of slope angle
- Decrease of front heat loss
- Increase in efficiency

Graph showing efficiency $\eta$ vs. $(t_m - t_a)/G$ [m$^2$.K/W] for different slopes: 15° (green), 45° (red), and 90° (blue).
Slope impact on collector efficiency

results from testing at cca 1000 W/m²
Convection in the back air gap

in the range $90 < \phi < 180^\circ$

$\text{Nu}_L = 1 + [\text{Nu}_L (\phi = 90^\circ) - 1] \sin \phi$

Arnold (1975)
Convection vs. slope

\[ \text{Nu} = 1 \]
Radiation between absorber and frame

Radiation heat exchange

\[ q_{s,\text{abs}-z2} = \sigma \frac{T_{\text{abs}}^4 - T_{z2}^4}{\frac{1}{\varepsilon_{z2}} + \frac{1}{\varepsilon_{\text{abs},z}}} - 1 \]

treatment of insulations:

aluminium foil
applied to mineral wool

\[ \varepsilon_{z2} = 0.1 \]

is it needed?
Radiation between absorber and frame

Example: internal frame insulation surface emittance $\varepsilon_{z2}$

$t_{\text{abs}} = 60 \, ^\circ\text{C}, \ t_a = 20 \, ^\circ\text{C}$, back insulation 30 mm, air gap $d_{z2}$

case 1: $d_{z2} = 20$ mm

$\varepsilon_{z2} = 0.5 \quad U_z = 1.10 \, \text{W/m}^2\text{K}$

$\varepsilon_{z2} = 0.1 \quad U_z = 0.96 \, \text{W/m}^2\text{K}$

15 %

case 2: $d_{z2} = 5$ mm

$\varepsilon_{z2} = 0.5 \quad U_z = 1.22 \, \text{W/m}^2\text{K}$

$\varepsilon_{z2} = 0.1 \quad U_z = 1.17 \, \text{W/m}^2\text{K}$

4 %
Radiation between absorber and frame

back air gap 20 mm

back air gap 5 mm

it has almost no impact if there is aluminium foil or not
Heat conduction through frame insulation

Heat conductance of insulation layer

\[ h_{\text{ins}} = h_{v,z1-z2} = \frac{\lambda_{\text{ins}}}{L_{\text{ins}}} \quad \text{[W/m}^2\text{K]} \]

Insulation – thermal conductivity:

- **Mineral wool:** \( \lambda = 0.045 \text{ W/mK} \)
- **PUR foam:** \( \lambda = 0.035 \text{ W/mK} \)
- **Polystyren:** weak resistance to thermal load

Thermal conductivity \( \lambda = f(t) \)

\[
\lambda_{\text{ins}} = \lambda_0 (1 + 0.0025t) \\
\lambda_{\text{ins}} = \lambda_0 + \lambda_1 \cdot t + \lambda_2 \cdot t^2
\]
Thermal conductivity of mineral wool

polynomic function
Insulation thickness in collector

- How can the thickness of insulation influence the thermal performance of the solar collector?
- Influence on efficiency curve
- Influence on the yields

- What insulation thickness is reasonable?
  
  - Compare 20 mm 30 mm 50 mm
Insulation thickness in collector

- run KOLEKTOR programme, open default.kol
- Glazind & insulation card change Insulation thickness
- Calculation card select Efficiency curve calculation
  - Calculate, Export results

- open Evaluation.xls
- make copies of the sheets for three alternatives
- open res files in excel, mind the semicolons as separators
- compare the efficiency curves based on \((t_m - t_e)/G\)
Insulation thickness in collector

![Graph showing efficiency versus (t_m - t_e)/G (m2K/W)]
Insulation thickness in collector

- how can the absorber emissivity influence the annual energy yields of collector?
  - 20 mm \( \eta_0 = 0.782 \) \( a_1 = 5.272 \) \( a_2 = 0.007 \)
  - 30 mm \( \eta_0 = 0.789 \) \( a_1 = 4.857 \) \( a_2 = 0.006 \)
  - 50 mm \( \eta_0 = 0.796 \) \( a_1 = 4.451 \) \( a_2 = 0.006 \)

