Analysis of Solar Flat-plate Collectors

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Content of lectures today

- design tool KOLEKTOR 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 KOLEKTOR 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 β

Outputs:

- efficiency curve η (based on aperture area A_a)
- output temperature from solar collector t_{out}
- detailed temperature distribution in collector, heat transfer coefficients

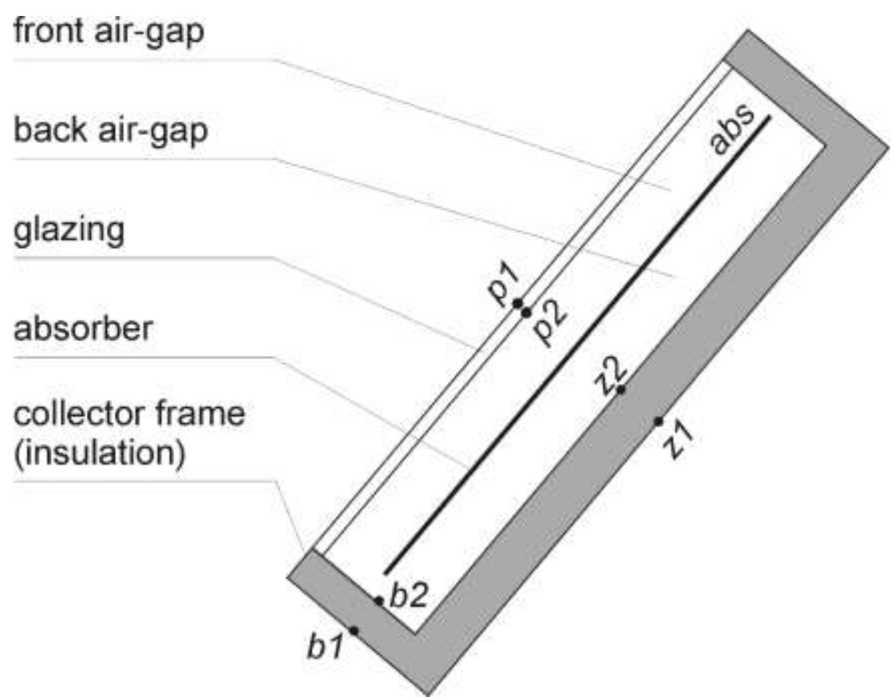
Principle of calculation

- 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

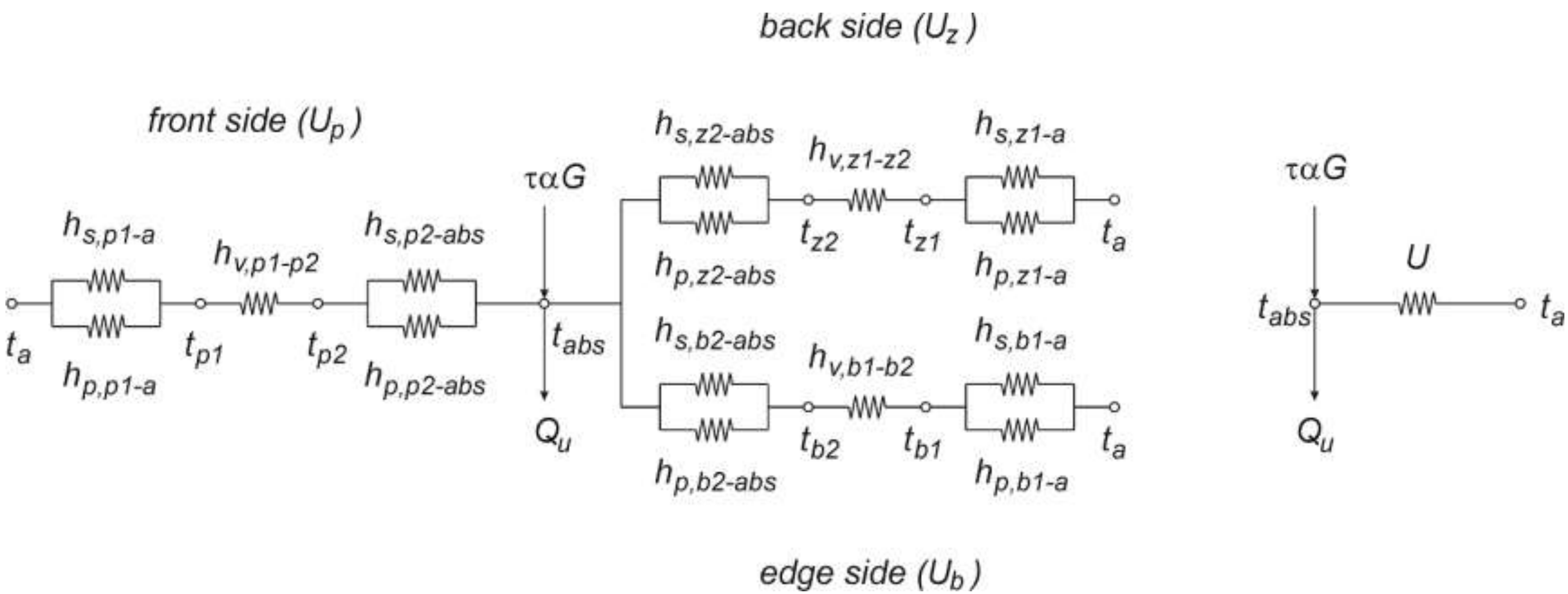
  heat removal factor F_R

  absorber temperature t_{abs}

  heat gain Q_u

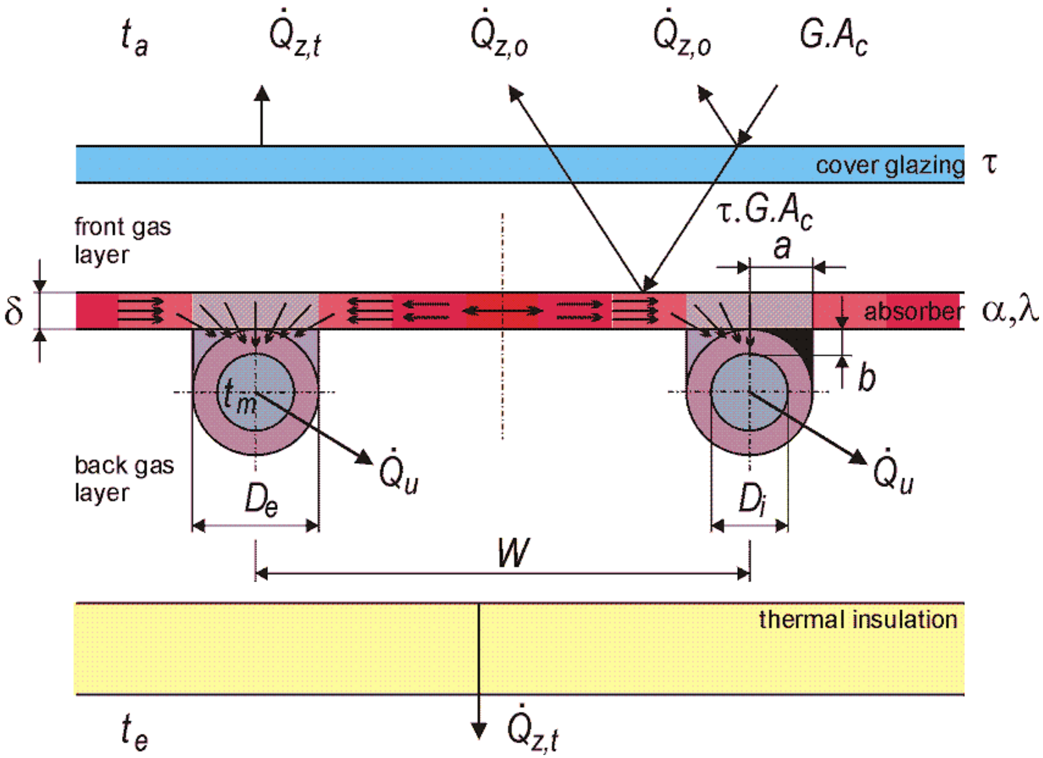
$$\dot{Q}_u = A_a F_R \left[\tau\alpha G - U\left(t_{in} - t_a\right)\right]$$

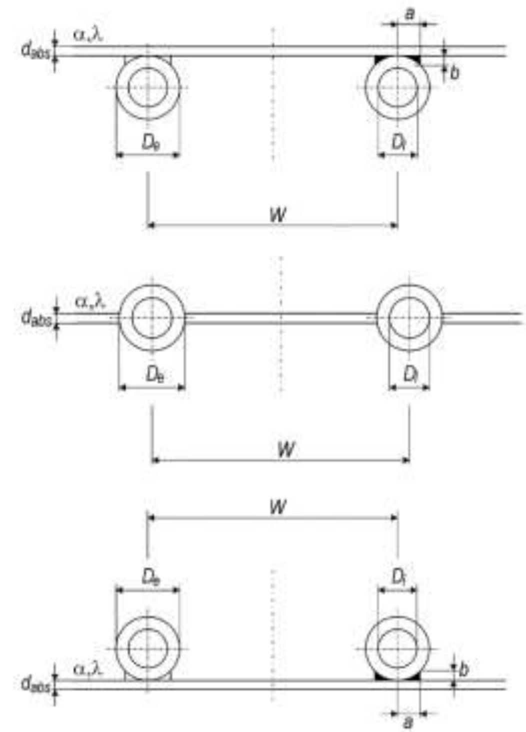
External balance of absorber



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

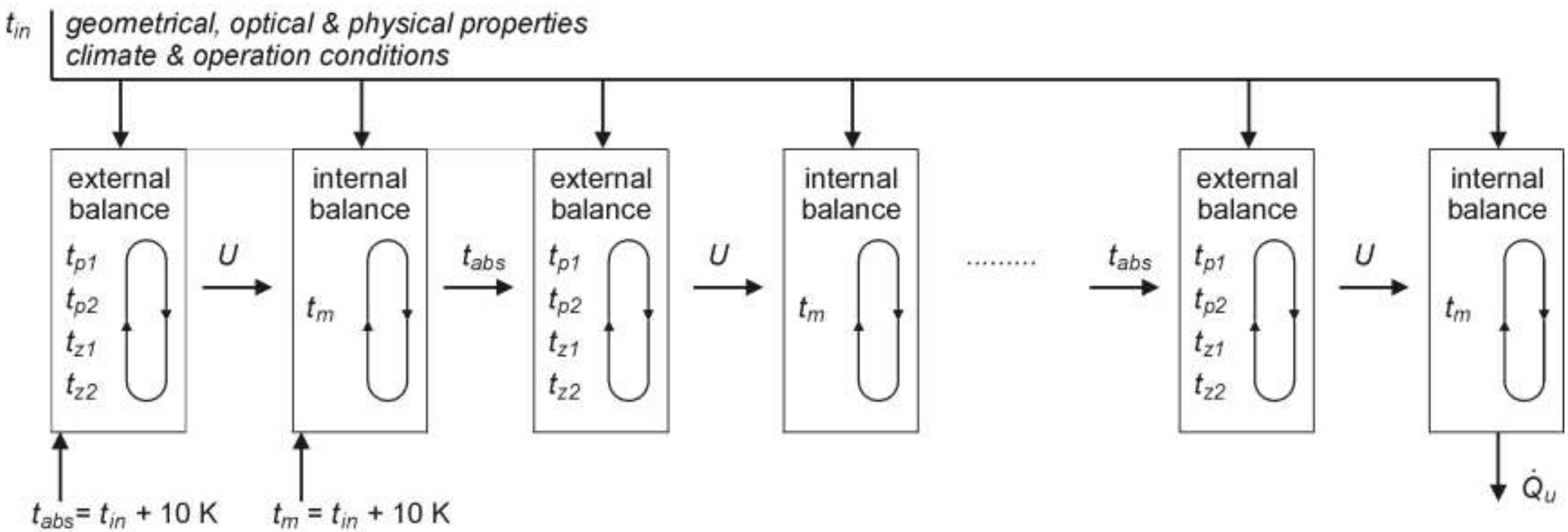
Internal balance of absorber





- heat transfer by fin conduction, bond conduction and forced convection in pipes
- iteration loop for mean fluid temperature (starting values based on $\boldsymbol{t_{in}}$, 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 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

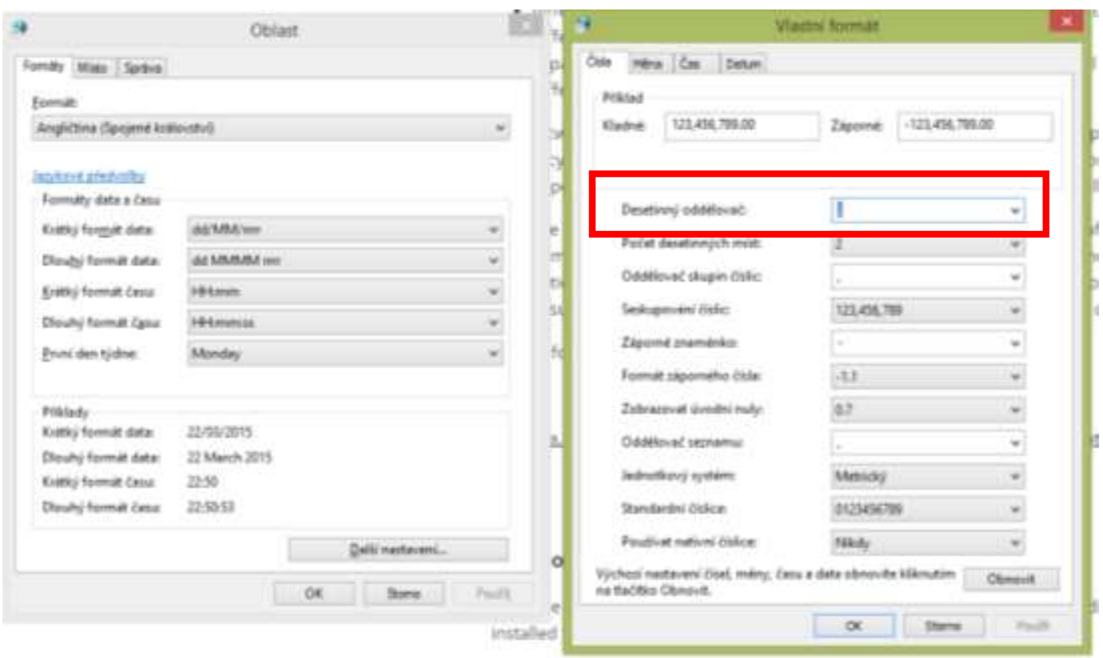
- user-friendly software KOLEKTOR 2.2 is useful for analysis, design and prototyping
- several experimental validations have been performed based on commercial collectors testing
- software tool KOLEKTOR 2.2 is freely available at:

  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 ...

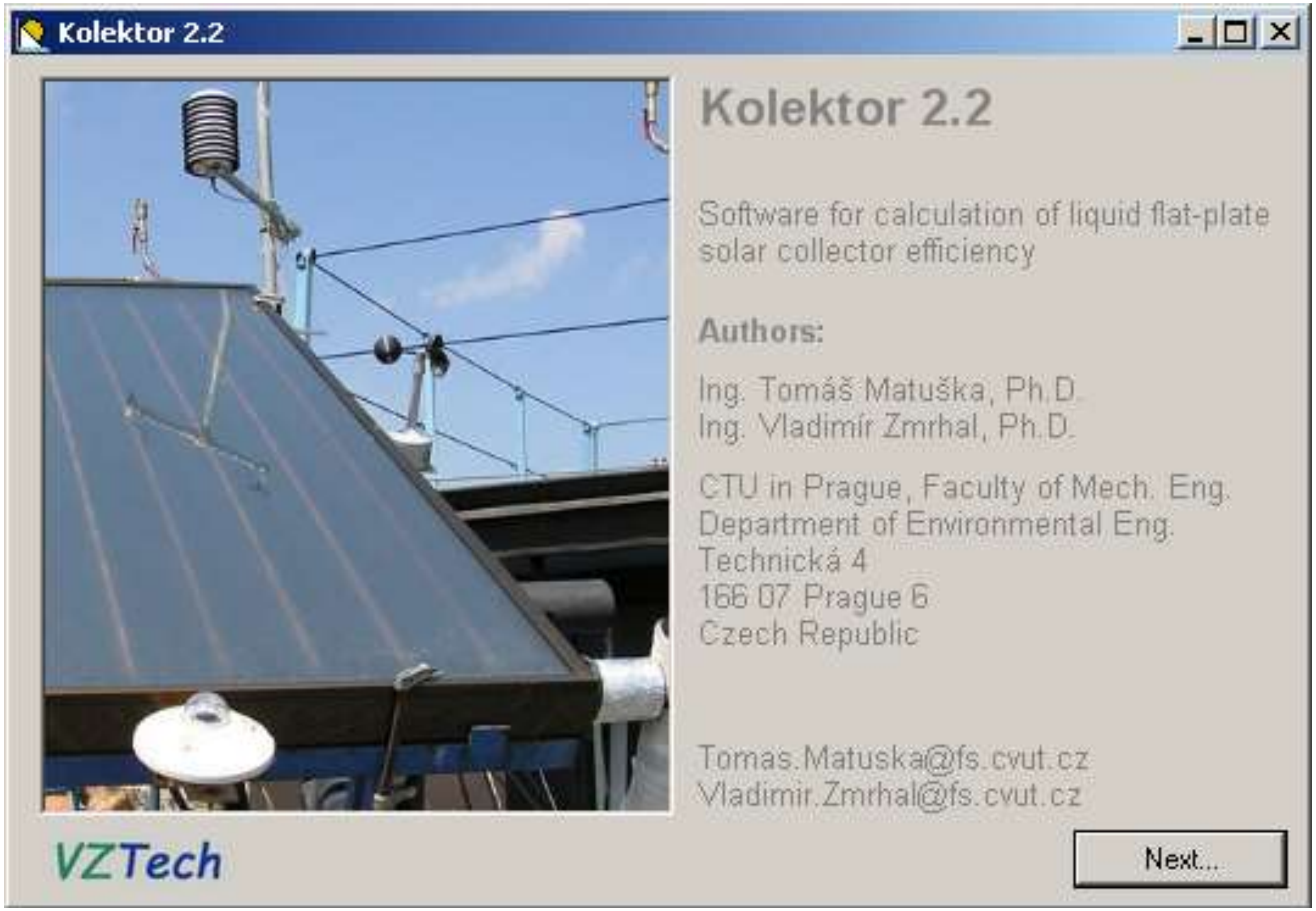
Download KOLEKTOR and install now ...

- http://users.fs.cvut.cz/tomas.matuska/?page_id=194

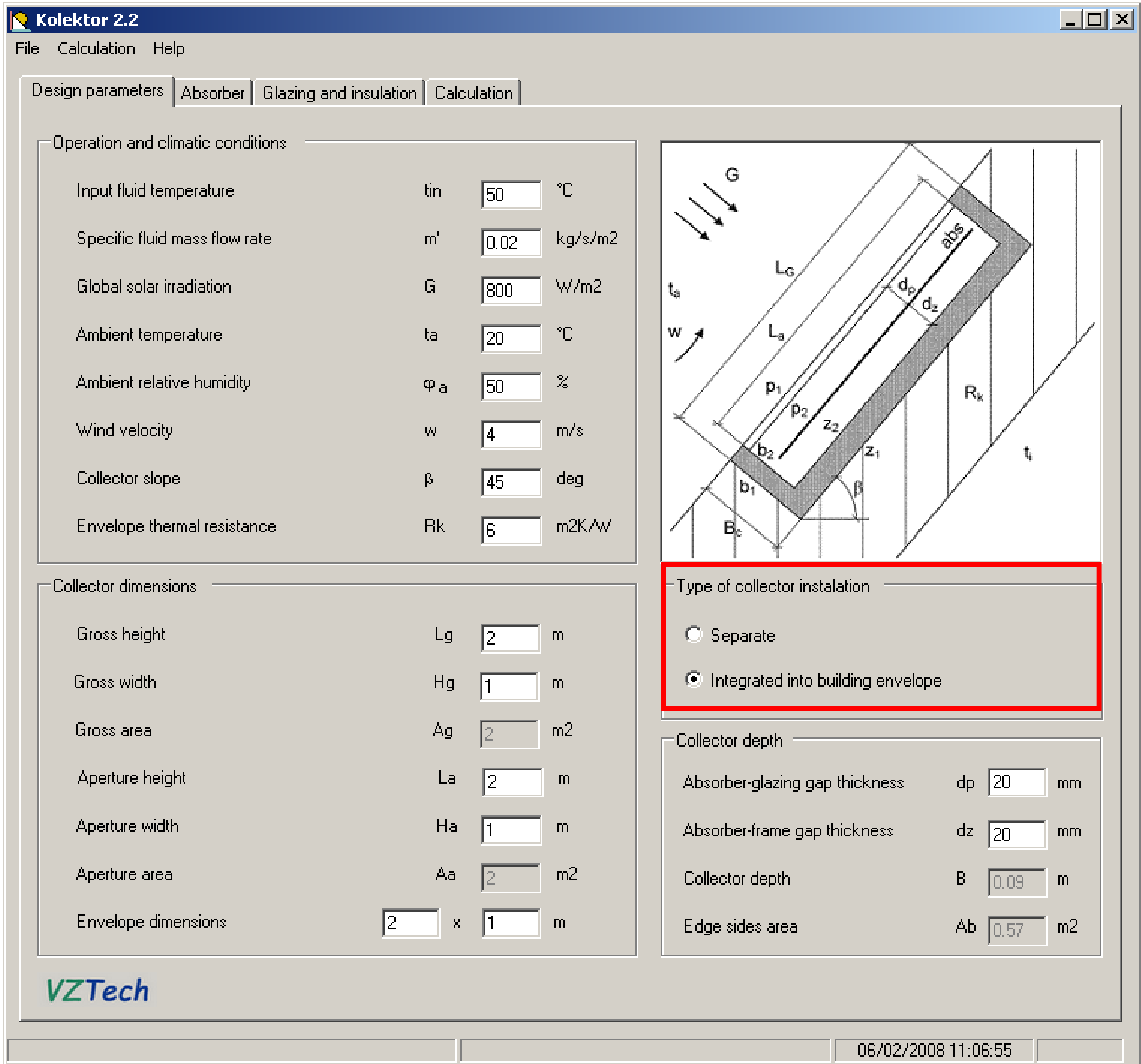
- download **kolektor22.zip**
- unzip it Program files – folder Kolektor will be created
- start setup.exe – start installation process
- setup also opens the program

- for Windows versions higher than XP you don’t need **Microsoft.NET Framework** should be included in ...