- open studio SDHW-SOLNET in TRNSYS
- make a change of coefficients in KOLEKTOR type
- results of collector yield find under Totals.txt
- compare the annual sums for both alternatives
Insulation thickness in collector

20 mm: 1887 kWh/a  -3 %
30 mm: 1858 kWh/a
50 mm: 1950 kWh/a  +3 %
Insulation thickness in collector

- compared to 30 mm of mineral wool double thickness of insulation brings only several percents
- **80 to 90 %** of the collector heat loss is the front side (top) loss
  - increase of insulation on the back side cannot proportionally help

- change of thermal conductivity by tens of percents due degradation will not significantly change the energy yields (by percents)
  - it does not matter for annual effectivity
  - facts above valid for **domestic hot water systems**
Radiation between frame and adj. surface

Radiation heat exchange between frame of collector and roof

\[ q_{s,z1-as} = \sigma \frac{T_{z1}^4 - T_{as}^4}{1 + \frac{1}{\varepsilon_{z1}} - 1} \]  [W/m²]

any treatment of external frame surface is useless

wind convection supress any influence

frame insulation
Collector heat loss coefficient: $U$-value

**front side ($U_p$)**

$$U_p = \frac{1}{h_{s,p1-a} + h_{p,p1-a}} + \frac{1}{h_{v,p1-p2}} + \frac{1}{h_{s,abs-p2} + h_{p,abs-p2}}$$

**back side ($U_z$)**

$$U_z = \frac{1}{h_{s,z1-a} + h_{p,z1-a}} + \frac{1}{h_{v,z1-z2}} + \frac{1}{h_{s,abs-z2} + h_{p,abs-z2}}$$
Collector heat loss coefficient: U-value

\[ \dot{Q}_{l,t} = U_G A_G (t_{abs} - t_a) \]

\[ \dot{Q}_{l,t} = U_a A_a (t_{abs} - t_a) \]

\[ U_G A_G = U_p A_G + U_z A_G + U_b A_b = U_a A_a \]

\[ U_a = \left( U_p + U_z + U_z \frac{A_b}{A_G} \right) \frac{A_G}{A_a} \]

for calculations:

\textbf{U-value} related to aperture area \( A_a \)
Iterations for temperatures

calculation of heat transfer coefficients:

temperatures $t_{\text{abs}}, t_{p1}, t_{p2}, t_{z1}, t_{z2}, (t_{b1}, t_{b2})$ are required
but not available at the start of calculation

example of iterative determination for front heat flow:

1. estimate: $t_{\text{abs}} = t_{\text{in}} + \Delta t$

2. estimate: $t_{p1}, t_{p2}$
   
   \[ t_{p2} = t_{\text{abs}} - \frac{t_{\text{abs}} - t_a}{3} \]
   
   \[ t_{p1} = t_a + \frac{t_{\text{abs}} - t_a}{3} \]

3. calculation of heat transfer coefficients $h$

4. calculation of overall heat flow rate $q_p$

5. reverse calculation of temperatures and then repeat from (3.)
Iterations for temperatures

\[
q_p = U_p (t_{abs} - t_a)
\]

\[
q_p = (h_{s,abs-p1} + h_{p,abs-p1})(t_{abs} - t_{p1})
\]

\[
q_p = h_{v,p1-p2}(t_{p1} - t_{p2})
\]

\[
q_p = (h_{s,p2-a} + h_{p,p2-a})(t_{p2} - t_a)
\]

temperature difference is proportional to heat resistance
1. estimation of temperatures
Last iteration
Convergence of calculations

absorber temperature 35 °C

ambient temperature 25 °C
Results of external balance

*U*-value of solar collector dependent on:

- temperature of absorber
- ambient temperature, wind velocity, sky temperature
- geometry of solar collector
- detailed properties of collector elements, used material (conductivity), surface emittance
- adjacent structure properties (emittance, envelope thermal resistance)
Analysis of Solar Flat-plate Collectors – part 2

Tomáš Matuška
Faculty of Mechanical Engineering
University Centre of Energy Efficient Buildings
Czech Technical University in Prague

Solar Thermal Collectors, 24.3.2015, Palma de Mallorca
Solar collector energy balance

- **External energy balance of absorber**
  - heat flow from absorber surface to ambient environment
  - heat losses
  - quality of solar collector envelope

- **Internal energy balance of absorber**
  - heat flow from absorber surface into heat transfer fluid
  - ability to transfer heat and remove it from collector
  - quality of absorber construction
Internal energy balance
Energy balance of absorber fin

width $W = \text{distance between riser pipes}$

$D_e = 2a = 2 \cdot \text{bond width}$

fin heat transfer $\Lambda_{\text{fin}}$

bond heat transfer $\Lambda_b$

pipe-fluid heat transfer $h_i \pi D_i$

bond temperature $t_b = t_w$ (wall temperature)
Energy balance of absorber fin

solar irradiation

base temperature $t_0$

insulated

$\Delta x$

$U$

$d_{abs}$

$T_a$

$\frac{(W - D_e)}{2}$

case of standard fin efficiency concept but with solar irradiation known from air-water heat exchangers theory
Energy balance of absorber fin element

\[ U \Delta x (T - T_a) - \frac{\lambda_{\text{abs}} d_{\text{abs}}}{d x} \frac{dT}{dx} \bigg|_x - \frac{\lambda_{\text{abs}} d_{\text{abs}}}{d x} \frac{dT}{dx} \bigg|_{x + \Delta x} = \tau \alpha G \Delta x \]

derivation of temperature distribution between tubes (in absorber fin)

\[ \tau \alpha G \Delta x - U \Delta x (T - T_a) + \left( - \lambda_{\text{abs}} d_{\text{abs}} \frac{dT}{dx} \right) \bigg|_x - \left( - \lambda_{\text{abs}} d_{\text{abs}} \frac{dT}{dx} \right) \bigg|_{x + \Delta x} = 0 \]

yields in solution of differential equation of 2nd order

\[ \frac{d^2 T}{d x^2} = \frac{U}{\lambda_{\text{abs}} d_{\text{abs}}} \left( T - T_a - \frac{S}{U} \right) \]
Fin efficiency $F$

**standard fin efficiency**
(rectangular profile)

$$ F = \frac{\tanh[m(W - D_e) / 2]}{m(W - D_e) / 2} $$

$(W - D_e)/2$ or $(W - 2a)/2$... active fin length

soldering

ultrasonic welding
Fin efficiency $F$

standard fin efficiency (rectangular profile)

$$F = \frac{\tanh[m(W-D_e)/2]}{m(W-D_e)/2}$$

$$m = \sqrt{\frac{U}{\lambda_{abs}d_{abs}}}$$

$U$ ... collector heat loss considered from absorber surface to ambient (result from external balance)

significant influence of thermal conductivity and thickness of absorber conductivity and thickness of absorber are impropotional

higher fin efficiency = higher heat removal from absorber
Fin efficiency $F$

**reference case:** fin efficiency 0.96

copper absorber (390 W/mK), $d_{abs} = 0.2$ mm, fin $W^* = W - 2a = 100$ mm

heat loss coefficient $U = 4$ W/m²K

---

aluminium (240 W/mK)

$d_{abs} = 0.20$ mm, $W^* = 79$ mm

$d_{abs} = 0.32$ mm, $W^* = 100$ mm

steel (80 W/mK)

$d_{abs} = 0.2$ mm, $W^* = 45$ mm

$d_{abs} = 1.0$ mm, $W^* = 100$ mm

EPDM (0.14 W/mK) $d_{abs} = 2.0$ mm, $W^* = 6$ mm
Plastic absorbers
Best absorbers – fully wetted metal sheet

\[ F' = 1.0 \]
Influence of material and geometry

- Copper (Cu): 390 W/(m.K)
- Aluminium (Al): 250 W/(m.K)
- Steel (Fe): 100 W/(m.K)

Graphs showing the relationship between \((t_m - t_e)/G\) and \(\eta\) for different materials and geometries:

- \(W = 50\) mm
- \(W = 125\) mm
- \(W = 200\) mm
Efficiency factor $F'$

how efficient is heat transfer from absorber surface to heat transfer fluid?

$$F' = \frac{U_o}{U} = \frac{\text{liquid to ambient loss}}{\text{abs. surface to ambient loss}}$$

or

$$F' = \frac{\dot{q}_c(t_{abs})}{\dot{q}_c(t_{abs} = t_m)}$$

heat resistances:
- conduction through fin
- conduction through bond
- convection inside pipe
Collector performance based on $t_m$

useful heat gain from collector

$$\dot{Q}_u = AF' [\tau \alpha G - U(t_m - t_a)]$$

$$t_m = \frac{t_{in} + t_{out}}{2}$$

collector efficiency

$$\eta = F' \left[ \tau \alpha - U \frac{t_m - t_a}{G} \right]$$

analogy to experimentally obtained efficiency curve reported by testing institutes

$$\eta = \eta_0 - a_1 \frac{t_m - t_a}{G} - a_2 \frac{(t_m - t_a)^2}{G}$$
Efficiency factor $F'$

\[ F' = \frac{1}{W} \left[ \frac{1}{U} \left( \frac{1}{C_b} \frac{1}{h_{fi} \pi D_i} \right) \right] \]

function of fin efficiency $F$
Efficiency factor – absorber designs

\[
F' = \frac{1}{U} \left[ \frac{1}{WU} \left( \frac{1}{h_f \pi D_i} + \frac{2a}{W} \frac{1}{C_b} + \frac{1}{U[2a+(W-2a)F]} \right) + \frac{1}{U[2a+(W-2a)F]} \right]
\]
Bond conductance

bond conductance – estimated from geometry and quality of contact

\[ C_b = \frac{\lambda_b a}{b} \]

good metal-to-metal contact needed (welding, soldering, pressing) no clamping of absorber to pipes!

bond conductance > 30 W/mK needed
For $C_b < 30 \text{ W/mK}$ significant reduction of collector performance

- Practically no difference between high and middle quality bonds

- $brazed\ contact$

- $ultrasonic\ welded\ contact$
Bond conductance: testing

\[ \eta = \frac{(t_m - t_a)}{G} \]

- clamped absorber
- touching absorber
- welded absorber

Touched by press

Clamped bond
forced convection heat transfer in pipes is given by:

- heat transfer fluid type
- collector flow rate
- concept of collector hydraulic (parallel, serial-parallel, serpentine)
- number of risers (flow rate distribution)
- diameter of risers
- temperature
Convection heat transfer inside pipes

Nusselt number as criterion, number of models are available for

- **laminar flow** (mostly present in collector), turbulent flow
- **constant heat flux case** (uniformly irradiated absorber condition)
- constant temperature (heat transfer with phase change)
- fully developed profile, entry region with developing profile of velocity and temperature

\[
h_{fi} = \text{Nu}_D \cdot \frac{\lambda_f}{D_i}
\]
Nusselt number in circular pipes

- **laminar flow**, fully developed velocity and temperature profile, constant heat flux

\[ \text{Nu}_D = \frac{48}{11} = 4.364 \quad \text{(Shah)} \]

- **laminar flow**, entry region of length \( L \), developing profile, constant heat flux

\[ \text{Nu}_D = \begin{cases} 
1.953 \cdot L^{*-1/3} & \text{if } L^{*} \leq 0.03 \\
4.364 + \frac{0.0722}{L^{*}} & \text{if } L^{*} > 0.03
\end{cases} \quad \text{(Shah)} \]

\[ L^{*} = \text{Gz}^{-1} = \frac{L / D_i}{\text{Re}_D \cdot \text{Pr}} \quad \text{Gz .. Graetz number} \]
Nusselt number in circular pipes