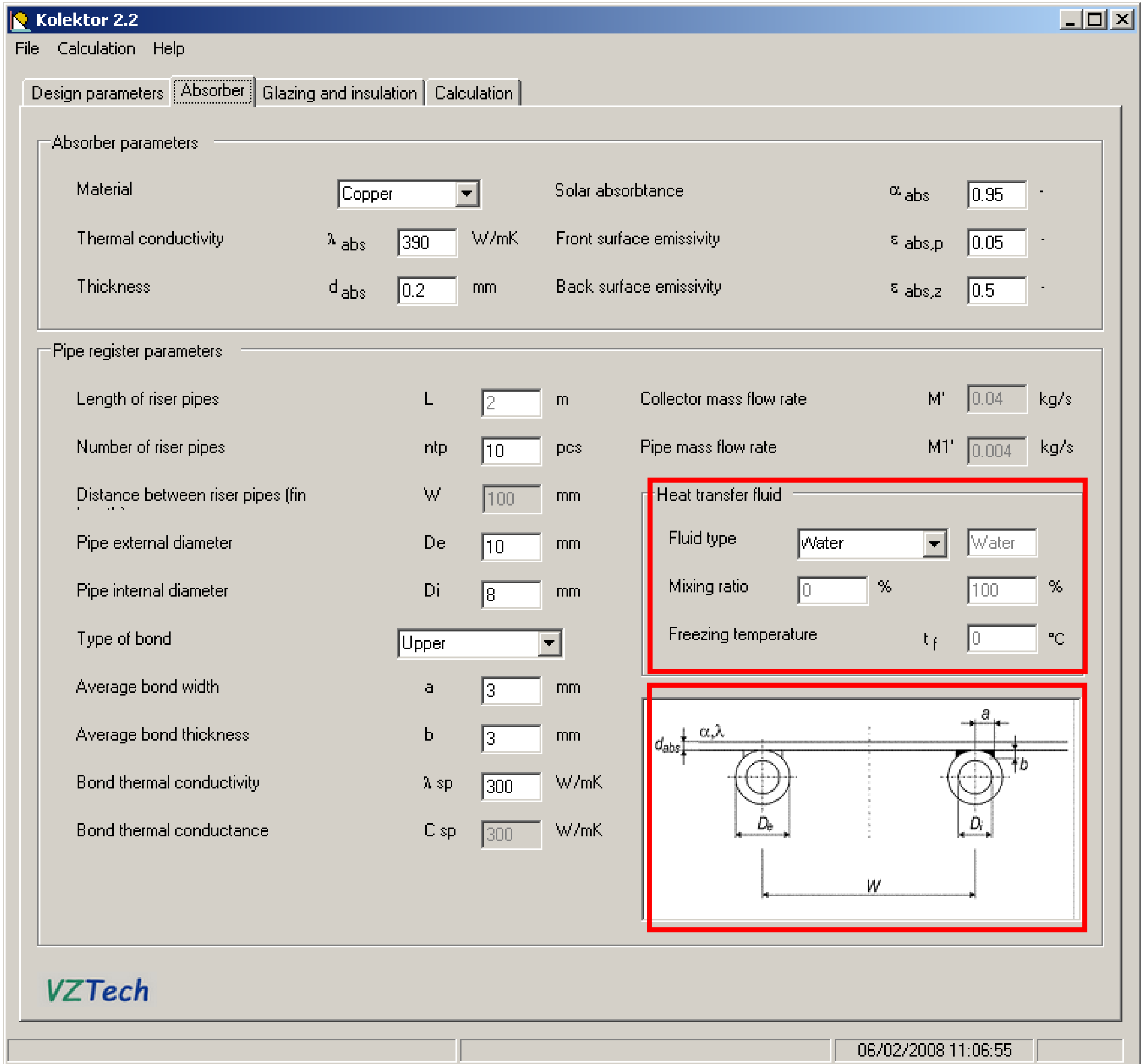
Design tool KOLEKTOR 2.2 (2009)



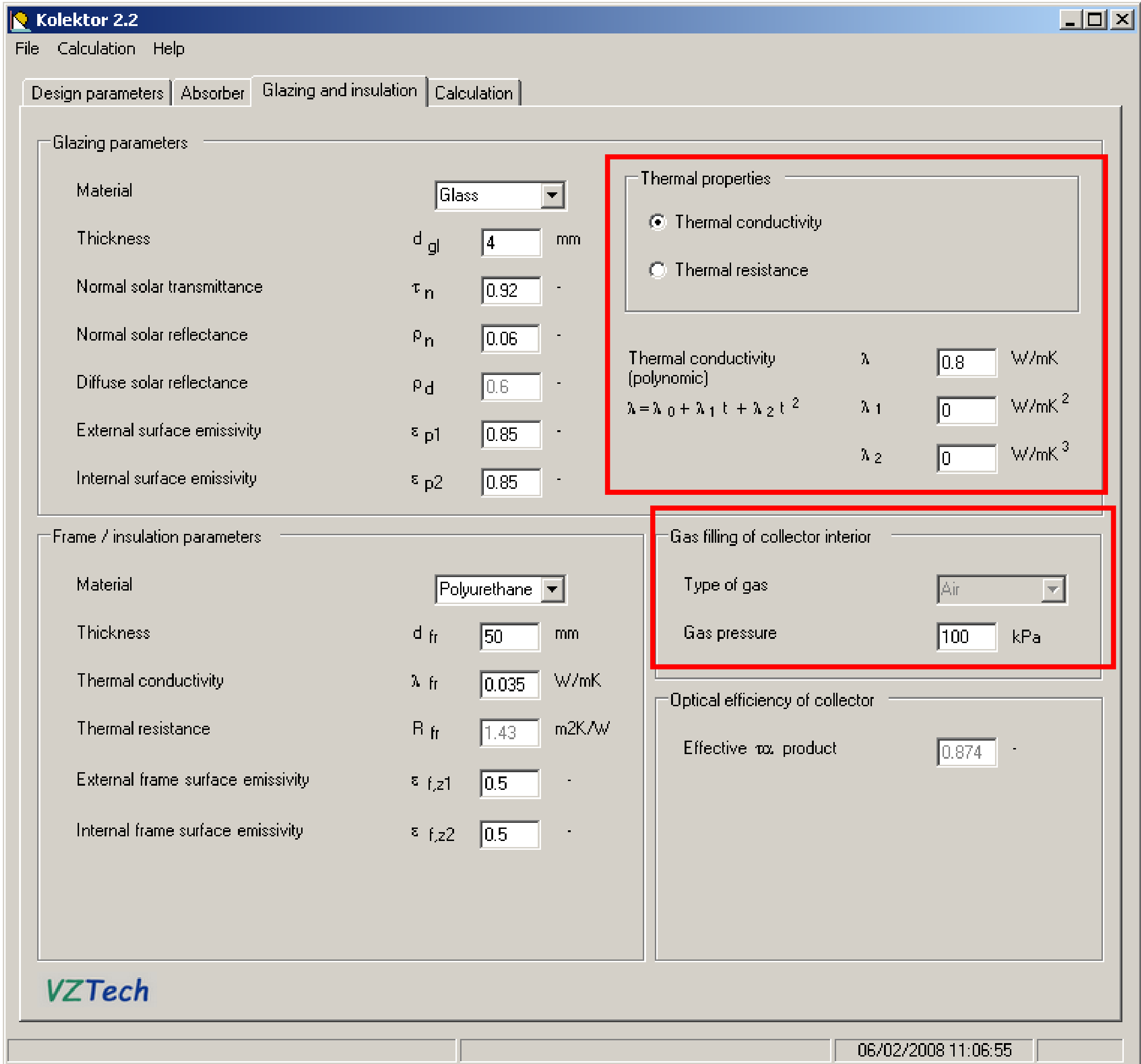
Design tool KOLEKTOR 2.2



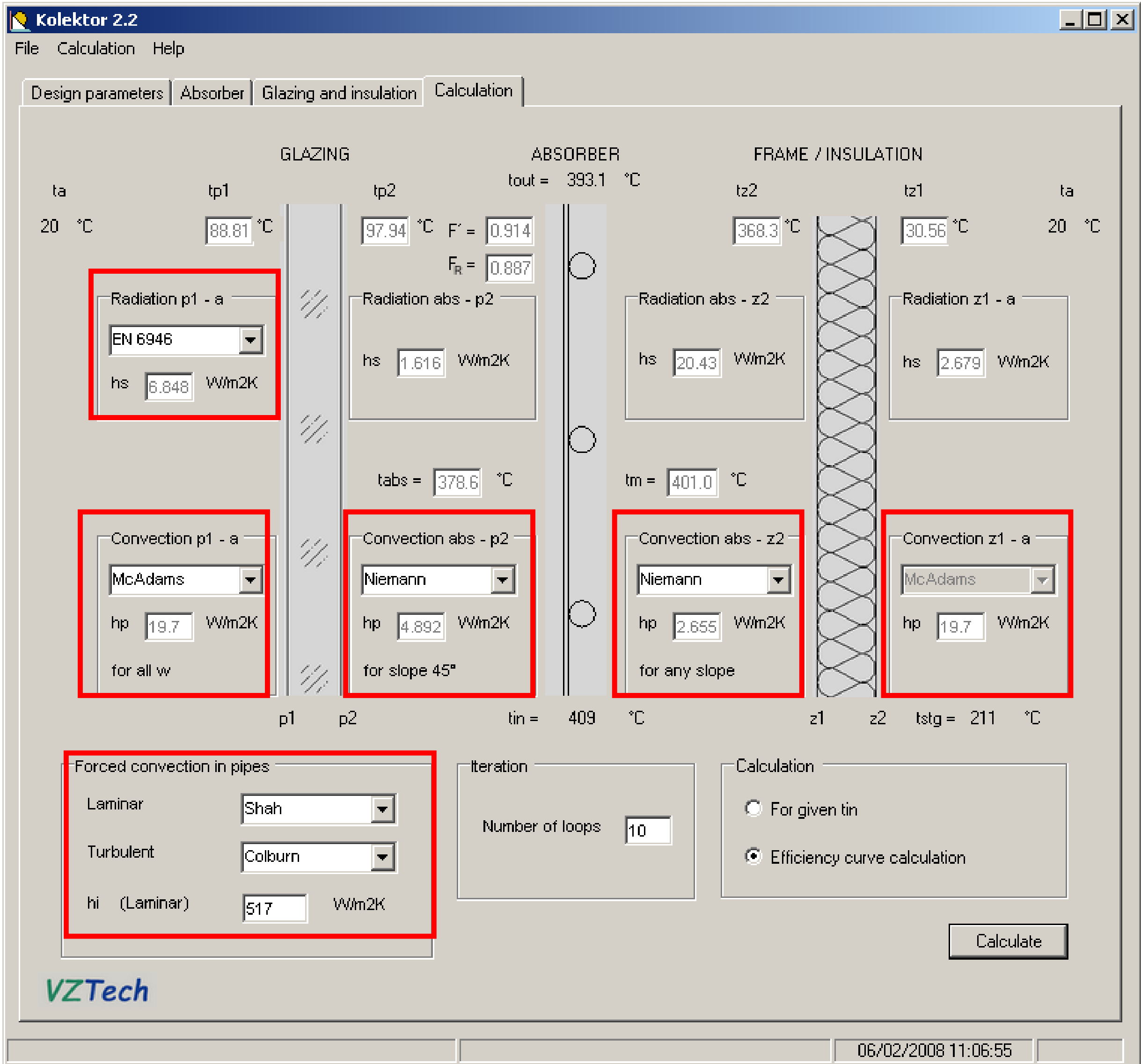
Design tool KOLEKTOR 2.2



Design tool KOLEKTOR 2.2



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:
    ε = 5 % α = 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**

run 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 $\boldsymbol{\eta_0}$, $\boldsymbol{a_1}$, $\boldsymbol{a_2}$ by use of **evaluation.xls**

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

- 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 m^2, 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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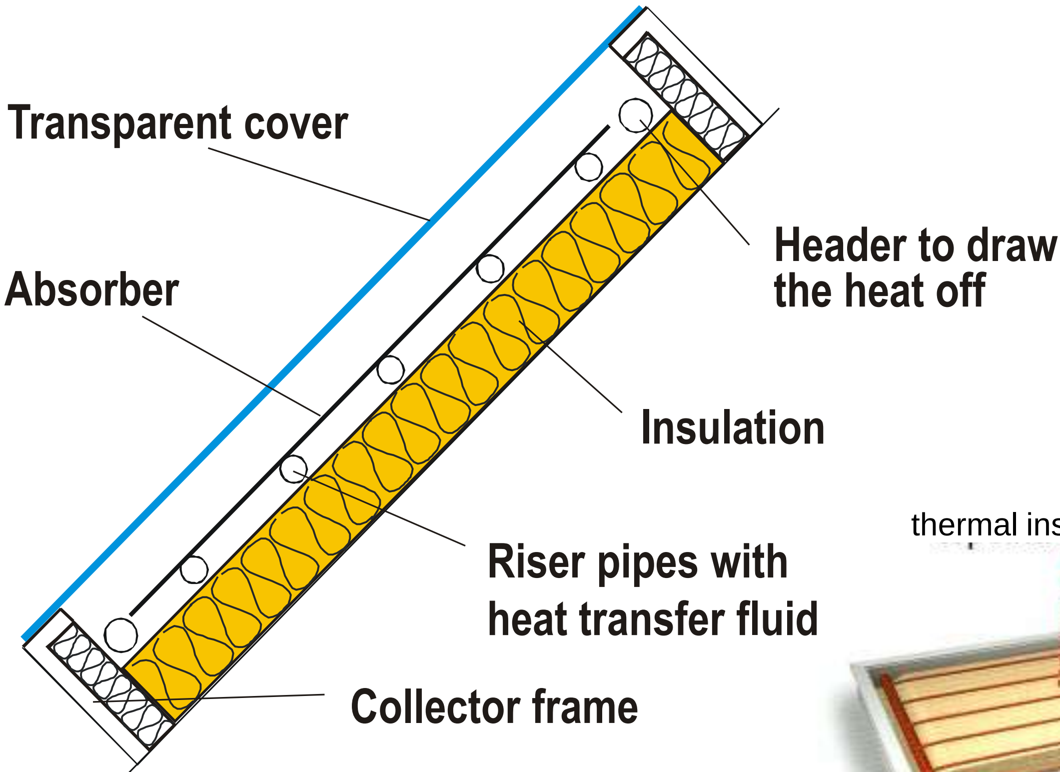
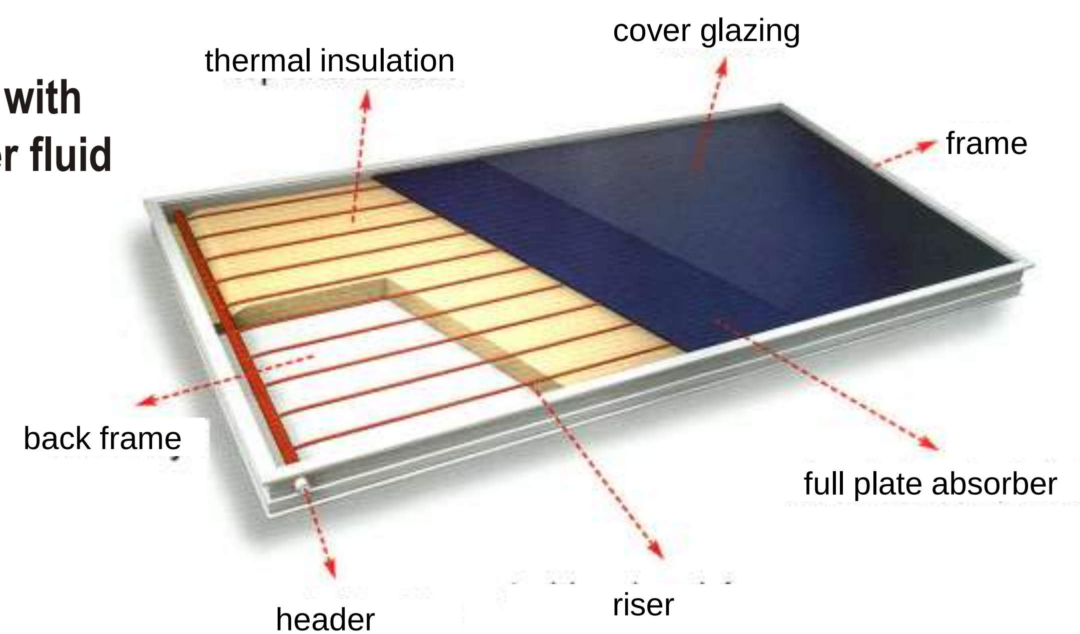
Content

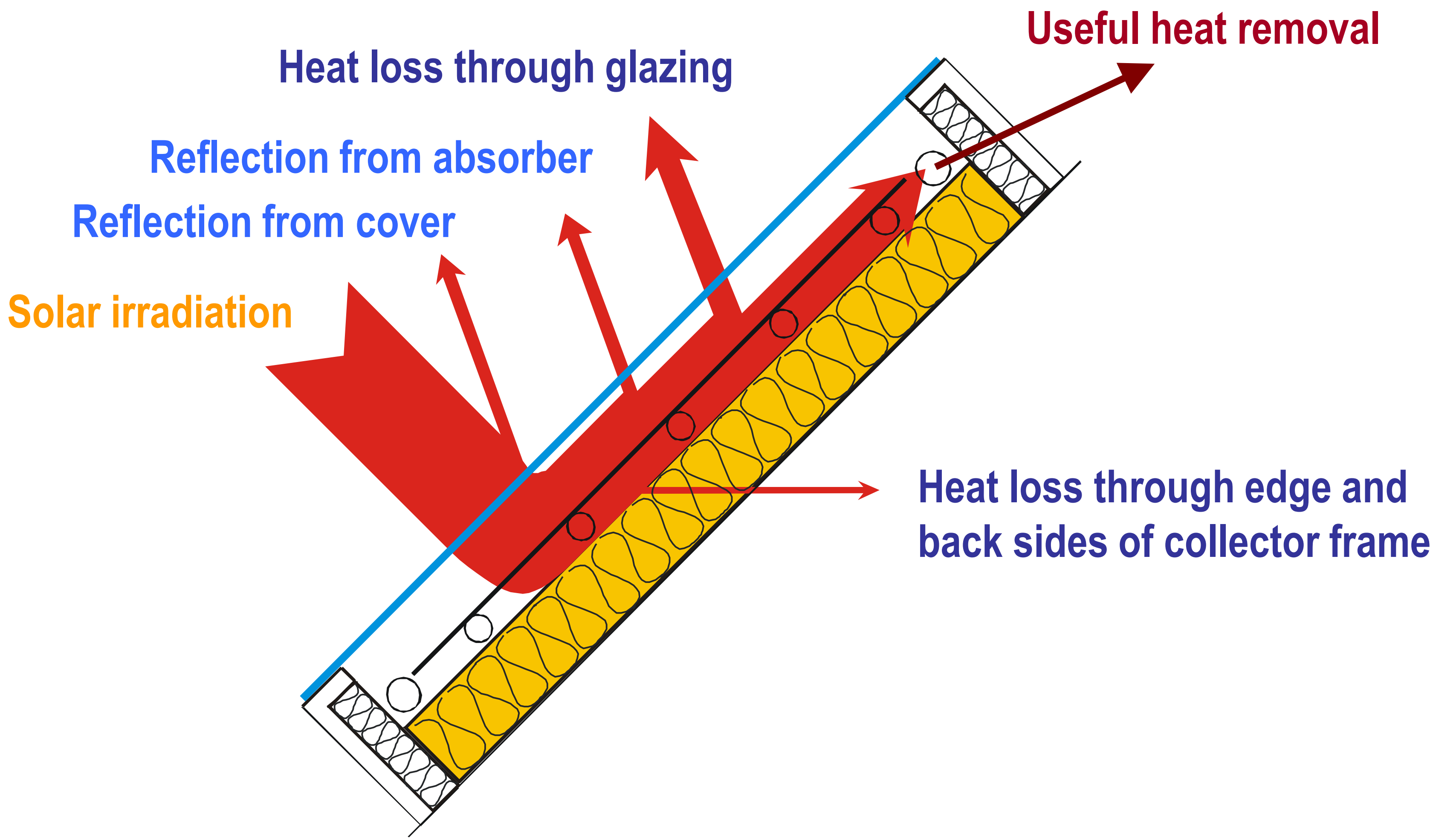
- how to calculate thermal output of solar collector on theoretical basis
  - from detailed data on construction
  - at given operation conditions
  - with use of different alternative models of heat transfer phenomena
- to predict the influence of
  - change in construction of the collector
  - operation conditions
- on efficiency curve
- on annual energy yields of the collector

Solar thermal flat-plate collector





Solar collector energy balance



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

$$\frac{\mathrm{d}Q}{\mathrm{d}t}=\dot{Q}_{\mathrm{s}}-\dot{Q}_{\mathrm{l,o}}-\dot{Q}_{\mathrm{l,t}}-\dot{Q}_{\mathrm{u}}$$ general description

steady state dQ/dt = 0 $$\dot{Q}_{\mathrm{u}}=\dot{Q}_{\mathrm{s}}-\dot{Q}_{\mathrm{l,o}}-\dot{Q}_{\mathrm{l,t}}$$

Q_s	solar energy input [W]	$Q_s = G.A_c$
$Q_{l,o}$	optical losses [W]	$Q_{l,o} = Q_s - Q_s(\tau\alpha)_{ef}$
$Q_{l,t}$	thermal losses [W]	$Q_{l,t} = U.A_c\,(t_{abs} - t_a)$
Q_u	useful heat removed from collector [W]	$Q = Mc(t_{out} - t_{in})$

External energy balance: efficiency

useful heat output:

$$\dot{Q}_{\mathrm{u}} = G A_{\mathrm{c}} (\tau\alpha)_{\mathrm{ef}} - U A_{\mathrm{c}} (t_{\mathrm{abs}} - t_{\mathrm{a}})$$

efficiency:

$$\eta = \frac{\dot{Q}_u}{\dot{Q}_s} = \frac{\dot{Q}_u}{G A_{\mathrm{c}}} = \frac{G A_{\mathrm{c}} (\tau\alpha)_{ef} - U A_c (t_{abs} - t_a)}{G A_{\mathrm{c}}}$$

$$\eta = (\tau\alpha)_{\mathrm{ef}} - U \frac{(t_{\mathrm{abs}} - t_{\mathrm{a}})}{G}$$

Efficiency formulas

$$\eta = (\tau\alpha)_{\mathrm{ef}} - U\frac{(t_{\mathrm{abs}} - t_{\mathrm{a}})}{G}$$

external balance, losses

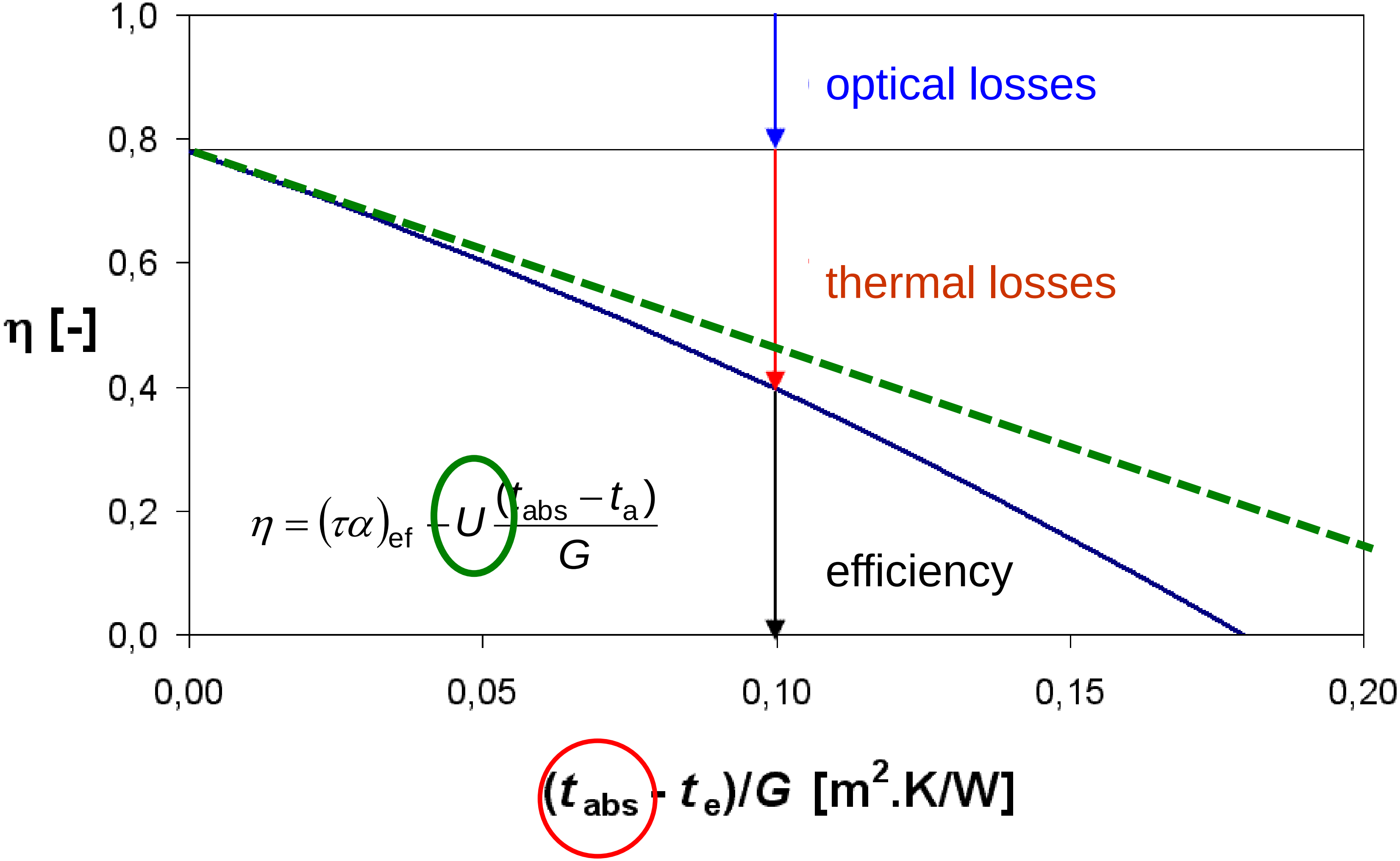
$$\eta = F'\left[(\tau\alpha)_{\mathrm{ef}} - U\frac{(t_{\mathrm{m}} - t_{\mathrm{a}})}{G}\right]$$

+ internal balance
influence of absorber design
European standards

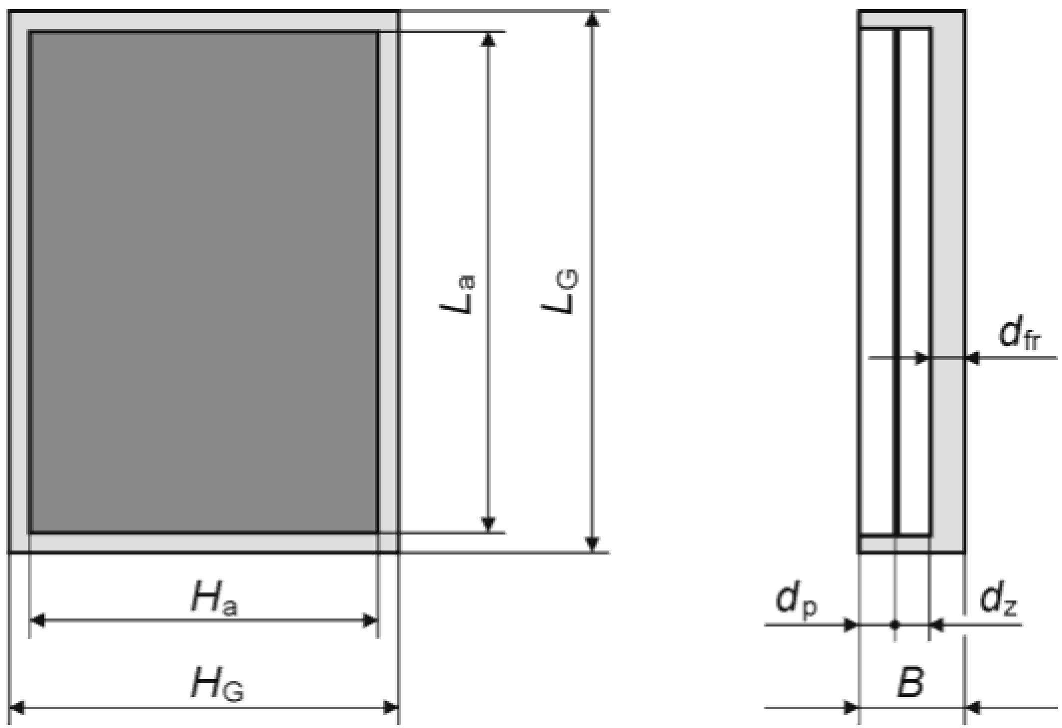
$$\eta = F_R\left[(\tau\alpha)_{\mathrm{ef}} - U\frac{(t_{\mathrm{in}} - t_{\mathrm{a}})}{G}\right]$$

+ influence of flowrate,
specific heat of fluid
US standards

Solar collector efficiency (based on t_{abs})



Reference area for calculations



gross area: A_G — maximum area = $H_G.L_G$

edge side area: A_b — $(2H_G + 2L_G).B$

aperture area: A_a — **area of glazing = $H_a.L_a$**

Reference area - comparison

Collector testing standards relates collector efficiency to:

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

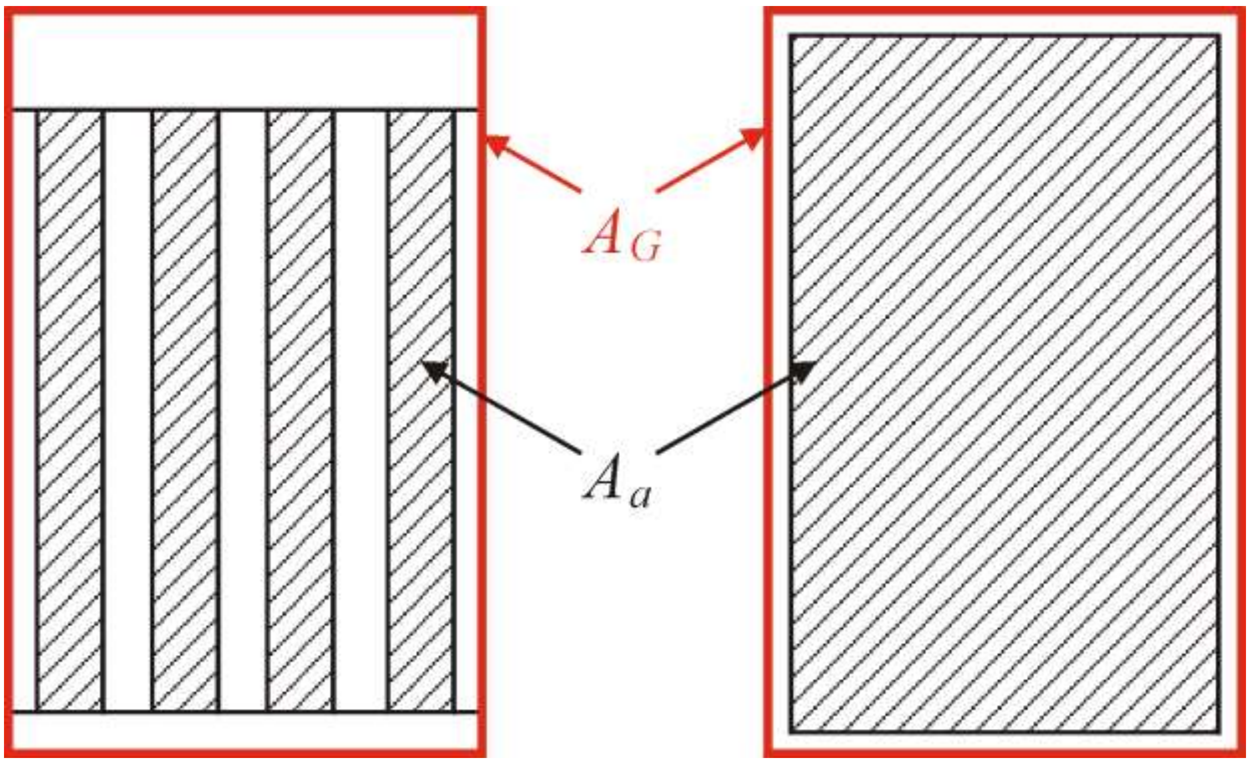
EN 12975-2

EN ISO 9806

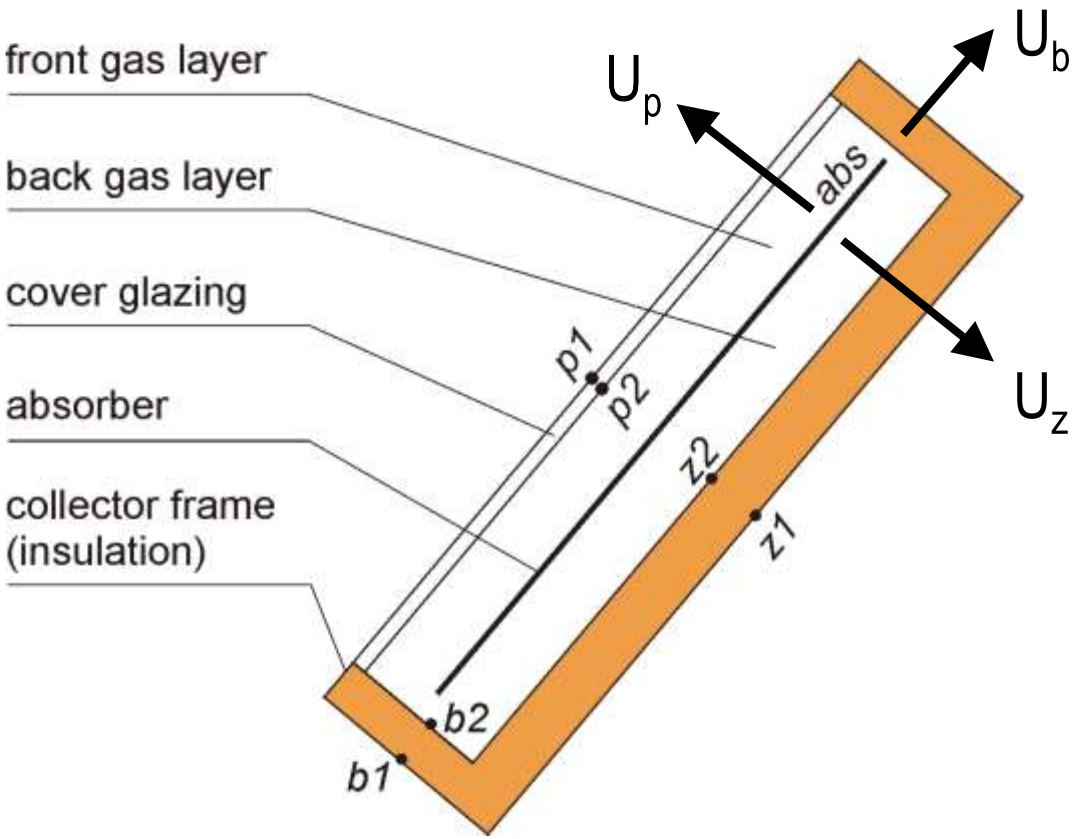
- aperture area: for comparison of collector properties or **design**
- gross area: for decision on roof production **potential**

for comparison of collector with different **active** areas

Reference area - comparison



External energy balance: detailed



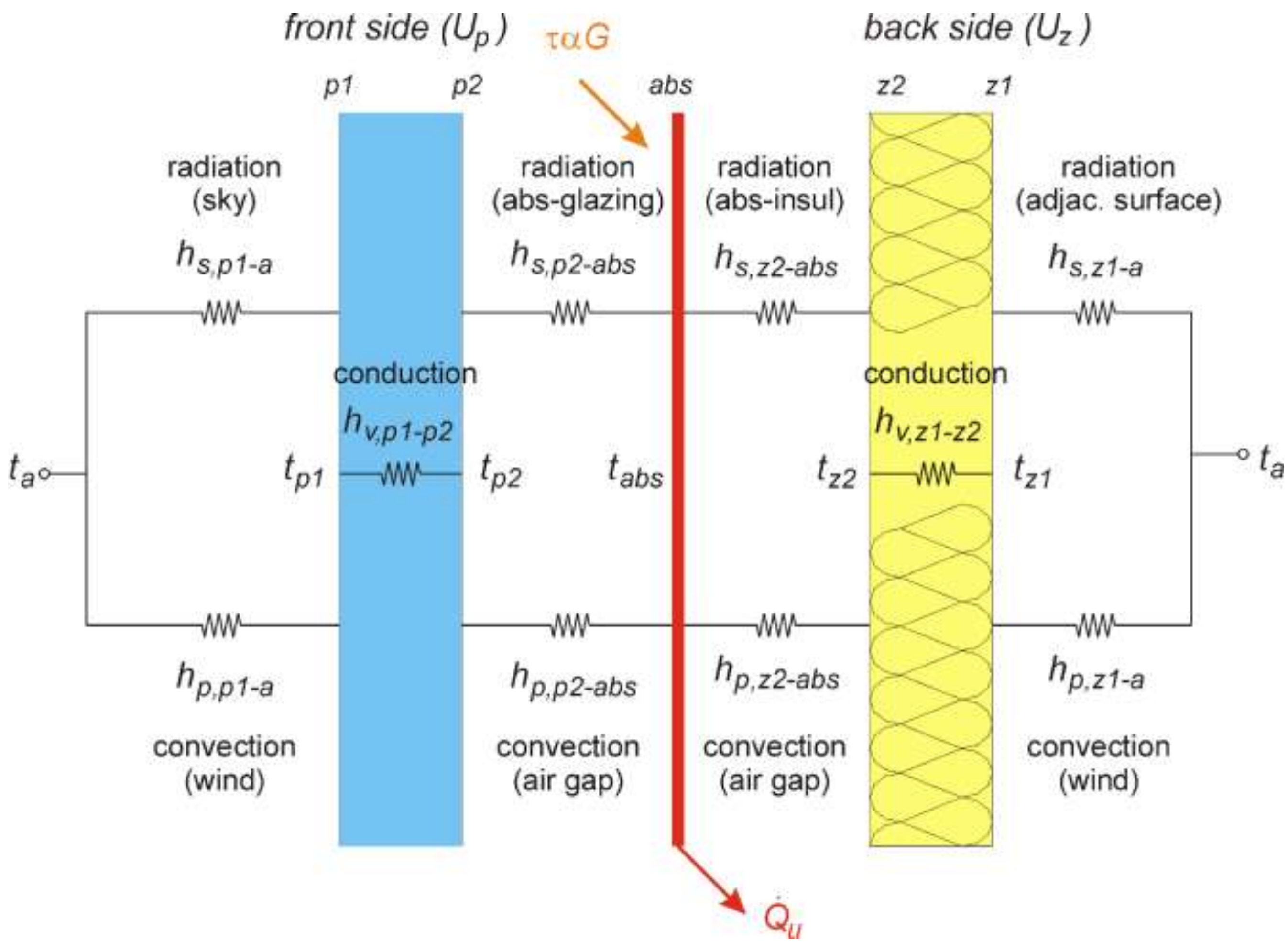
U_p = front U-value

U_z = back U-value

U_b = side U-value

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

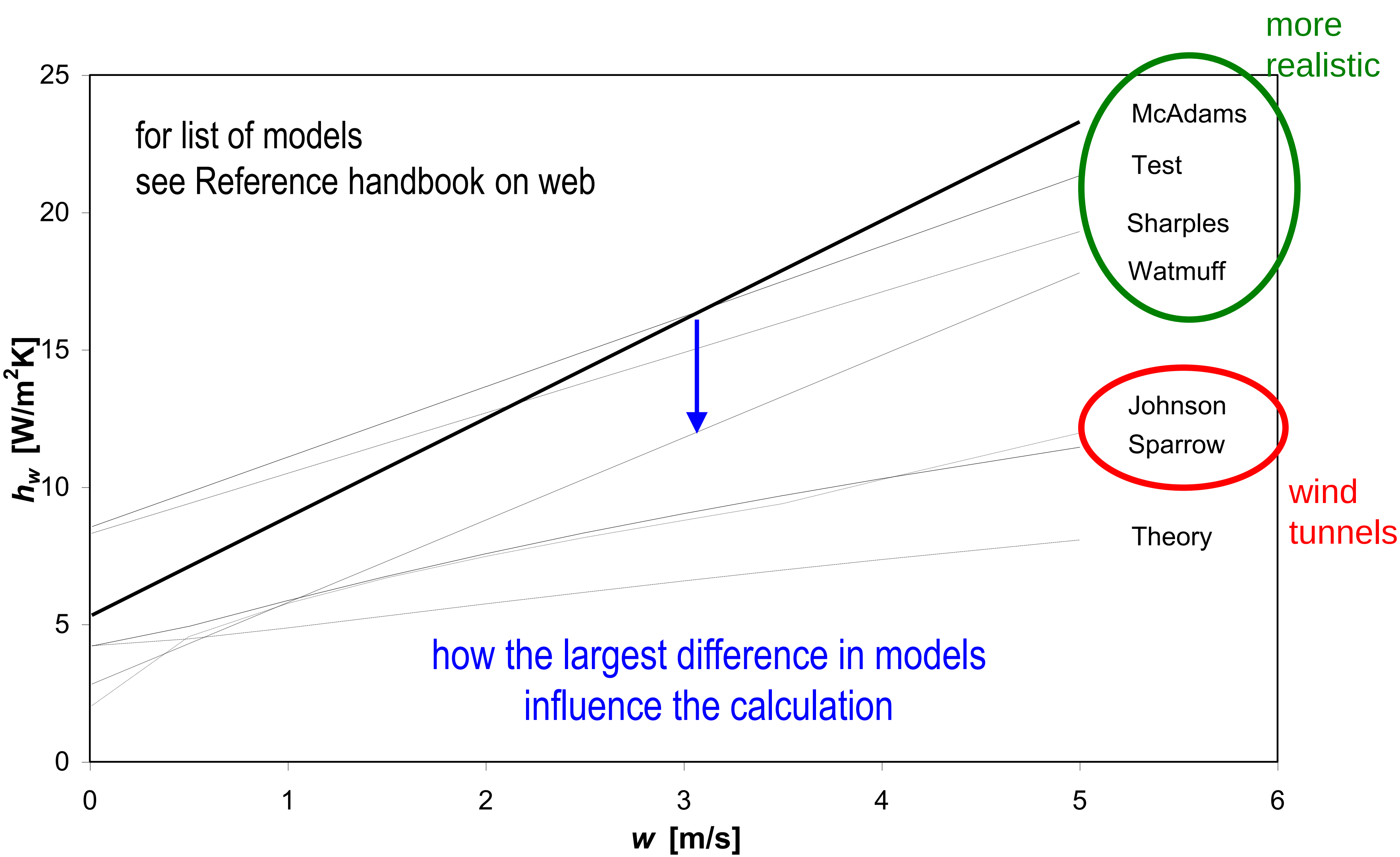
Scheme of external balance



Wind convection

- Wind induced convection heat transfer from cover to ambient:
  - mix of natural and forced covection
  - 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 $h_{\mathrm{w}} = a + bw$

Wind convection



Wind convection for w = 3 m/s

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

  select McAdams ... Calculate

  select Watmuff ... Calculate

McAdam’s correlation
h_w = **16.1** W/m^{2}K
t_{out} = **55.6 °C**

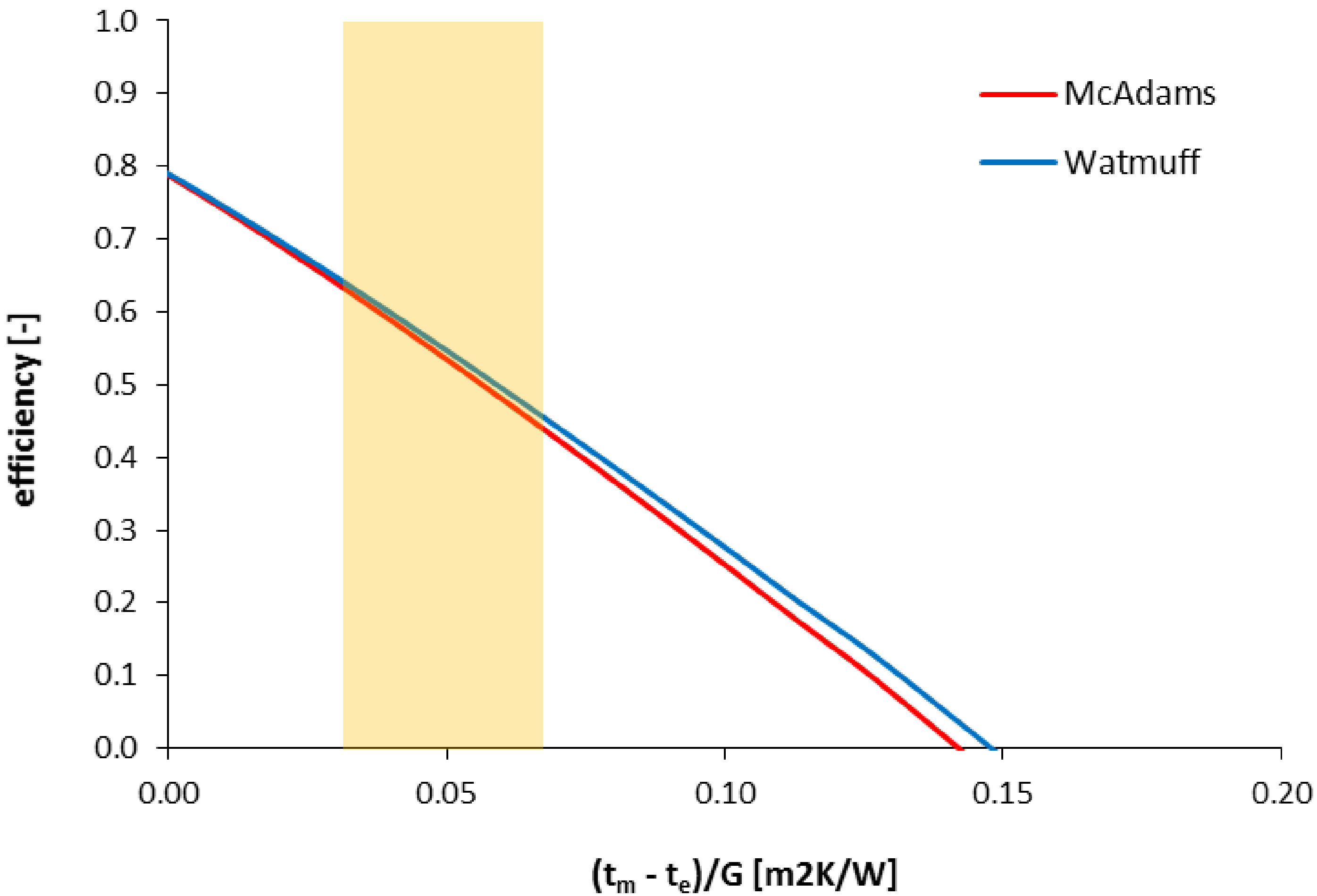
Watmuff correlation
h_w = **11.3** W/m^{2}K
t_{out} = **55.7 °C**

Wind convection for *w* = 3 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 m/s



Wind convection for *w* = 3 m/s

- **how can the model selection influence the annual energy yields?**
  - McAdams η_0 = 0.789 a_1 = 4.857 a_2 = 0.006
  - Watmuff η_0 = 0.791 a_1 = 4.664 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