- **turbulent flow**

\[ \text{Nu}_D = A \, \text{Re}_D^m \, \text{Pr}^n \]  
(Dittus-Boelter, Colburn, Sieder-Tate)

\[ \text{Nu}_D = \frac{(f/8) (\text{Re}_D - 1000) \, \text{Pr}}{1 + 12.7 \, (f/8)^{1/2} \left(\frac{\text{Pr}^{2/3}}{\text{Re}_D^{1/2}} - 1\right)} \]  
(Gnielinski)

where \( f = \left(0.79 \ln \text{Re}_D - 1.64\right)^{-2} \) for smooth pipes

for more models see Reference handbook to KOLEKTOR
Convection heat transfer inside pipes

- high heat transfer require
  - high velocity
  - higher flow rate
  - low dimension
  - low viscosity
  - high conductivity of liquid

high Reynolds number

high Nusselt number
Convection heat transfer inside pipes

$h_{fi} = 200 \text{ to } 400 \text{ W/m}^2\text{K}$

high flow: 0.025 kg/s.m$^2$
low flow: 0.005 kg/s.m$^2$
Influence of riser pipe diameter

\[ F' = \frac{1}{U} \left( \frac{1}{W[U(2a + (W - 2a)F] + \frac{1}{C_b} + \frac{1}{h_i \pi D_i]} \right) \]

\[ \frac{1}{h_i \pi D_i} = \frac{D_i}{\text{Nu}_f \pi D_i} \frac{1}{\text{Nu}_f \pi} \]

term is dependent directly on \( \text{Nu} \)

laminar flow

\( \text{Nu}_D = f(\text{constant}) = 4.364 \)

\( \text{Nu}_D = f(L^*) \)

thermal diffusivity

\[ L^* = \frac{L}{D_i u D_i} = \frac{L \alpha \pi D_i^2}{D_i^2 4V} = \frac{\pi L \alpha}{4V} \]

term \( \frac{1}{h_i \pi D_i} \) is independent of \( D \) in laminar flow
Analysis of efficiency factor $F'$

length $L = 2$ m, fin width $W = 100$ mm, absorber thickness $d = 0.2$ mm, copper $\lambda = 350$ W/m.K, pipe $D_e/D_i = 10/8$ mm

bond conductance $C_b = 250$ W/K

$h_i = 400$ W/m$^2$K

| $U = 4$ W/m$^2$.K | 2,6 | 0,004 | 0,099 |

heat transfer coefficient: **minor influence** to $F'$  
if $h_i > 200$ W/m$^2$K

bond conductance: **minor influence** to $F'$  
if $C_b > 30$ W/mK

geometry is principal property of absorber  
if low heat loss collector
Do we need for turbulators in pipes?

length $L = 2 \text{ m}$, fin width $W = 110 \text{ mm}$, absorber thickness $d = 0.2 \text{ mm}$, copper $\lambda = 350 \text{ W/m.K}$, pipe $D_e/D_i = 10/8 \text{ mm}$

bond conductance $C_b = 200 \text{ W/K}$

$U = 4 \text{ W/m}^2\text{K}$

<table>
<thead>
<tr>
<th>$h_i$</th>
<th>$\frac{1}{h_i \cdot \pi \cdot D_i}$</th>
<th>$F'$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$h_i = 400 \text{ W/m}^2\text{K}$</td>
<td>0.099</td>
<td>0.919</td>
</tr>
<tr>
<td>$h_i = 600 \text{ W/m}^2\text{K}$</td>
<td>0.066</td>
<td>0.931</td>
</tr>
</tbody>
</table>

enhancement of heat transfer inside pipe by **50 %**

**minor influence** to $F'$ change by **1.3 %**
Fully wetted absorber vs. fin&tube

- how can the geometry of the absorber influence the thermal performance of the solar collector?
- influence on efficiency curve
- influence on the yields

- fully wetted absorber from steel
  - type of bond middle, pipes $D_e/D_i = 6/5$ mm, bond $a/b = 3/1$ mm
  - material steel, thickness 0.5 mm
  - bond conductivity 100 W/mK
  - number of channels 145 pcs (not 150 pcs) ... $W = 6.01$ mm $> D_e$
Fully wetted absorber vs. fin&tube