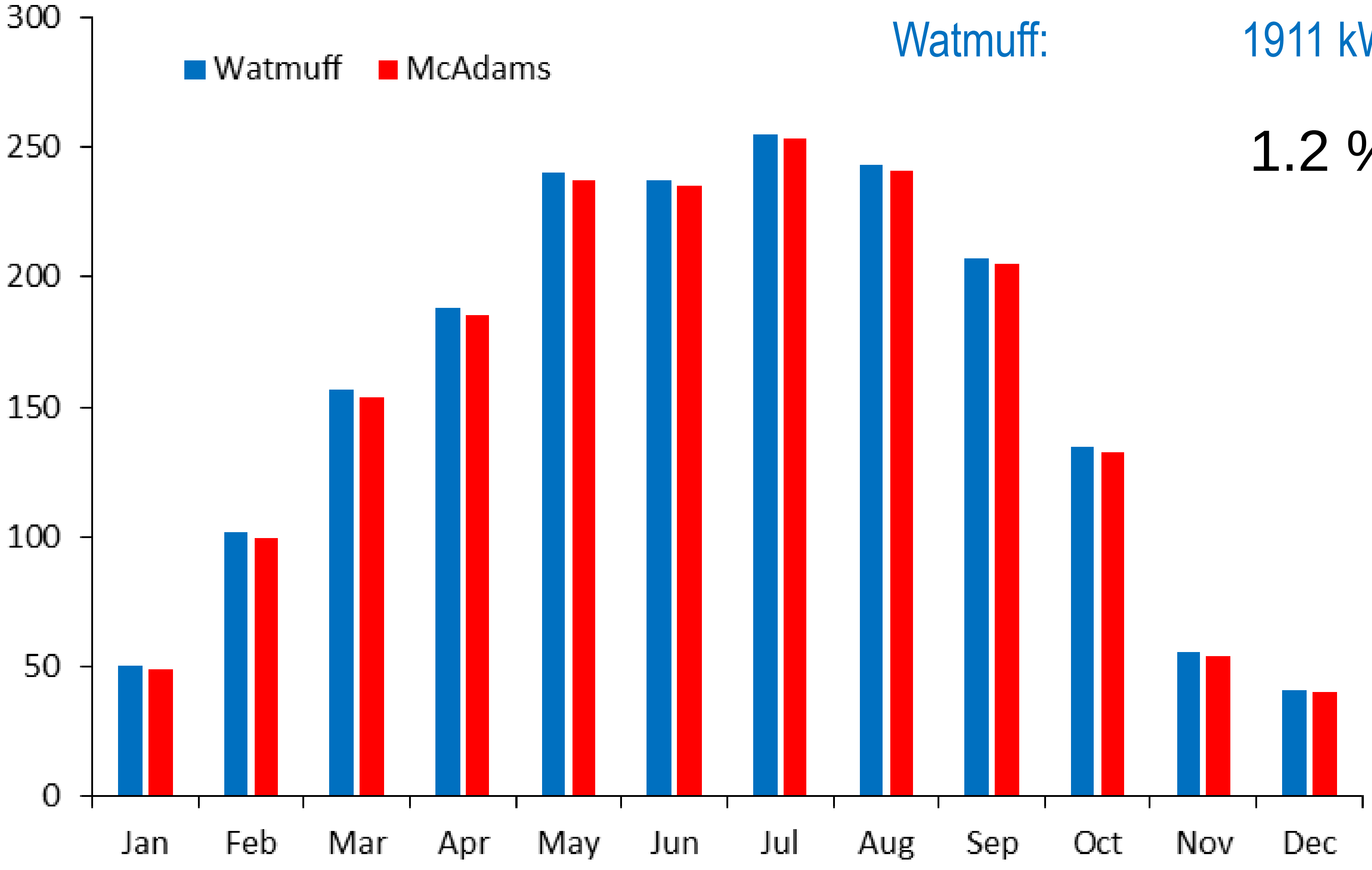
- **compare the annual sums for both alternatives**

Wind convection for *w* = 3 m/s

McAdams: 1887 kWh/a

Watmuff: 1911 kWh/a

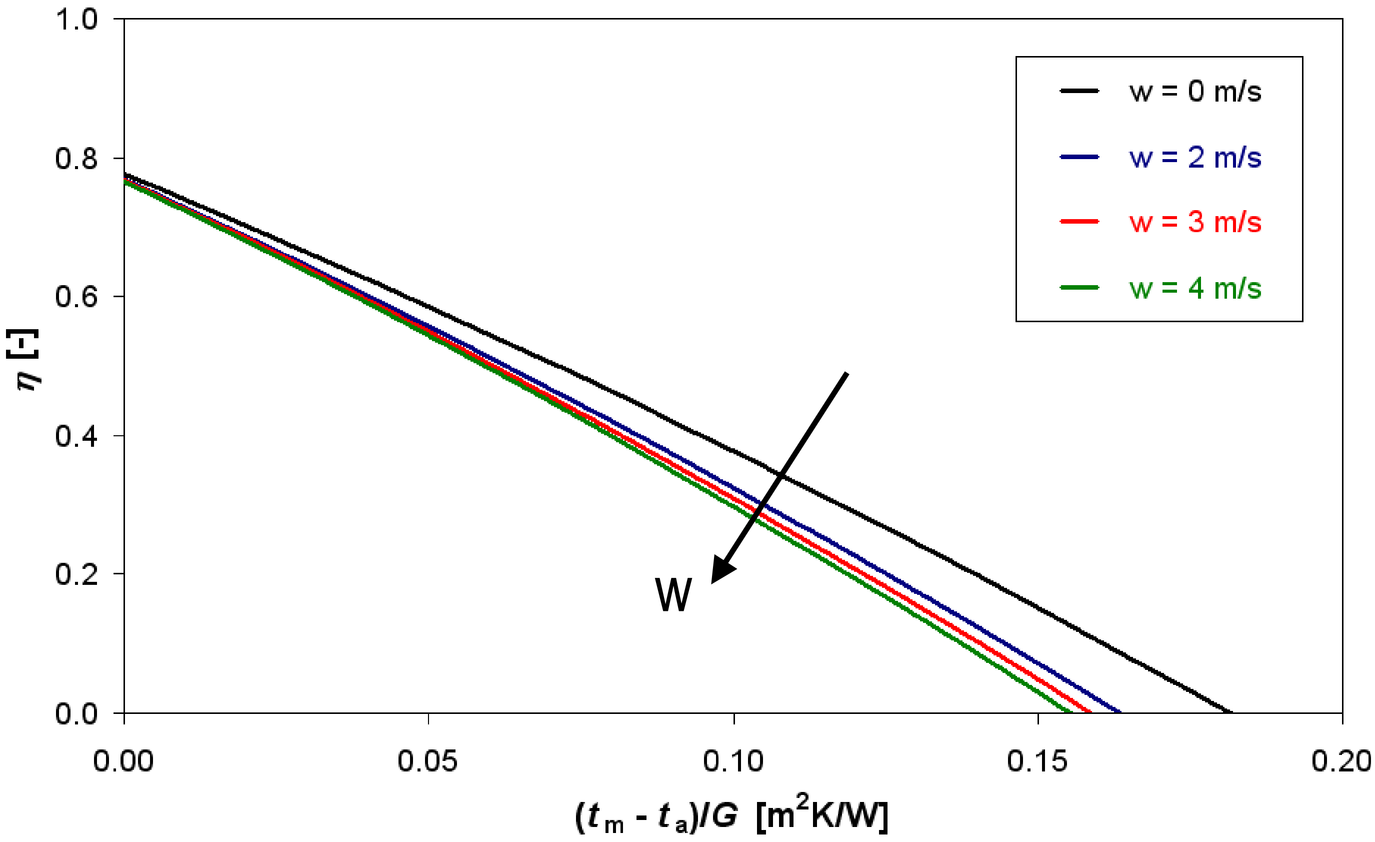
1.2 %



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



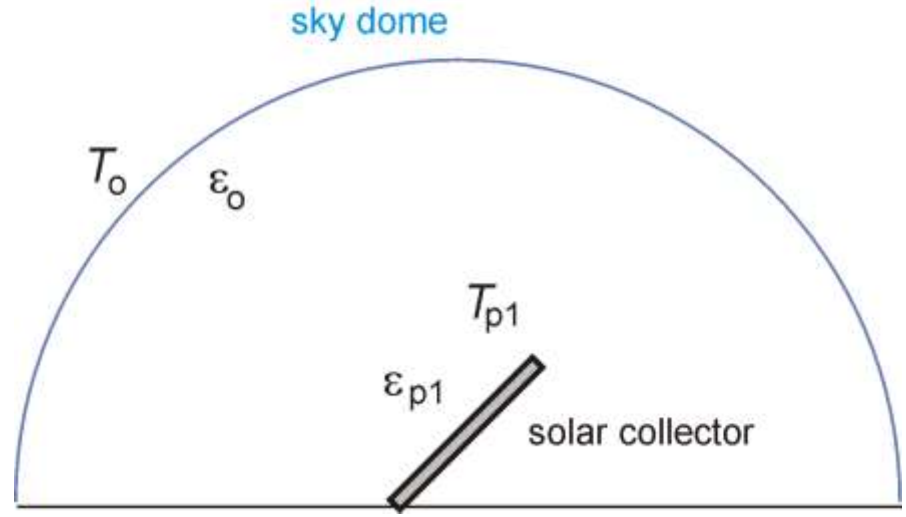
testing standard: air velocity during the testing procedure ***w* > 3 m/s**

use of artifical wind needed!

Sky radiation

Radiation heat exchange

$$q_{\mathrm{p1-o}} = \varepsilon_{\mathrm{p1}} \sigma (T_{\mathrm{p1}}^4 - T_{\mathrm{o}}^4) \qquad [\mathrm{W/m^2}]$$



Sky temperature, sky emittance

$$T_{\mathrm{o}}^4 = \varepsilon_{\mathrm{o}} T_{\mathrm{a}}^4 \qquad [\mathrm{K}]$$

Sky radiation heat transfer coefficient related to ambient temperature

$$h_{\mathrm{s,p1-a}} = \varepsilon_{\mathrm{p1}} \sigma \frac{T_{\mathrm{p1}}^4 - T_{\mathrm{o}}^4}{T_{\mathrm{p1}} - T_{\mathrm{a}}} \qquad [\mathrm{W/m^2K}]$$

Sky temperature models

Number of models for sky temperature T_o calculation based on:

	cloudy sky	clear sky
ambient temperature T_a	$T_o = T_a$	$T_o = 0.0552\left(T_a\right)^{1.5}$ (Swinbank, 1963)

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

for more see
Reference handbook on web

- clear sky conditions
- cloudy sky conditions

Sky temperature models

Sky temperature calculation based on:

$$\boxed{T_o = T_a}$$

$$\boxed{T_o = 0.0552\,(T_a)^{1.5}}$$

e_{p1} = 0.85, t_{p1} = 40 °C, t_a = 20 °C

$h_{s,p1\text{-}a}$ = **5.38** W/m²K

t_o = 3.9 °C

$h_{s,p1\text{-}a}$ = **8.97** W/m²K

e_{p1} = 0.10, t_{p1} = 40 °C, t_a = 20 °C

$h_{s,p1\text{-}a}$ = **0.63** W/m²K

$h_{s,p1\text{-}a}$ = **1.06** W/m²K

considerable decrease of heat transfer coefficient if low-e coating

does it mean also improvement of collector performance?

Coupling sky radiation + wind convection

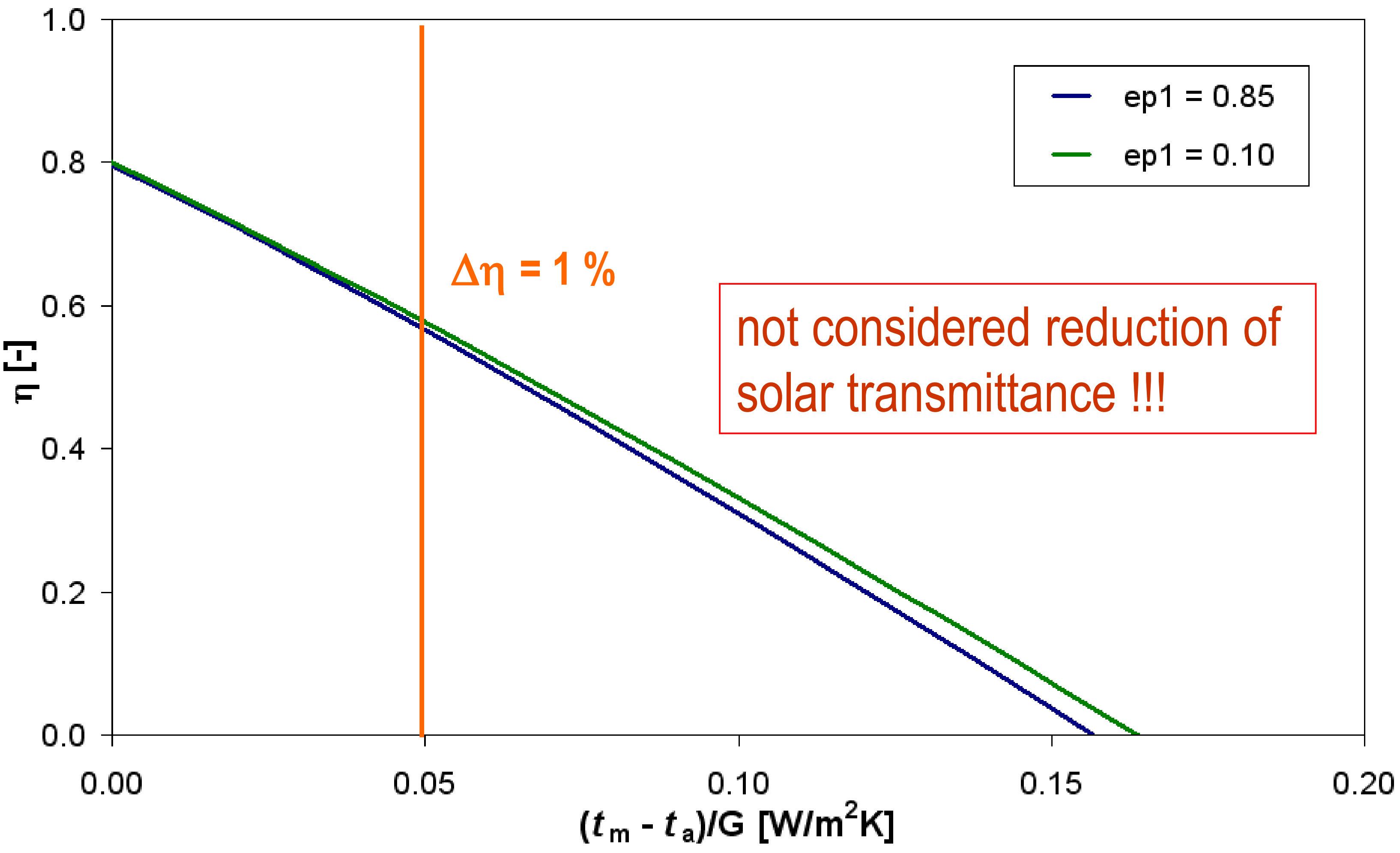
heat transfer from glazing exterior surface: **proportions**

$q = q_s$ (sky radiation) + q_p (wind convection)

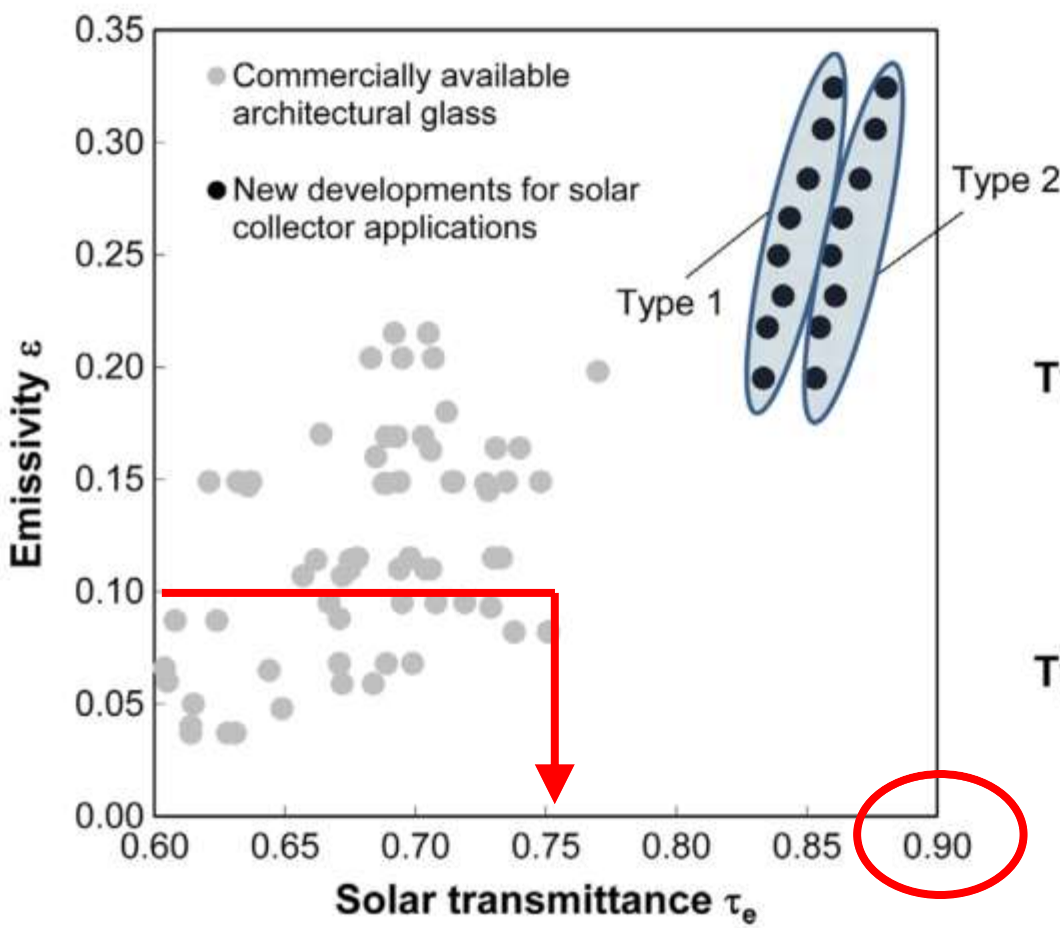
w [m/s]	h_s [W/m²K]	h_p [W/m²K]	$h_s + h_p$ [W/m²K]
0	5.1 / 0.6	5.3	10.4 / 5.9
2	5.0 / 0.6	12.5	17.5 / 13.1
4	5.0 / 0.6	19.7	24.7 / 20.3
6	5.0 / 0.6	26.2	31.2 / 26.8

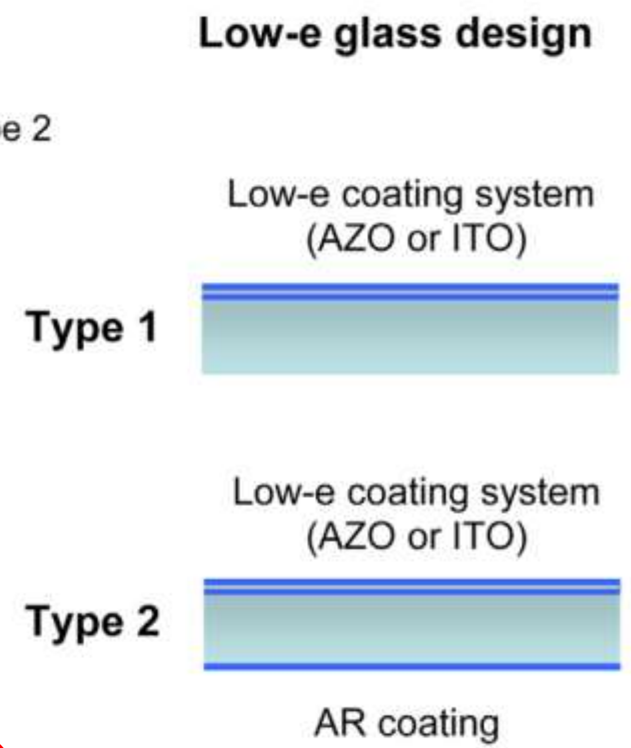
convection heat transfer is dominant (if wind is present ...)

Influence of sky radiation reduction



Influence of sky radiation reduction





Conduction through cover glazing

heat conductance

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

<u>single glazing:</u> glass: λ_{gl} = 0.8 W/mK, L_{gl} = 4 mm

h_{gl} = 200 W/m²K

PC: λ_{gl} = 0.2 W/mK, L_{gl} = 4 mm

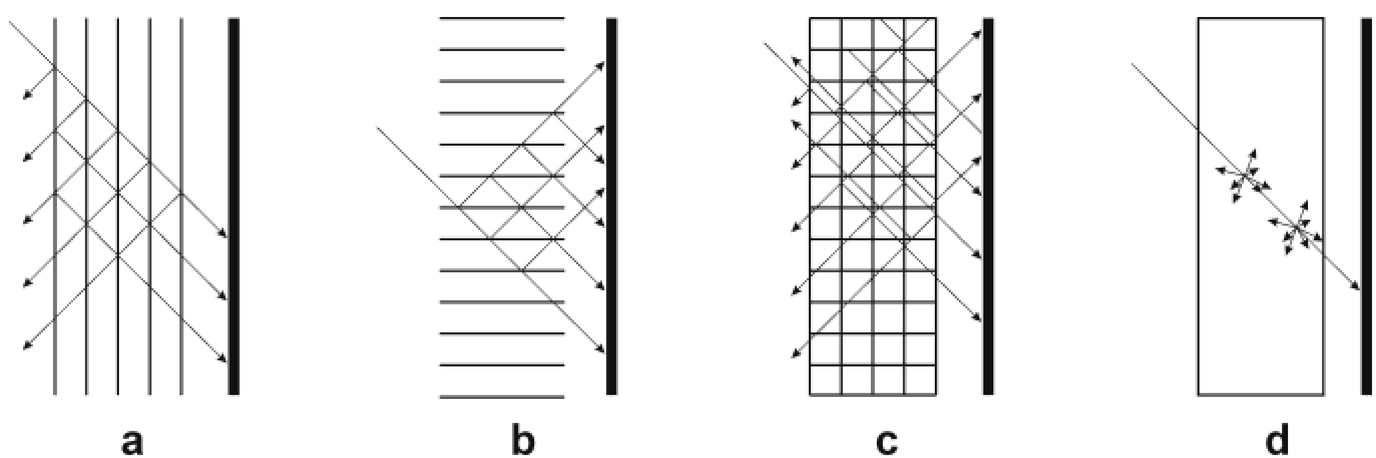
h_{gl} = 50 W/m²K

practicaly negligible

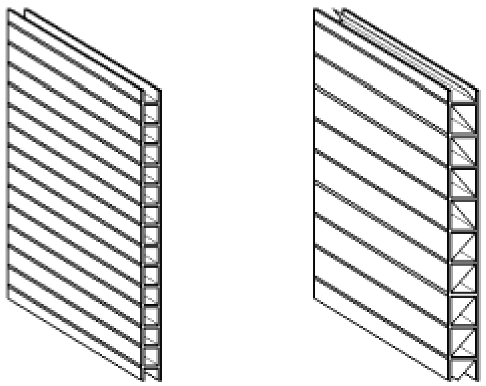
Conduction through cover glazing

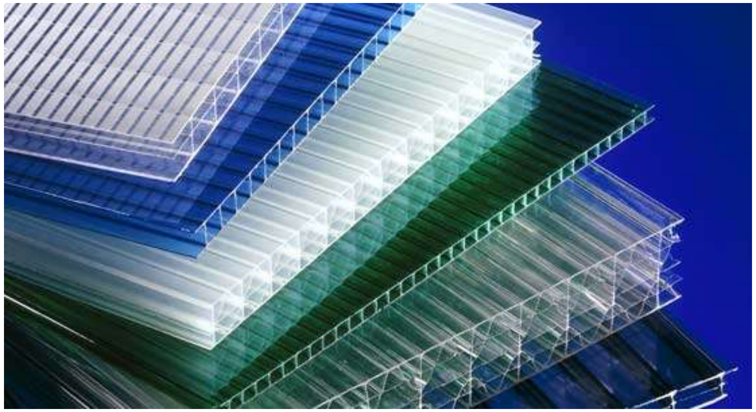
- **transparent insulation structures**
  - channel structures: h_{gl} = 2 to 8 W/m^{2}K
  - honeycombs: h_{gl} = 1 to 2 W/m^{2}K
  - aerogels: h_{gl} = less than 1 W/m^{2}K

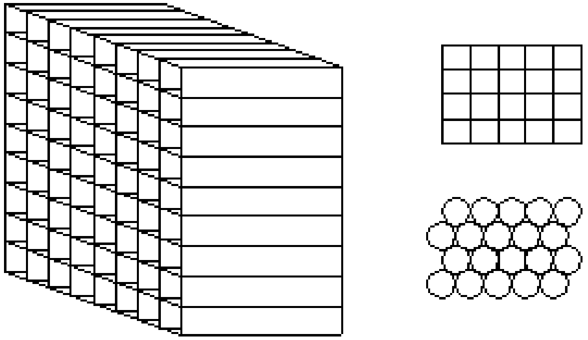
function of mean temperature



Conduction through cover glazing



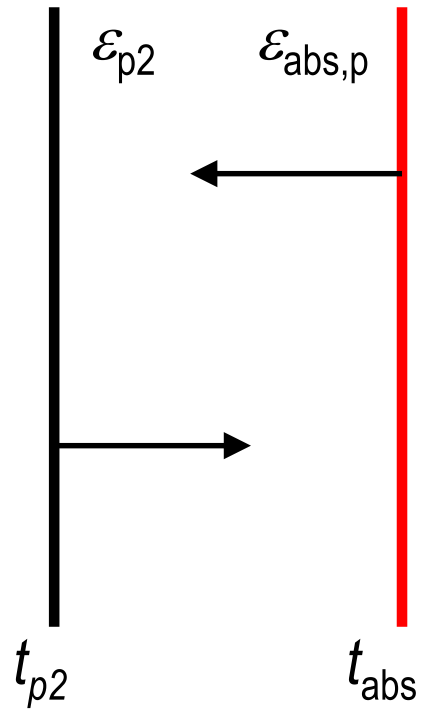




Radiation between absorber and cover

Radiation heat exchange

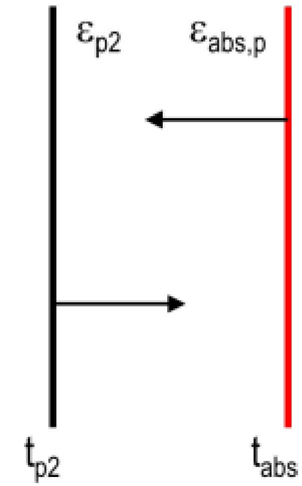
$$q_{\text{s,abs-p2}} = \sigma \frac{T_{\text{abs}}^4 - T_{\text{p2}}^4}{\dfrac{1}{\varepsilon_{\text{p2}}} + \dfrac{1}{\varepsilon_{\text{abs,p}}} - 1} \qquad [\text{W/m}^2]$$



Radiation between absorber and cover

influence of absorber emissivity

t_{abs}	40 °C		
t_{p2}	20 °C		
$\varepsilon_{abs,p}$	0.85	0.10	0.05
ε_{p2}	0.85	0.85	0.85
$h_{s,p1\text{-}a}$	4.7 W/m^{2}K	0.6 W/m^{2}K	0.3 W/m^{2}K



8x

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 $\varepsilon_{abs} = 0.85$ $\varepsilon_{abs} = 0.10$ $\varepsilon_{abs} = 0.05$

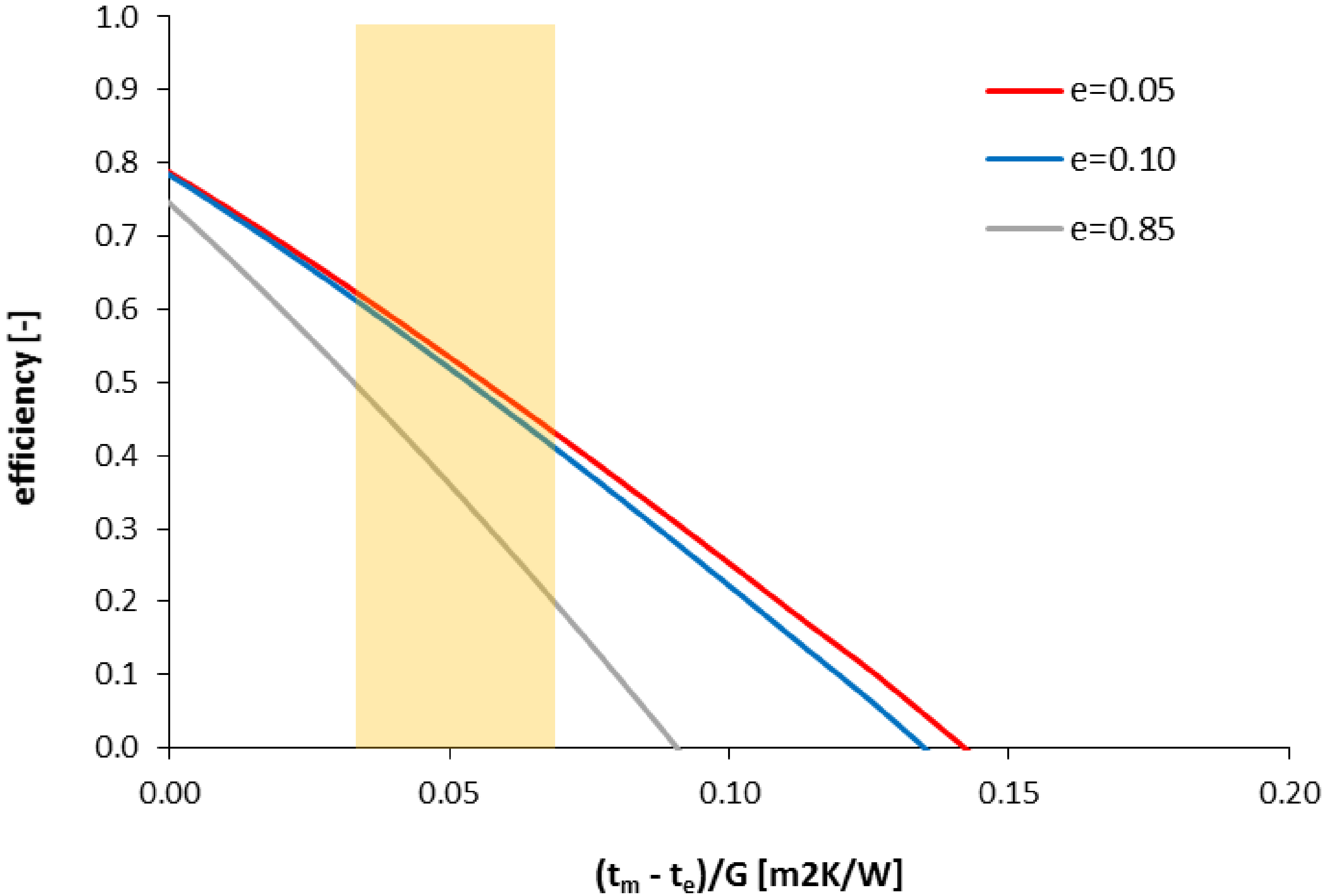
Spectrally selective absorber

- 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

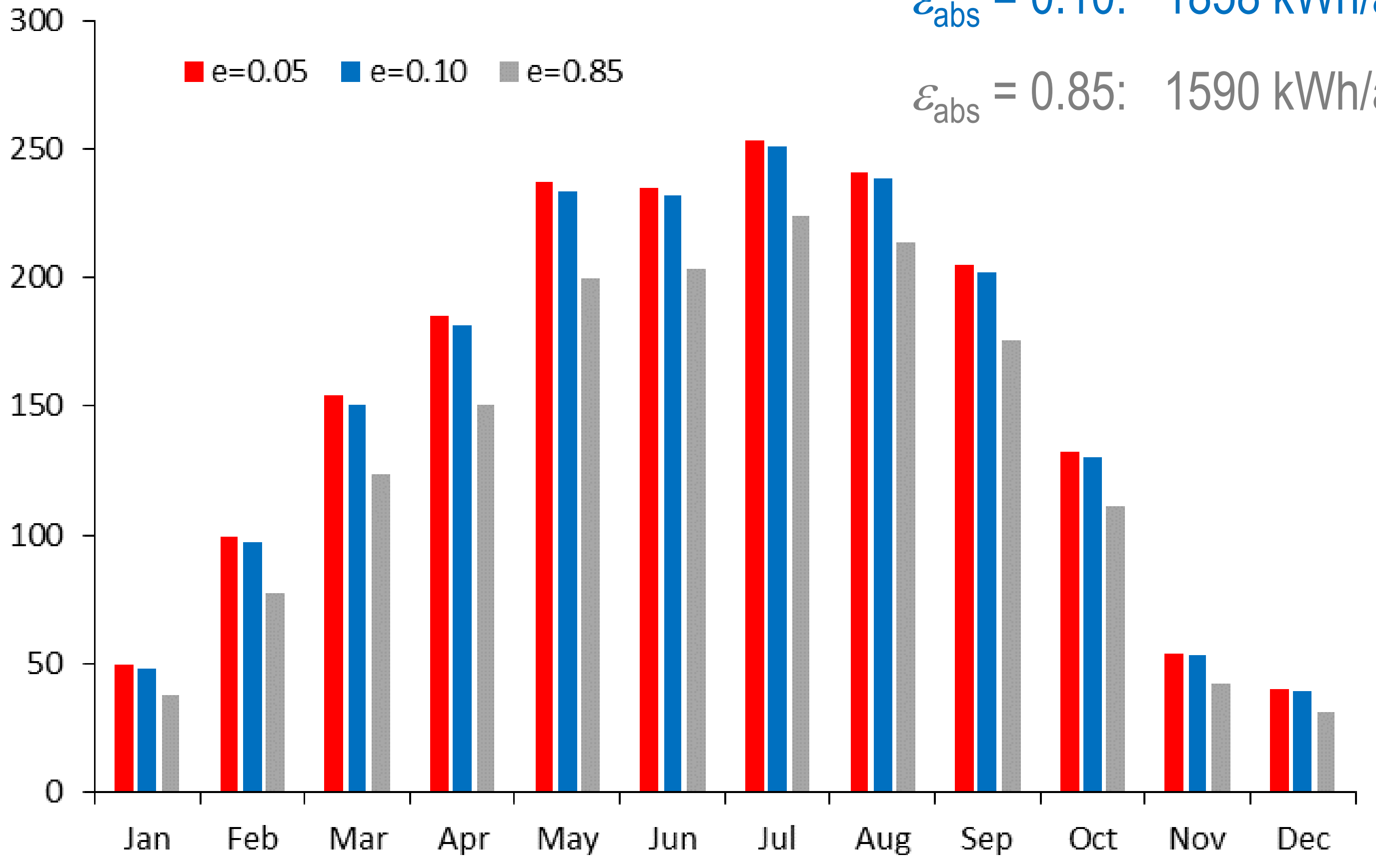
- **how can the absorber emissivity influence the annual energy yields of collector?**
  - $\varepsilon_{abs} = 0.05$ $\eta_0 = 0.789$ $a_1 = 4.857$ $a_2 = 0.006$
  - $\varepsilon_{abs} = 0.10$ $\eta_0 = 0.785$ $a_1 = 5.006$ $a_2 = 0.008$
  - $\varepsilon_{abs} = 0.85$ $\eta_0 = 0.744$ $a_1 = 6.773$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

ε_{abs} = 0.05: 1887 kWh/a

ε_{abs} = 0.10: 1858 kWh/a -1.5 %

ε_{abs} = 0.85: 1590 kWh/a -16 %



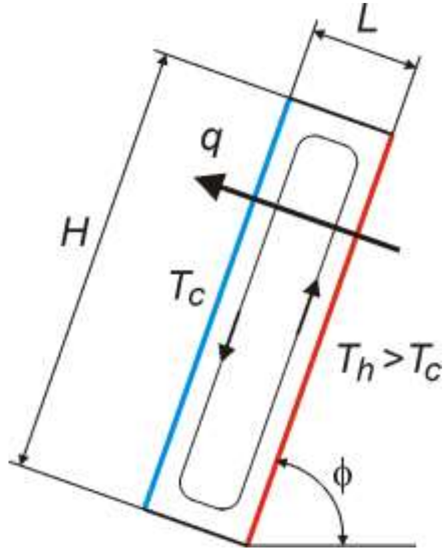
Spectrally selective absorber

- 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

$$h_{p+v} = \mathrm{Nu}_L \frac{\lambda_g}{L} \qquad [\mathrm{W/m^2K}]$$



Characteristic dimension:
thickness of gas layer L

Nusselt number:
characterization of conduction heat transfer → λ_g/L
enhancement by fluid convection (driven by buoyancy) → Nu_L
definition: ratio of the total heat transfer to conductive heat transfer

Convection in the air gap - criteria

Nusselt number: $\mathrm{Nu_L} = f\,(\mathrm{Gr_L}, \mathrm{Pr})$

Grashof number: $$\mathrm{Gr_L} = \beta \frac{g L^3 \Delta t}{\nu^2} = \frac{2}{(T_h + T_c)} \cdot \frac{g L^3 (t_\mathrm{h} - t_\mathrm{c})}{\nu^2}$$

ratio of the buoyancy to viscous force acting on a fluid

Prandtl number: $$\mathrm{Pr} = \frac{\nu}{a} = \frac{\nu \rho c}{\lambda}$$

express conformity of velocity and temperature fields,
ratio of momentum diffusivity (kinematic viscosity) to thermal diffusivity

Convection in the air gap - criteria

Rayleigh number:

$$\mathrm{Ra} = \mathrm{Gr} \cdot \mathrm{Pr} = \frac{\beta g L^3 \left(t_\mathrm{h} - t_\mathrm{c}\right)}{\nu a}$$

ratio of buoyancy forces to thermal and momentum diffusivities of fluid

for standard conditions

400 < Gr < 60 000

0.72 < Pr < 0.73

300 < Ra < 44 000

Structure of the convection models

Hollands:

$$\mathrm{Nu}_L = 1 + 1.44\left[1 - \frac{1708}{\mathrm{Ra}_L \cos\phi}\right]^{+}\left(1 - \frac{(\sin 1.8\phi)^{1.6}\,1708}{\mathrm{Ra}_L \cos\phi}\right) + \left[\left(\frac{\mathrm{Ra}_L \cos\phi}{5830}\right)^{1/3} - 1\right]^{+}$$

Buchberg:

$$\mathrm{Nu}_L = 1 + 1.446\left(1 - \frac{1708}{\mathrm{Ra}_L \cos\phi}\right)^{+}$$

$$\mathrm{Nu}_L = 0.157\left(\mathrm{Ra}_L \cos\phi\right)^{0.285}$$

Randall:

$$\mathrm{Nu}_L = 0.118\left[\mathrm{Ra}_L \cos^2(\phi - 45)\right]^{0.29}$$

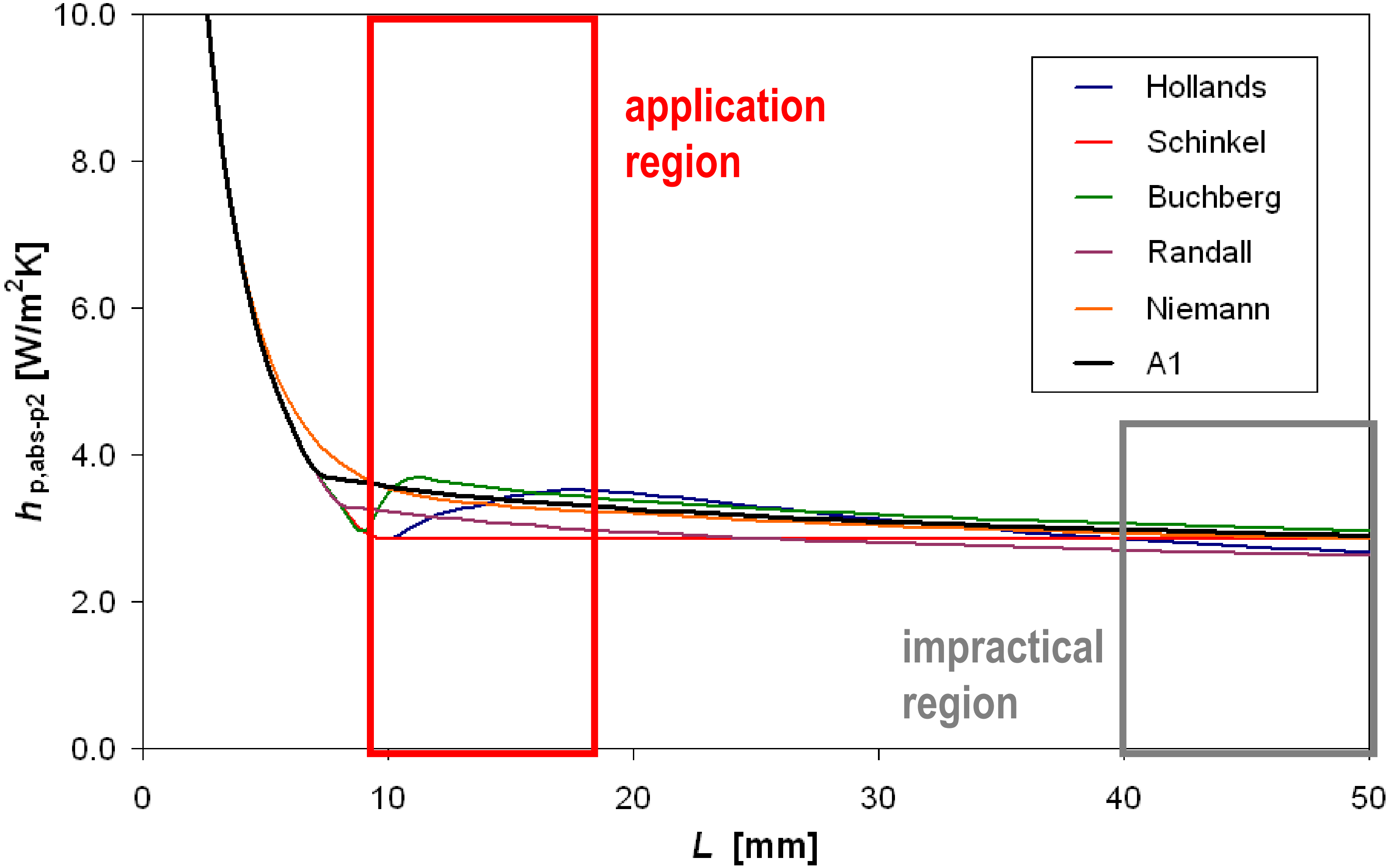
Niemann:

$$\mathrm{Nu}_L = 1 + \frac{m(\mathrm{Ra}_L)^K}{\mathrm{Ra}_L + n}$$

for more see
Reference handbook on web

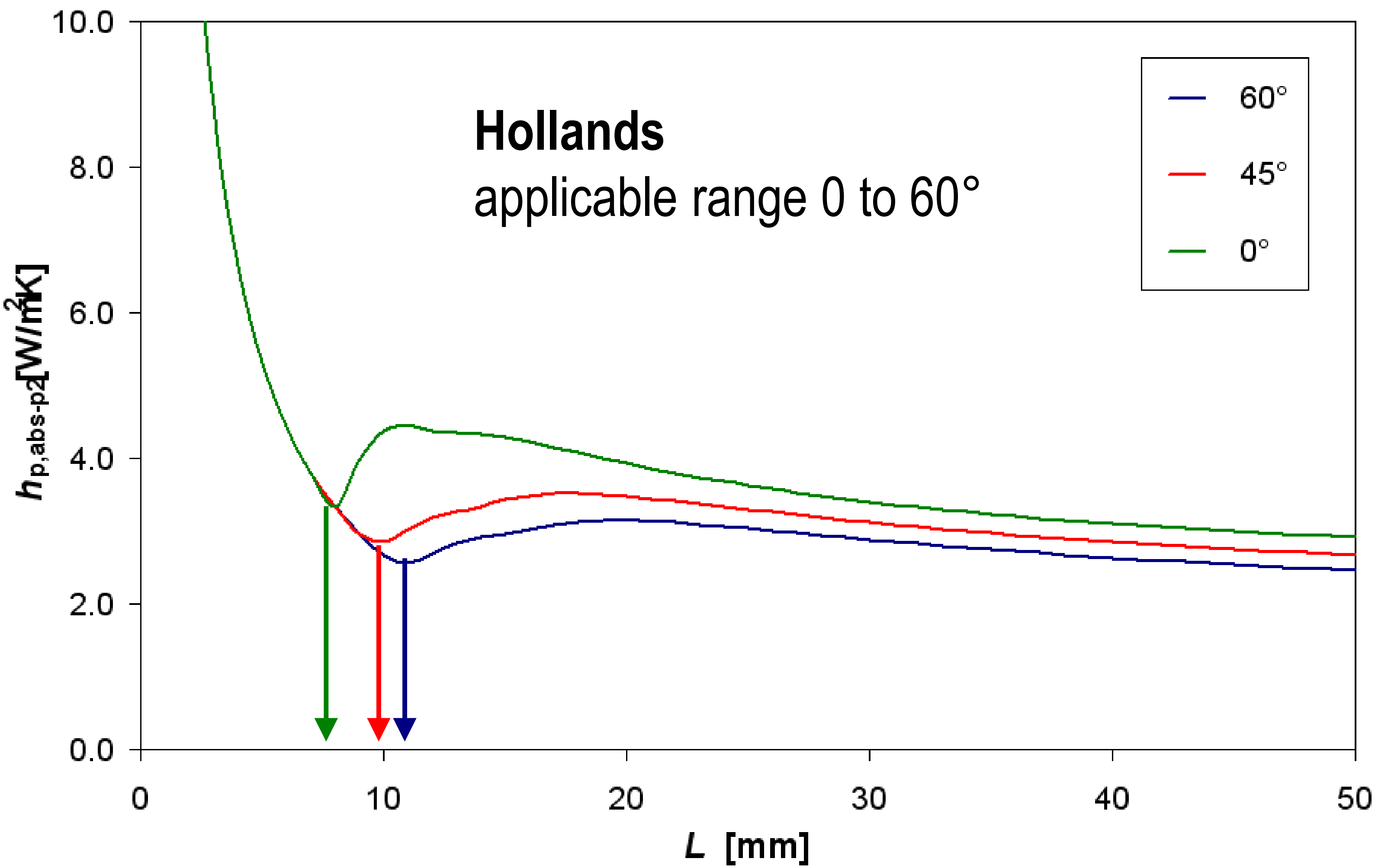
Convection in the air gap – optimum *L*

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



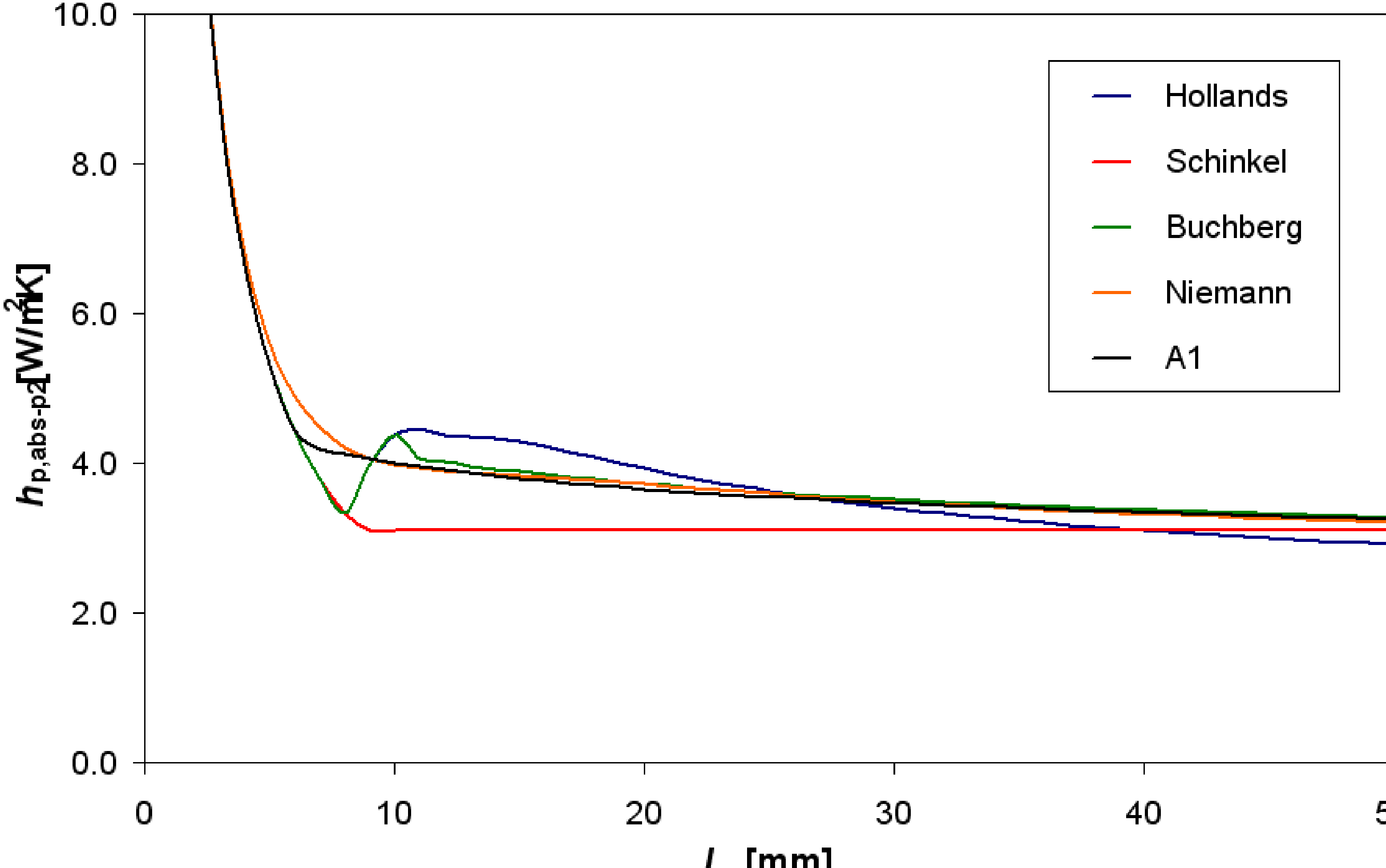
Slope dependence

absorber 60 °C, glass 20 °C



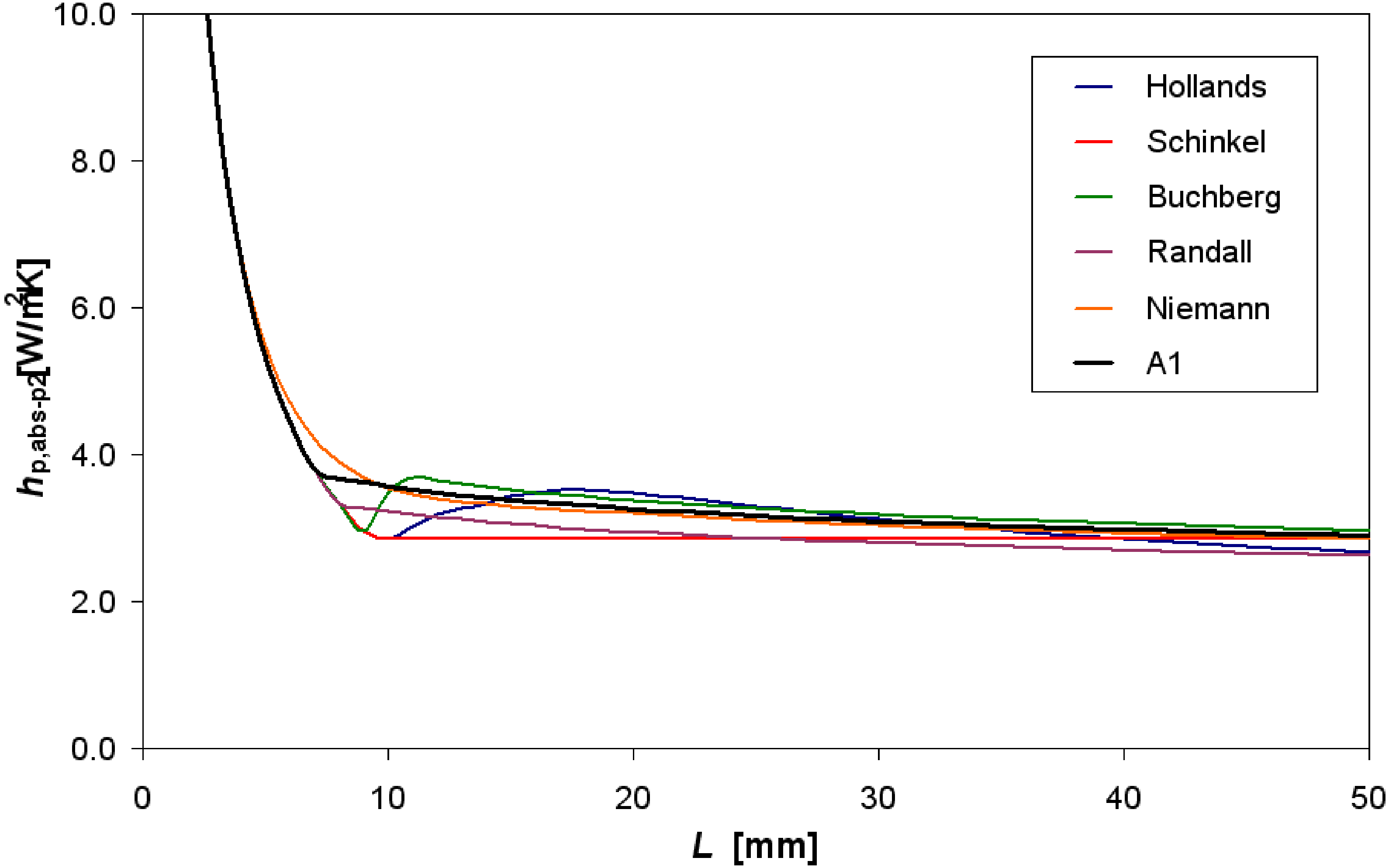
Slope dependence 0°

absorber 60 °C, glass 20 °C



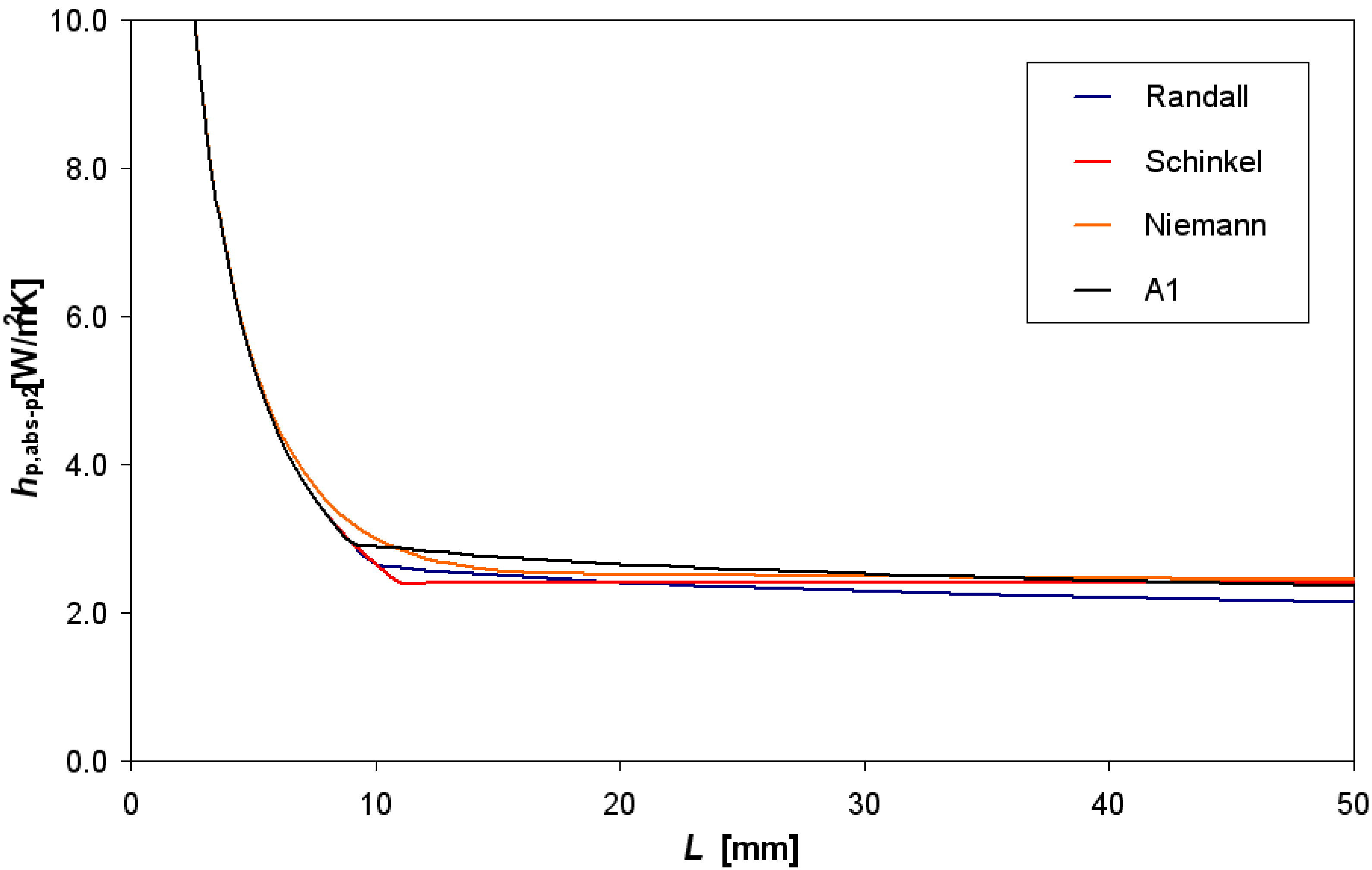
Slope dependence 45°

absorber 60 °C, glass 20 °C

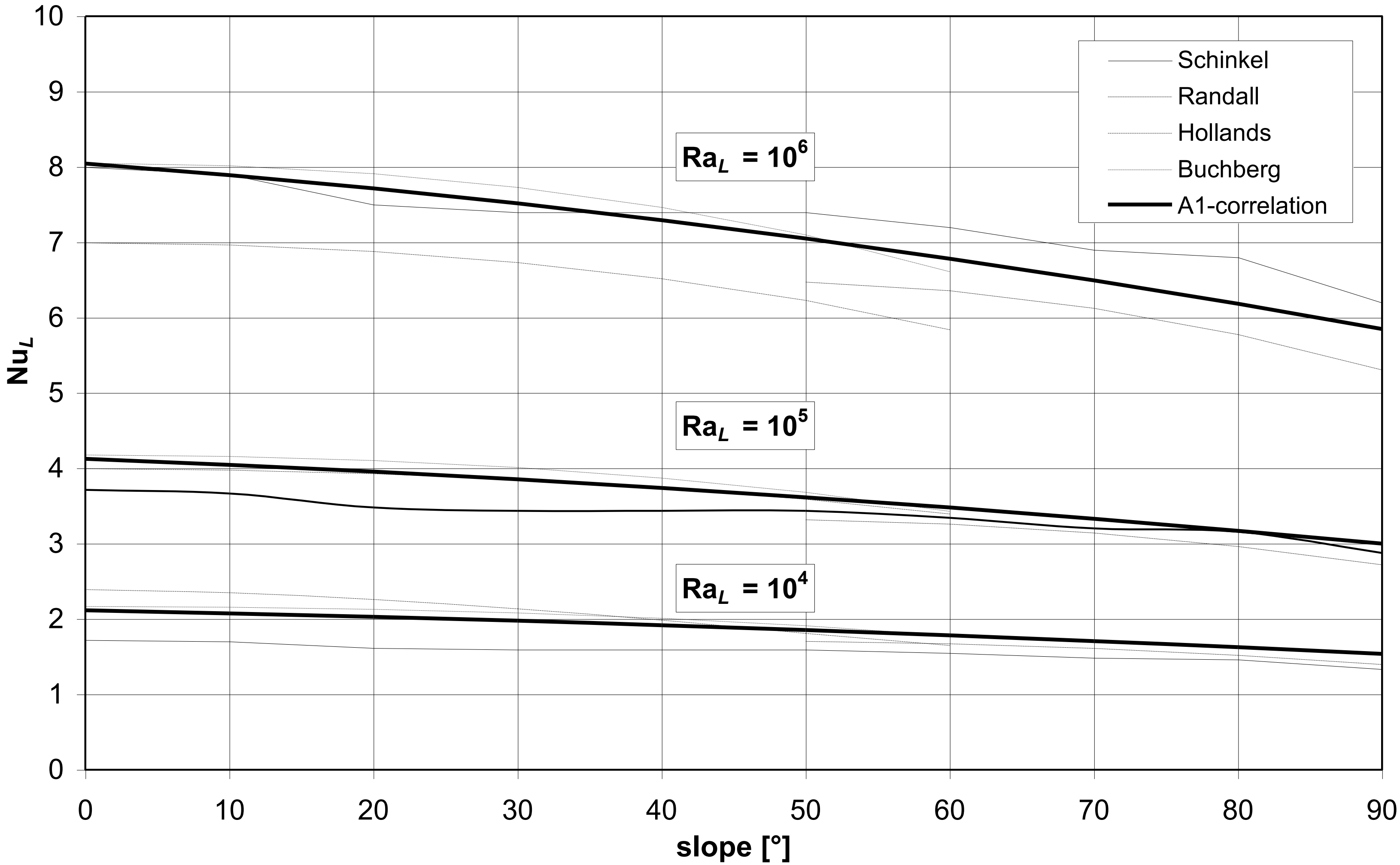


Slope dependence 90°

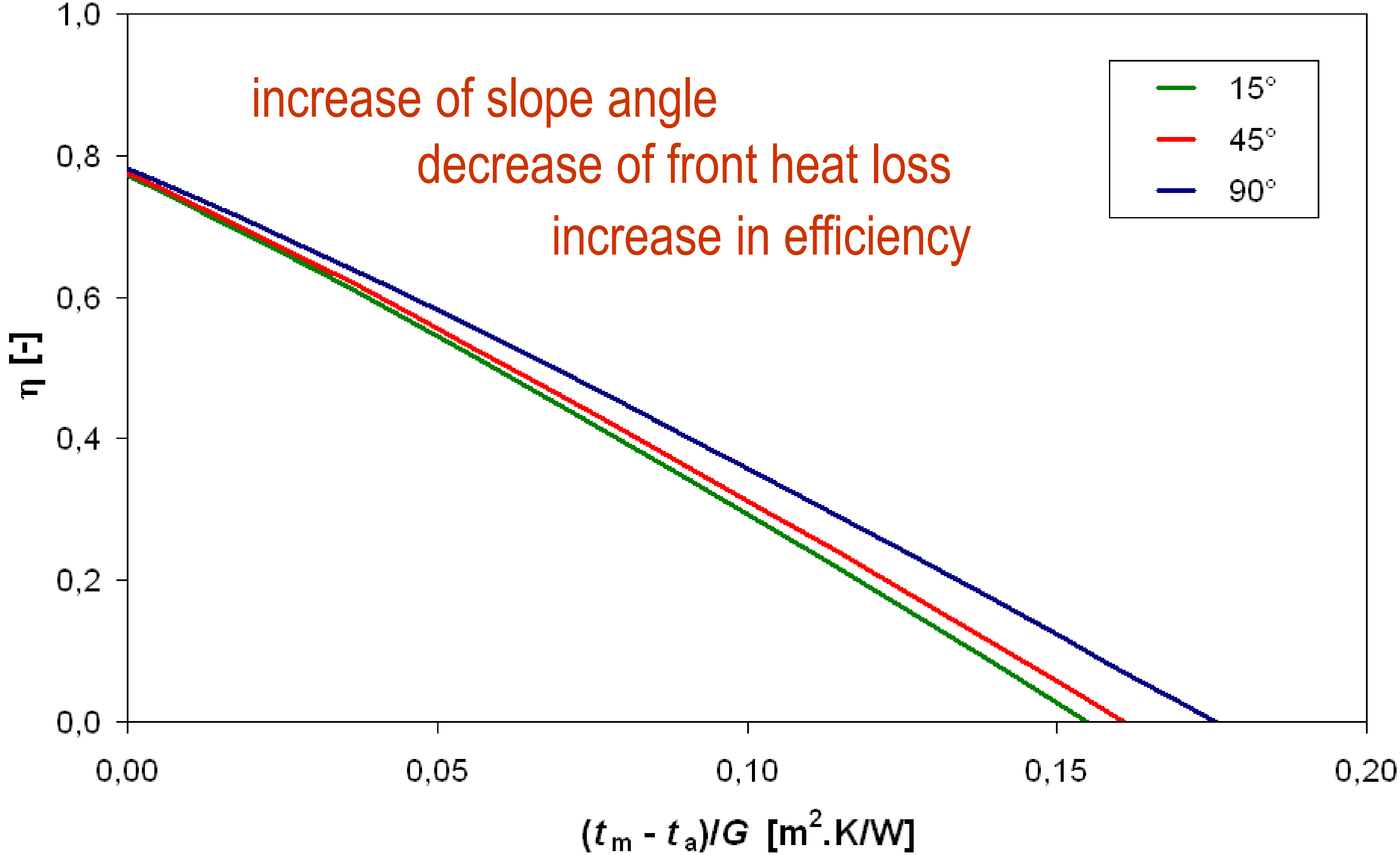
absorber 60 °C, glass 20 °C



Convection in inclined air layer

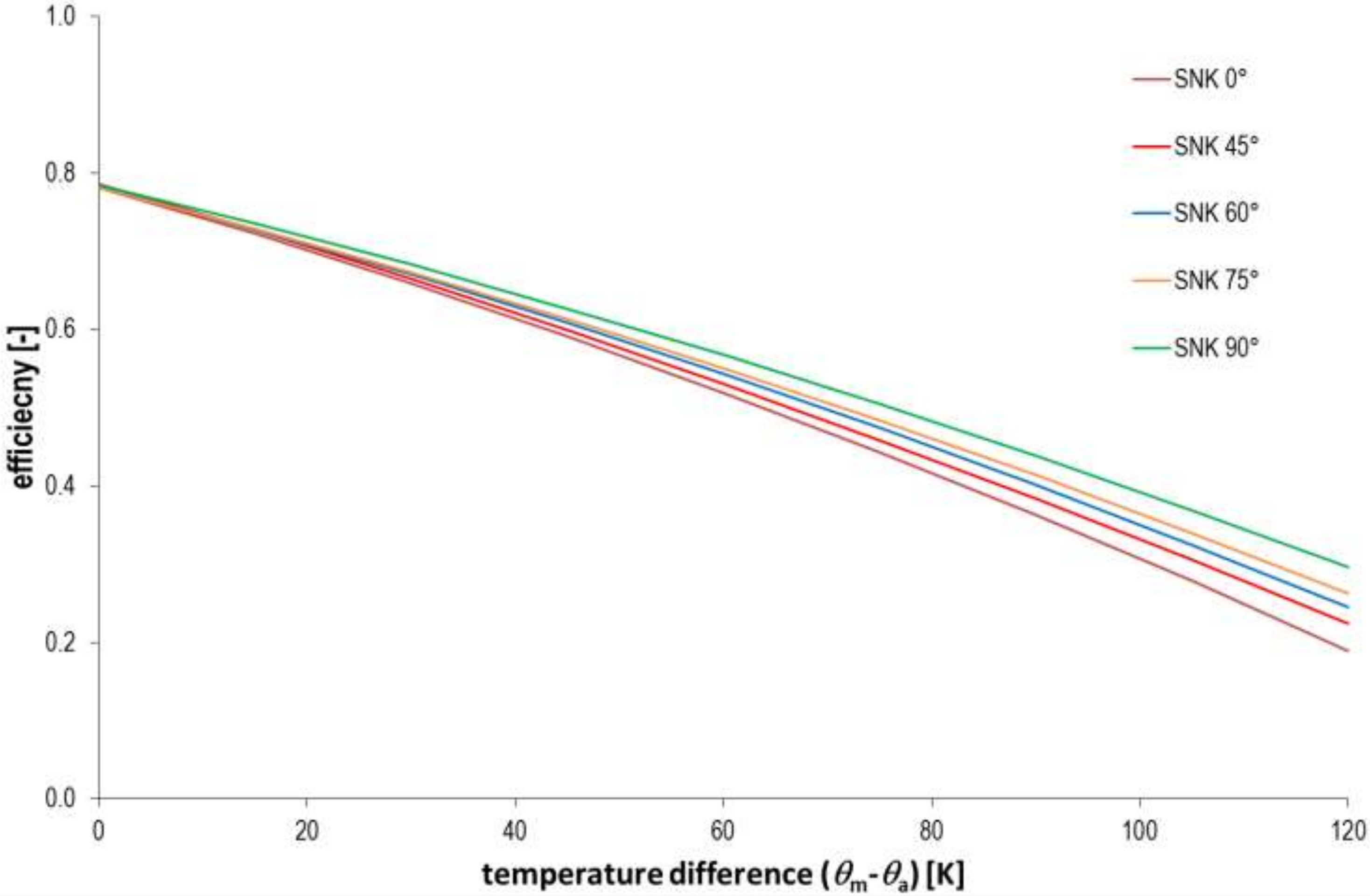


Slope impact on collector efficiency



Slope impact on collector efficiency

results from testing at cca 1000 W/m^2

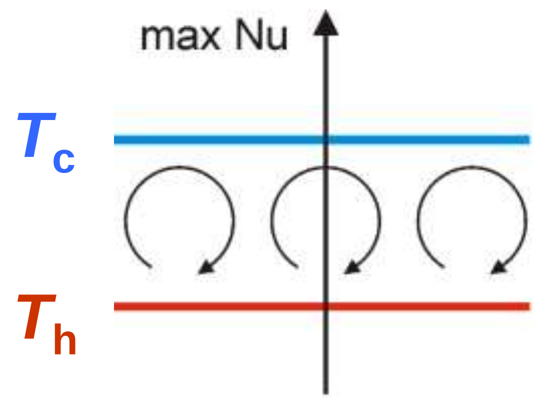


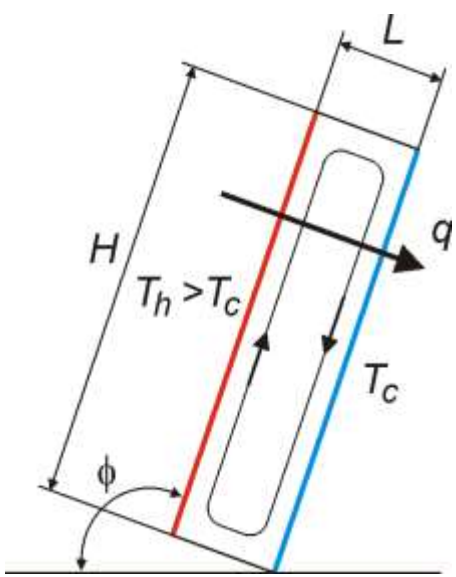
Convection in the back air gap

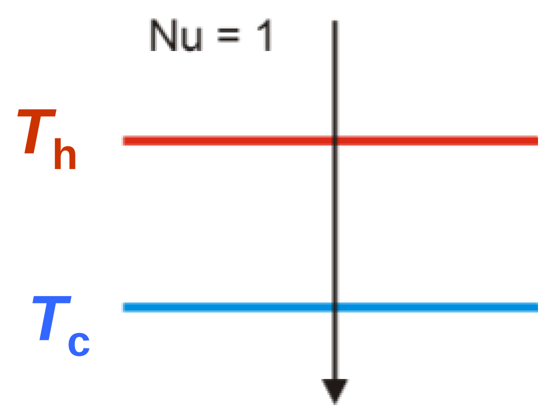
in the range 90 < ϕ < 180°

$$\mathrm{Nu}_L = 1 + [\mathrm{Nu}_L(\phi = 90°) - 1]\sin\phi$$

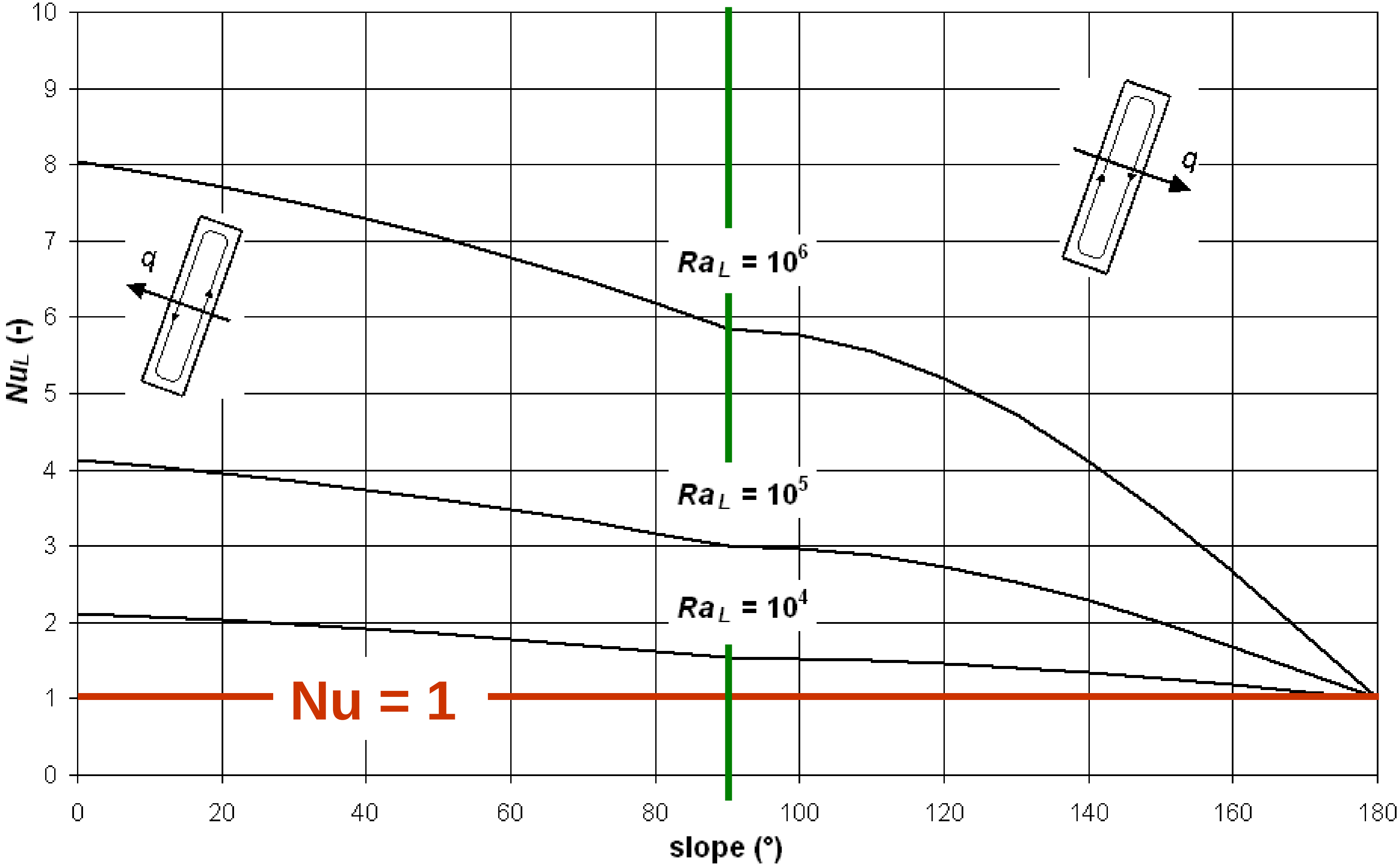
Arnold (1975)







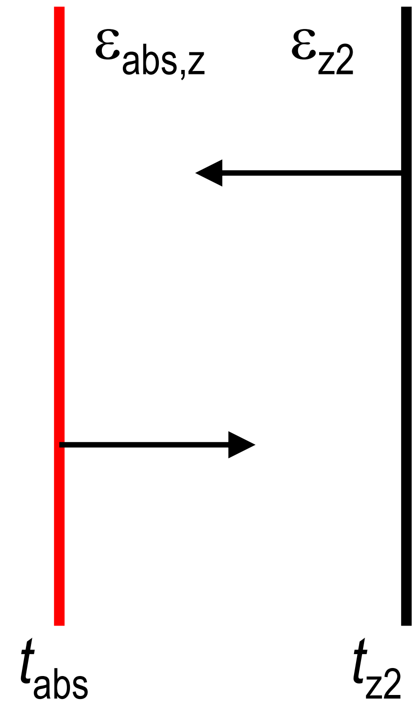
Convection vs. slope



Radiation between absorber and frame

Radiation heat exchange

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



<u>treatment of insulations:</u>

aluminium foil
applied to mineral wool

$\varepsilon_{z2} = 0.1$

is it needed?