- run KOLEKTOR programme, open default.kol
- Absorber card make the changes **Material, Type of bond**
  change Geometry, dimensions
- Calculation card select Efficiency curve calculation
  Calculate, Export results
- open Evaluation.xls
- make a copy for the wetted alternative
- open res file in excel, mind the semicolons as separators
- compare the efficiency curves based on \((t_m - t_e)/G\)
Fully wetted absorber vs. fin&tube
Fully wetted absorber vs. fin&tube

- how can the absorber geometry influence the annual energy yields of collector?
  - fin&tube \( \eta_0 = 0.789 \quad a_1 = 4.857 \quad a_2 = 0.006 \)
  - fully wetted \( \eta_0 = 0.877 \quad a_1 = 5.371 \quad a_2 = 0.007 \)

- open studio SDHW-SOLNET in TRNSYS
- make a change of coefficients in KOLEKTOR type
- results of collector yield find under Totals.txt
- compare the annual sums for both alternatives
Fully wetted absorber vs. fin&tube

fin & tube: 1887 kWh/a
fully wetted: 1999 kWh/a (+6%)
Fully wetted absorber vs. fin & tube

- fully wetted absorber could improve the annual energy performance by more than 5% while steel could be cheaper material than copper
Fully wetted absorber vs. fin&tube

- did we make a mistake in calculation? can geometry influence losses?

both collectors have same losses!
both collectors have same $U$-value!

coefficients $\eta_0$, $a_1$, $a_2$ are one complex and cannot be separated

Fully wetted
$\eta_0 = 0.877$
$a_1 = 5.371$
$a_2 = 0.007$

fin&tube
$\eta_0 = 0.789$
$a_1 = 4.857$
$a_2 = 0.006$

$F' = 0.996$
$F' = 0.901$

$U = 5.39$ W/m$^2$K for given $\Delta T$
Vacuum tube collectors x efficiency factor

**Single glass vacuum tube / flat absorber**

- direct flow
- concentric direct flow
- heat pipe

firm metal contact absorber-pipe provides high $F'$
Vacuum tube collectors x efficiency factor

Double-glass (Sydney) vacuum tube

- all glass concentric tube (Dewar)
  - cylindric absorber (internal) glass tube
  - cover (external) glass tube
  - vacuum in space between
- absorber coating applied on exterior surface of internal glass tube
- conductive heat transfer fin in contact with interior surface

problematic contact absorber-pipe ...
Sydney vacuum tube collectors
Thermal analysis of direct flow VTC

given by Sydney tube collector scheme of heat transfer

\[ \eta = F' \left[ \tau \alpha - U \frac{(t_m - t_a)}{G} \right] \]

contact fin: short (W), conductive (\lambda), thick (d)

bond-contact: conductive; absorber tube-fin, fin-pipes

heat removal: laminar / turbulent flow in pipes
Influence of contact fin

Sydney vacuum tube collectors with direct flow U-pipe register

$G > 700 \text{ W/m}^2$

$\frac{(t_m - t_e)}{G} \text{ [m}^2\cdot\text{K/W]}$

Contact fin is a principle element in Sydney collector
Contact resistance in heat pipe VTC

**Heat pipe** (phase change of working liquid, high heat transfer coefficients)

- evaporator (in contact with fin transferring the heat from inner glass tube)
- condenser (placed into manifold box)
Contact resistance in heat pipe VTC

\[ \eta (t_m - t_e) / G [m^2 \cdot K/W] \]

- dry contact of condenser in manifold box
- heat conductive paste application

15-20 %
Temperature distribution in flow direction

Derivation of temperature distribution between tubes (in absorber fin)

\[
\left( \frac{\dot{M}}{n} \right) c T_f \big|_Y - \left( \frac{\dot{M}}{n} \right) c T_f \big|_{Y+\Delta Y} + \Delta y q_u' = 0
\]

Yields in solution of differential equation of 1rst order
Heat removal factor $F_R$

Mathematical derivation of temperature profile: see Duffie, Beckman (2006)