Radiation between absorber and frame

<u>example:</u> internal frame insulation surface emittance ε_{z2}

t_{abs} = 60 °C, t_a = 20 °C, back insulation 30 mm, air gap d_{z2}

case 1: d_{z2} = 20 mm

$\varepsilon_{z2} = 0.5$ U_z = 1.10 W/m²K

15 %

$\varepsilon_{z2} = 0.1$ U_z = 0.96 W/m²K

case 2: d_{z2} = 5 mm

$\varepsilon_{z2} = 0.5$ U_z = 1.22 W/m²K

4 %

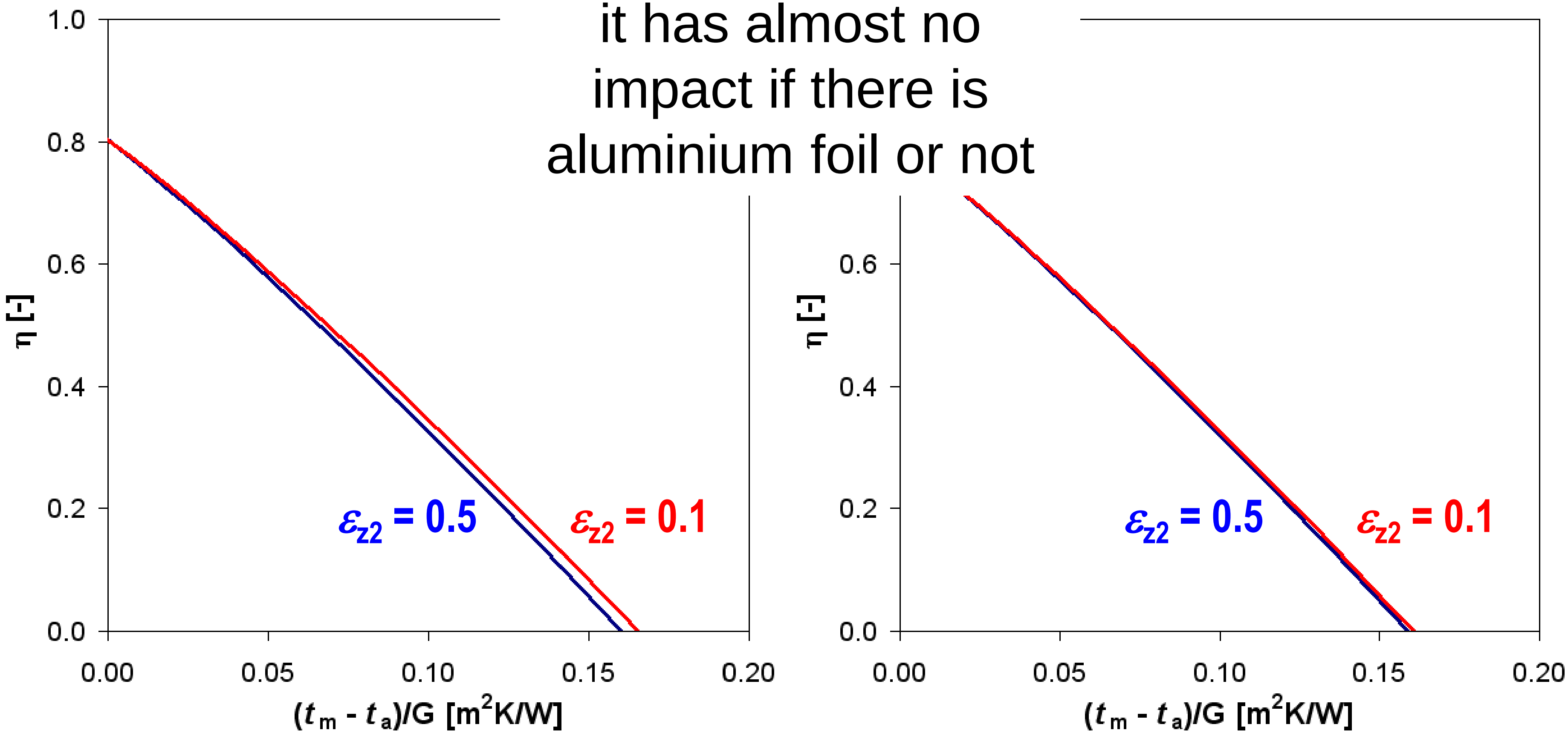
$\varepsilon_{z2} = 0.1$ U_z = 1.17 W/m²K

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

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

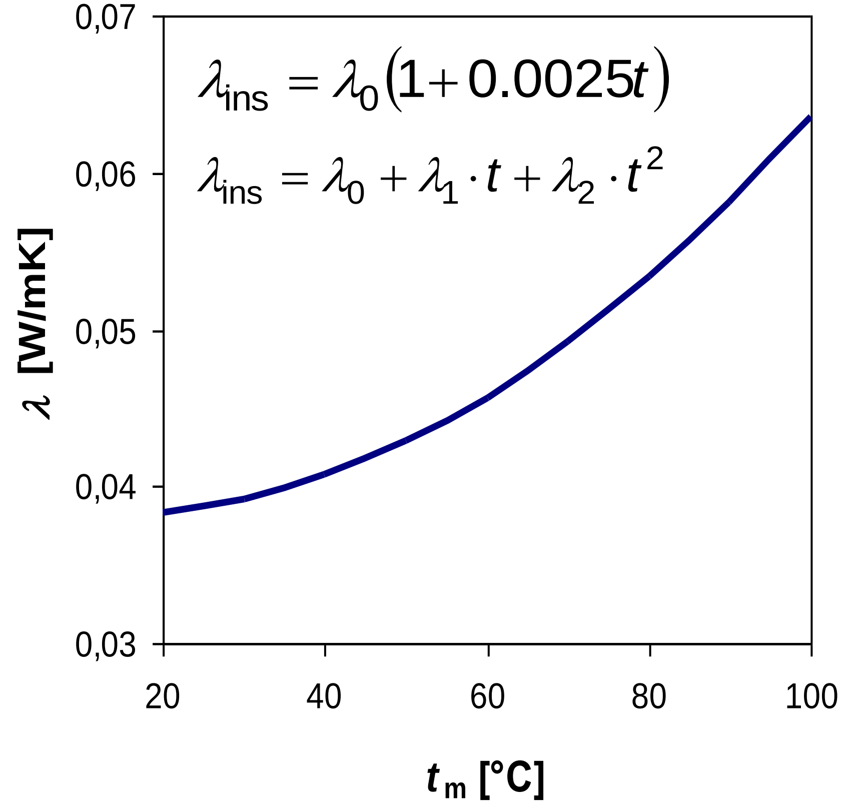
<u>insulation – thermal conductivity:</u>

mineral wool: λ = 0.045 W/mK

PUR foam: λ = 0.035 W/mK

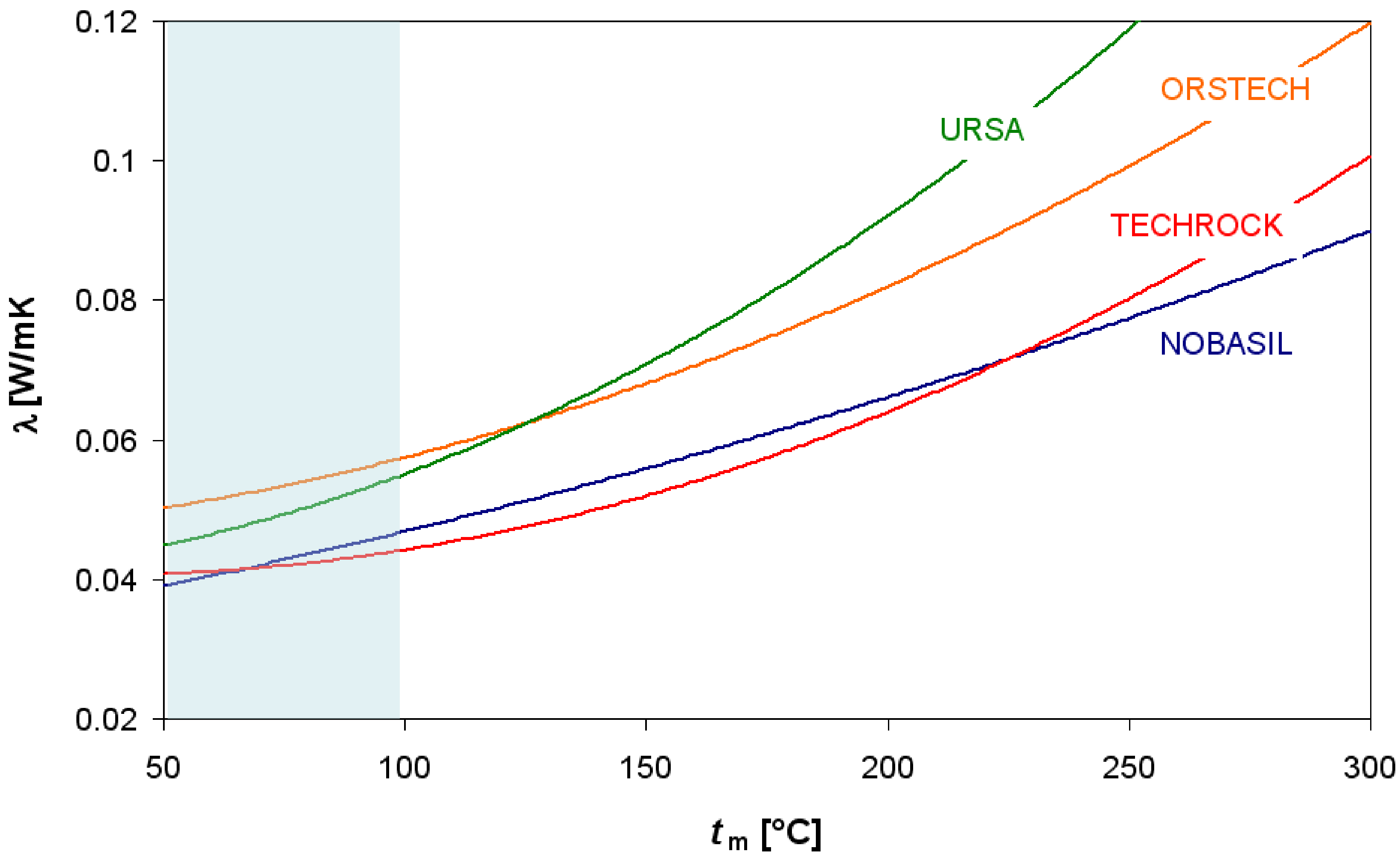
polystyren: weak resistance to thermal load

thermal conductivity $\lambda = \mathrm{f}\,(t)$



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

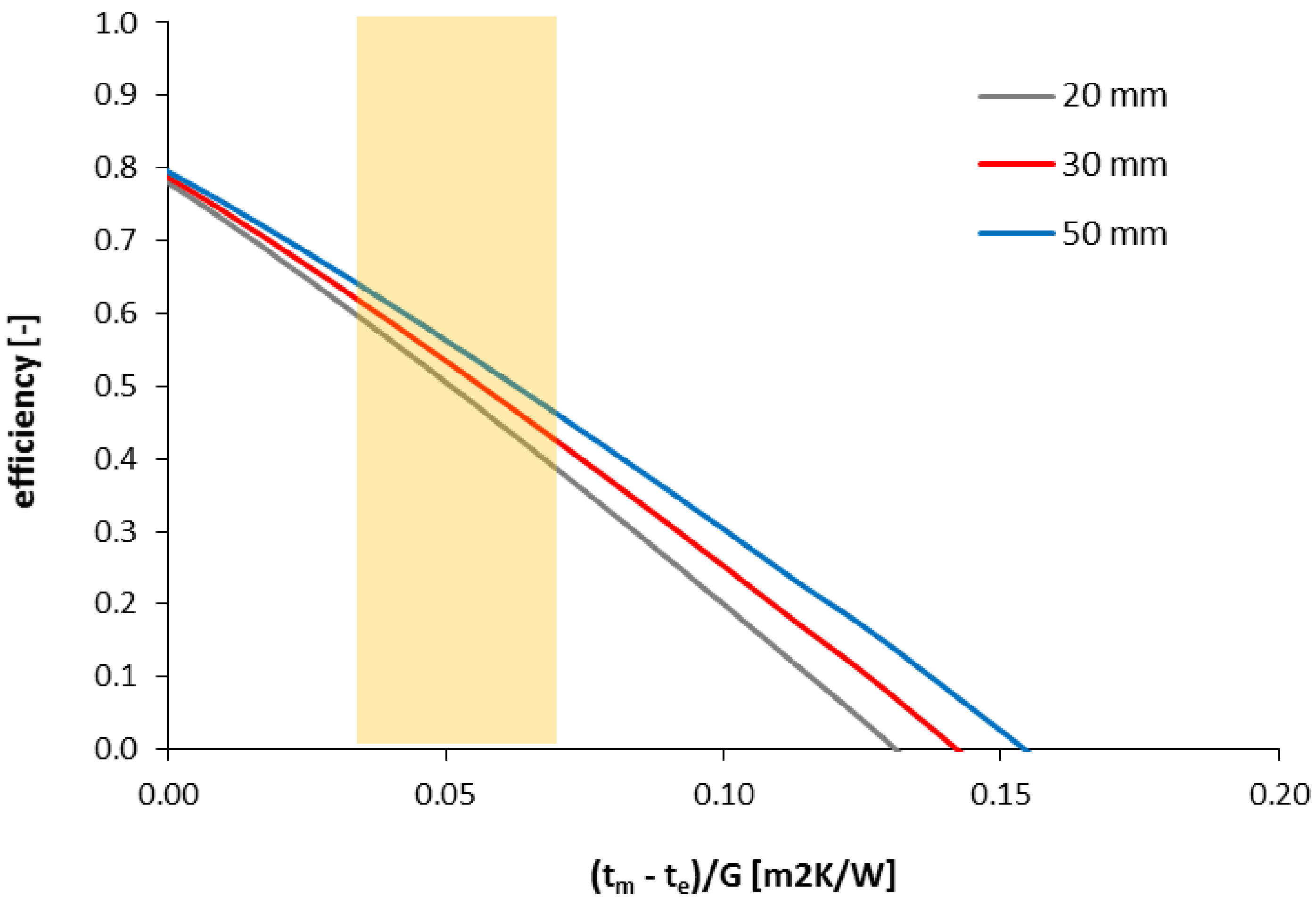
Insulation thickness in collector

- 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



Insulation thickness in collector

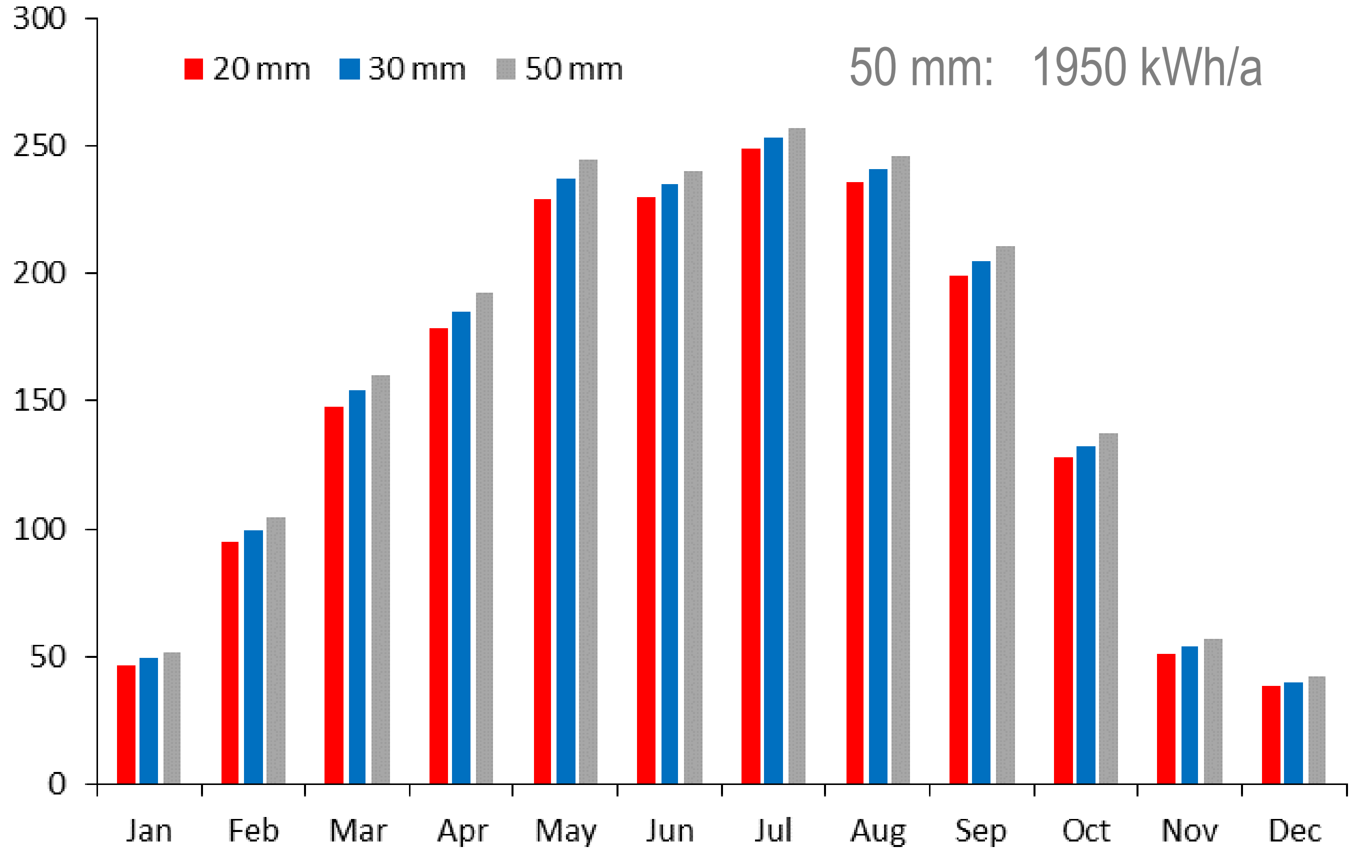
- **how can the absorber emissivity influence the annual energy yields of collector?**
  - 20 mm $\eta_0 = 0.782$ $a_1 = 5.272$ $a_2 = 0.007$
  - 30 mm $\eta_0 = 0.789$ $a_1 = 4.857$ $a_2 = 0.006$
  - 50 mm $\eta_0 = 0.796$ $a_1 = 4.451$ $a_2 = 0.006$
- open studio SDHW-SOLNET in TRNSYS
- make a change 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 %