Definition: relates the actual collector gain to gain if absorber surface at fluid inlet temperature

$$F_R = \frac{\dot{M}c(t_{out} - t_{in})}{A_c[\tau\alpha G - U(t_{in} - t_a)]}$$

$$F_R = \frac{\dot{M}c}{A_cU} \left[ 1 - \exp\left(-\frac{A_cUF'}{\dot{M}c}\right) \right]$$

Equivalent to effectiveness of heat exchanger, ratio of actual heat transfer to maximum possible heat transfer
Heat removal factor $F_R$

Heat removal factor is dependent on:

- efficiency factor $F'$ (geometry, quality of absorber, fin efficiency $F$, ...)
- collector $U$-value (heat losses)
- specific heat of fluid $c$ (type of fluid)
- specific mass flow through collector $M / A_c$ (thermal capacity of flow)
Water vs. propylene glycol

- how can the heat transfer fluid influence the thermal performance of the solar collector?
- influence on efficiency curve – collectors are tested with water, does anything change when filled with glycol?

- propylene glycol / water
  - mixture 50 % / 50 % for freezing point -32 °C
  - 8x viscous at 20 °C, high influence of temperature, only 2x at 80 °C
  - lower specific heat by cca 25 %
  - worse heat transfer
Water vs. propylene glycol

- run KOLEKTOR programme, open default.kol
- Absorber card make the change of fluid
- Calculation card select Efficiency curve calculation
  Calculate, Export results
- open Evaluation.xls
- make a copy for glycol alternative
- open res file in excel, mind the semicolons as separators
- compare the efficiency curves based on \( (t_m - t_e)/G \)
Water vs. propylene glycol

![Graph comparing water and propylene glycol efficiency against \((t_m - t_e)/G\) in m²K/W.](image)
Efficiency curve

\[ \eta = f \left( \frac{t - t_a}{G} \right) \]

\[ \eta = f \left( t_{in} \right) \]

\[ \eta = f \left( t_m \right) \]

\[ \eta = f \left( t_{abs} \right) \]
Useful heat output of solar collector

based on absorber temperature, external balance of absorber

\[ \dot{Q}_u = A_c [\tau \alpha G - U(t_{abs} - t_a)] \]

based on mean fluid temperature, experimental work

\[ \dot{Q}_u = F' A_c [\tau \alpha G - U(t_m - t_a)] \]

based on fluid inlet temperature, simulation tools

\[ \dot{Q}_u = F_R A_c [\tau \alpha G - U(t_{in} - t_a)] \]
Solar collector temperatures

mean absorber temperature

\[ t_{abs} = t_{in} + \frac{\dot{Q}_u}{A_c} \left( \frac{1 - F_R}{F_R U} \right) \]

mean fluid temperature

\[ t_m = t_{in} + \frac{\dot{Q}_u}{A_c} \left( \frac{1 - F_R}{F_R U} \right) \]

based on fluid inlet temperature

fluid output temperature

\[ t_{out} = 2t_m - t_{in} \]
Solar collector efficiency

based on absorber temperature

$$\eta = \tau \alpha - U \frac{(t_{abs} - t_a)}{G}$$

based on mean fluid temperature

$$\eta = F' \left[ \tau \alpha - U \frac{(t_m - t_a)}{G} \right]$$

based on fluid inlet temperature

$$\eta = F_R \left[ \tau \alpha - U \frac{(t_{in} - t_a)}{G} \right]$$
Solar collector performance calculation

calculation model

climatic data
optical properties

operation parameters

material properties
geometry

external balance
internal balance

superior iteration cycle

\[ Q_u \]
\[ \eta \]
\[ t_{out} \]
Conclusions of the day

- KOLEKTOR is a design tool for virtual prototyping
  - to perform calculations of designs without need for fabrication prototypes
  - to optimize construction and design of collector

- KOLEKTOR is not perfect, but relevant TRNSYS type is on the way
  - for modelling solar thermal collectors (Slava Shemelin)
  - extension for glazed PVT collectors (Nikola Pokorný)

- Efforts for extremely high performing solar thermal collectors could be sometimes useless if target is annual performance and not marketing