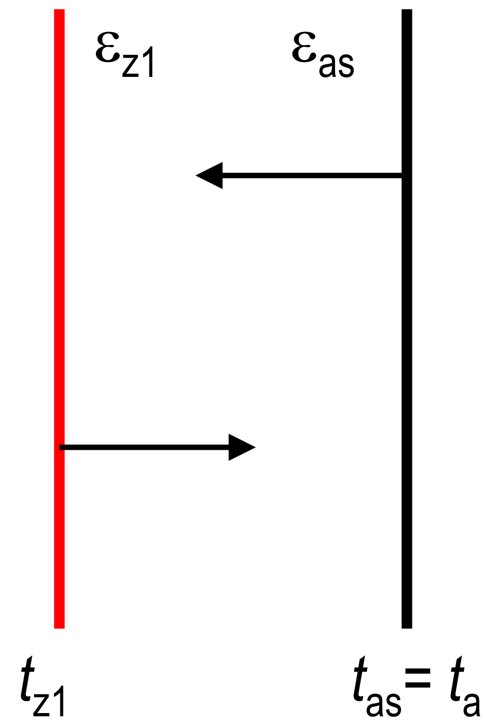
Insulation thickness in collector

- 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

$$q_{s,z1\text{-}as} = \sigma \frac{T_{z1}^4 - T_{as}^4}{\dfrac{1}{\varepsilon_{z1}} + \dfrac{1}{\varepsilon_{as}} - 1} \qquad [\mathrm{W/m^2}]$$



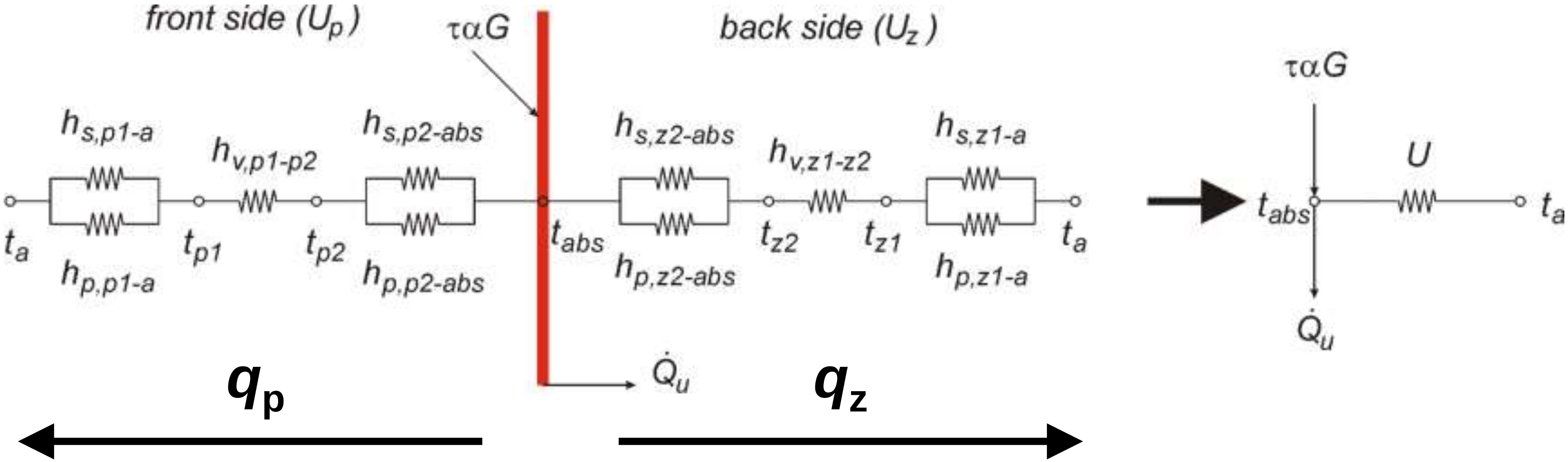
<u>any treatment of external frame surface is useless</u>

wind convection

frame insulation

supress any influence

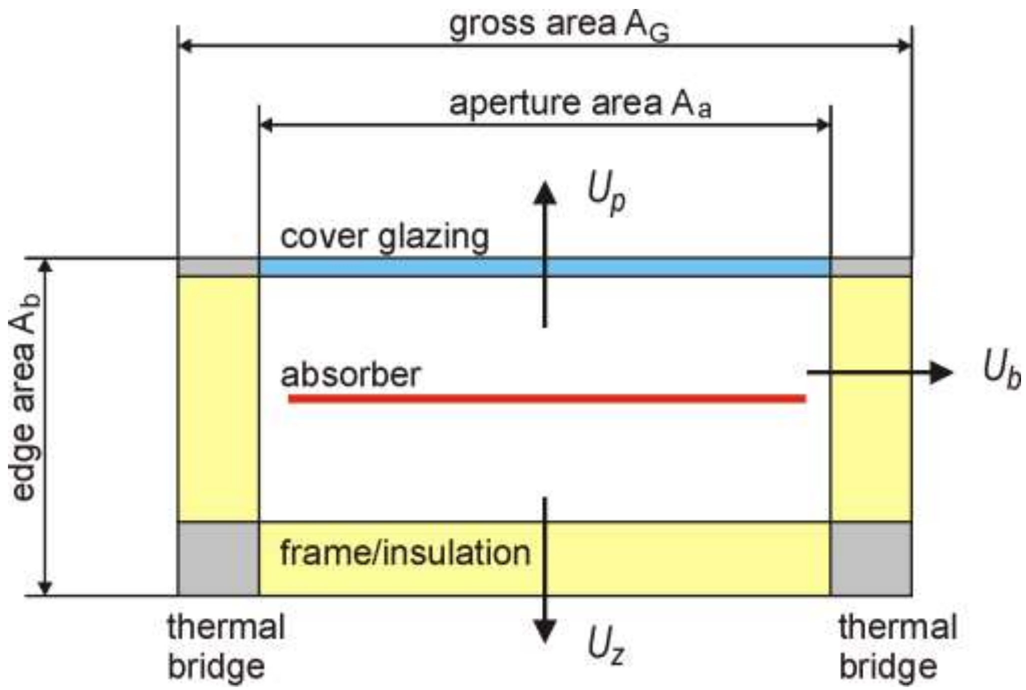
Collector heat loss coefficient: *U*-value



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

$$U_z = \frac{1}{\frac{1}{h_{s,z1-a} + h_{p,z1-a}} + \frac{1}{h_{v,z1-z2}} + \frac{1}{h_{s,abs-z2} + h_{p,abs-z2}}}$$

Collector heat loss coefficient: U-value



$$\dot{Q}_{l,t} = U_G A_G \left(t_{abs} - t_a\right)$$

$$\dot{Q}_{l,t} = U_a A_a \left(t_{abs} - t_a\right)$$

$$U_G A_G = U_p A_G + U_z A_G + U_b A_b = U_a A_a$$

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

for calculations:

***U*-value** related to aperture area $\boldsymbol{A_a}$

Iterations for temperatures

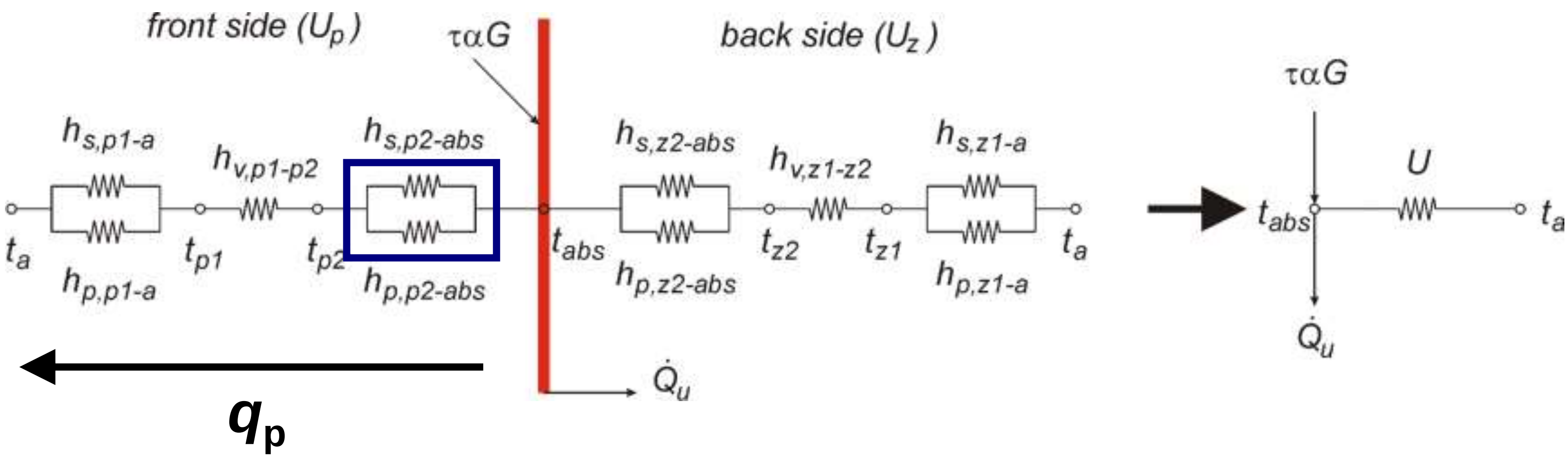
calculation of heat transfer coefficients:

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

<u>example of iterative determination for front heat flow:</u>

1. estimate: $t_{abs} = t_{in} + \Delta t$
2. estimate: t_{p1}, t_{p2} $\quad t_{p2} = t_{abs} - \frac{t_{abs} - t_a}{3} \qquad t_{p1} = t_a + \frac{t_{abs} - t_a}{3}$
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



$$q_p = U_p(t_{abs} - t_a)$$

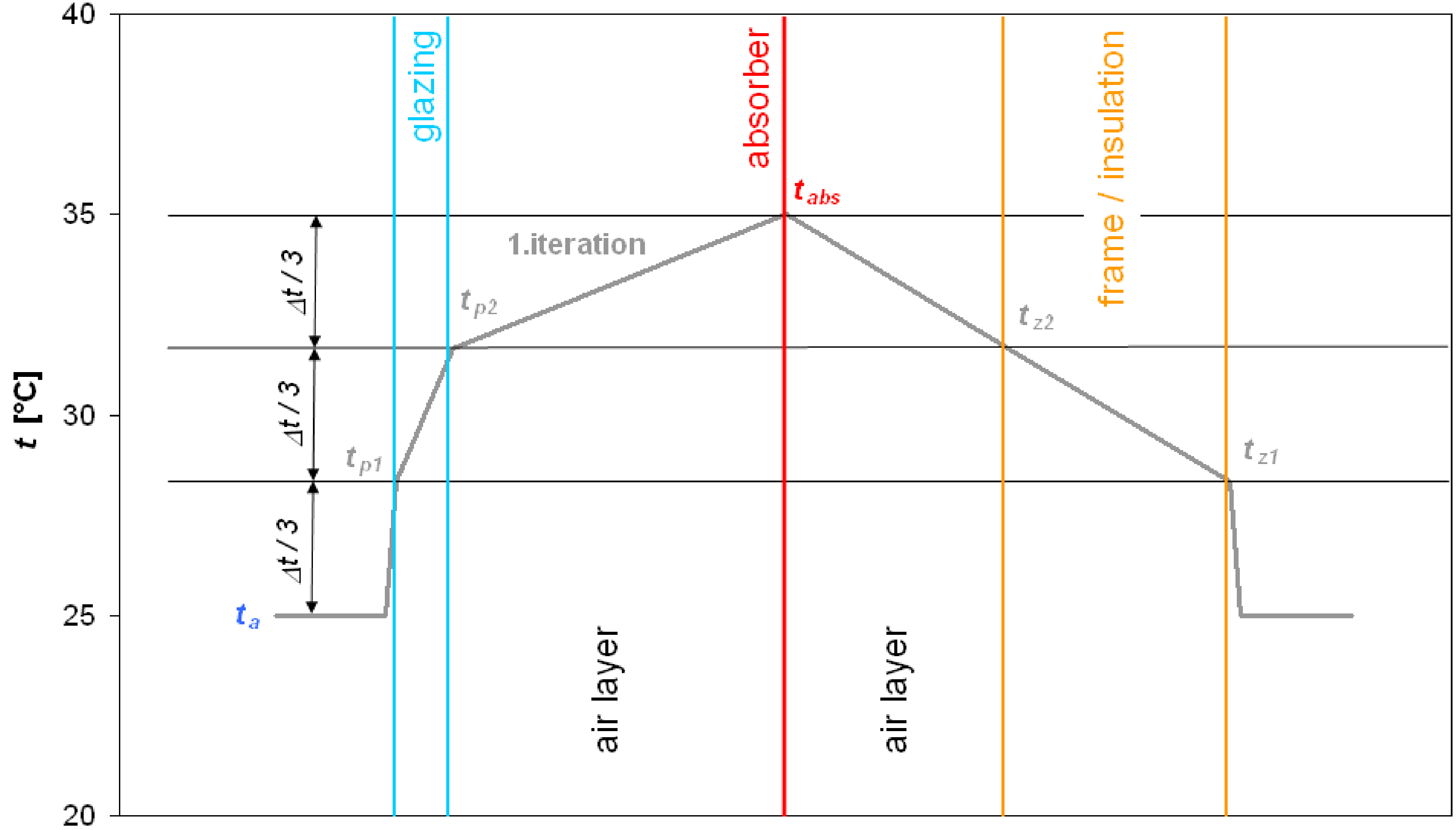
$$\boxed{q_p = (h_{s,abs-p1} + h_{p,abs-p1})(t_{abs} - t_{p1})}$$

$$q_p = h_{v,p1-p2}(t_{p1} - t_{p2})$$

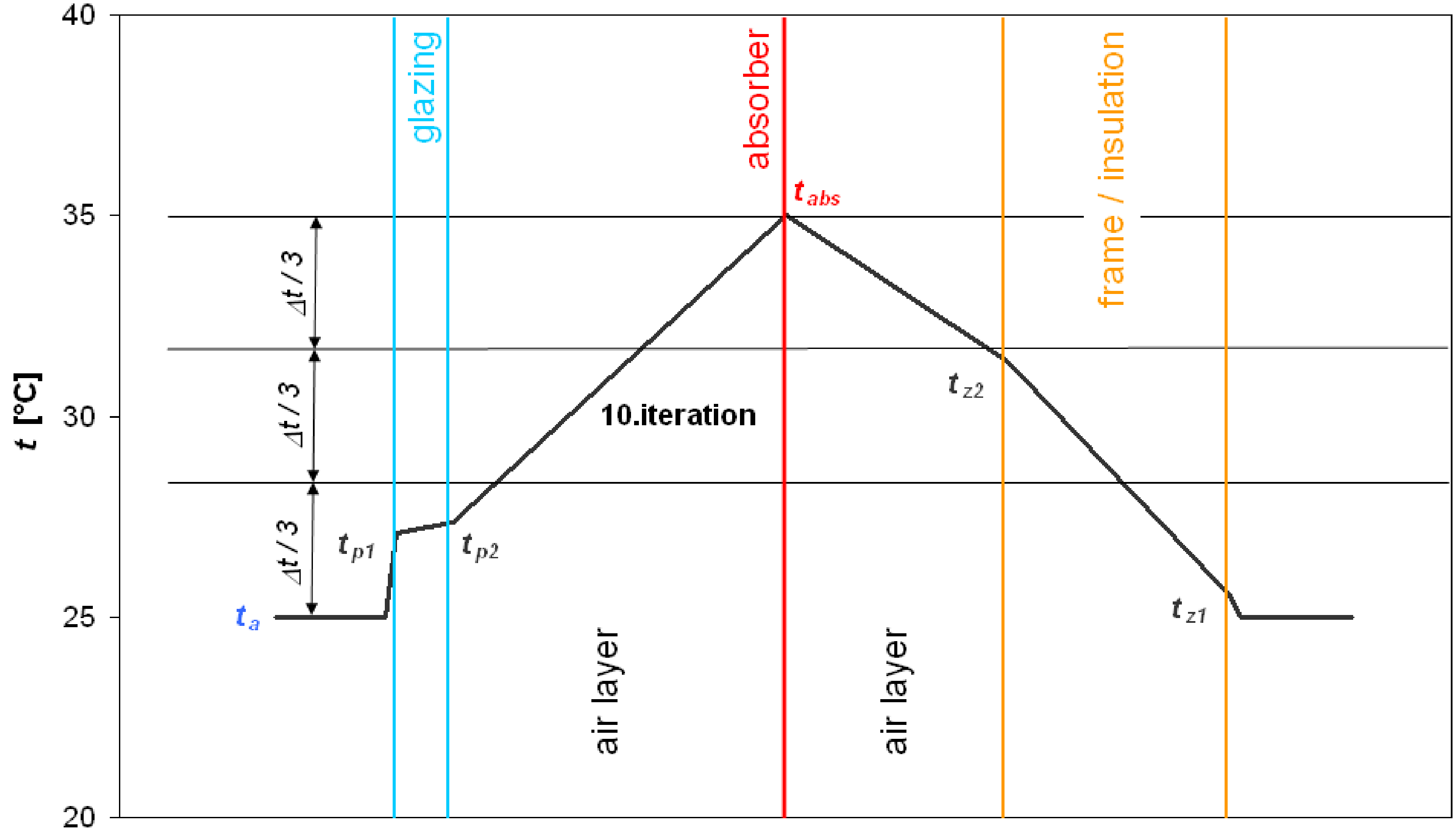
$$q_p = (h_{s,p2-a} + h_{p,p2-a})(t_{p2} - t_a)$$

temperature difference is proportional to heat resistance

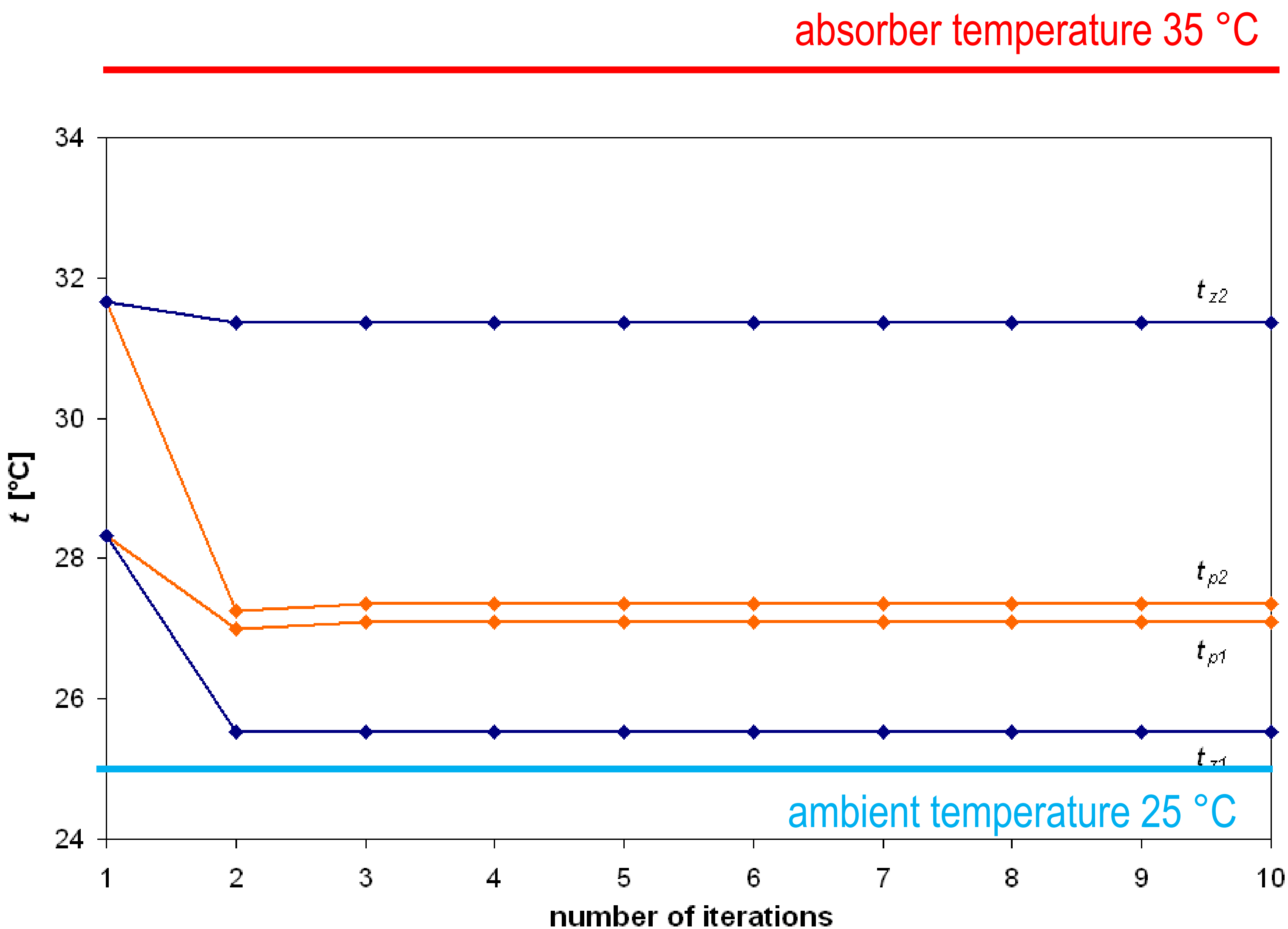
1. estimation of temperatures



Last iteration



Convergence of calculations



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

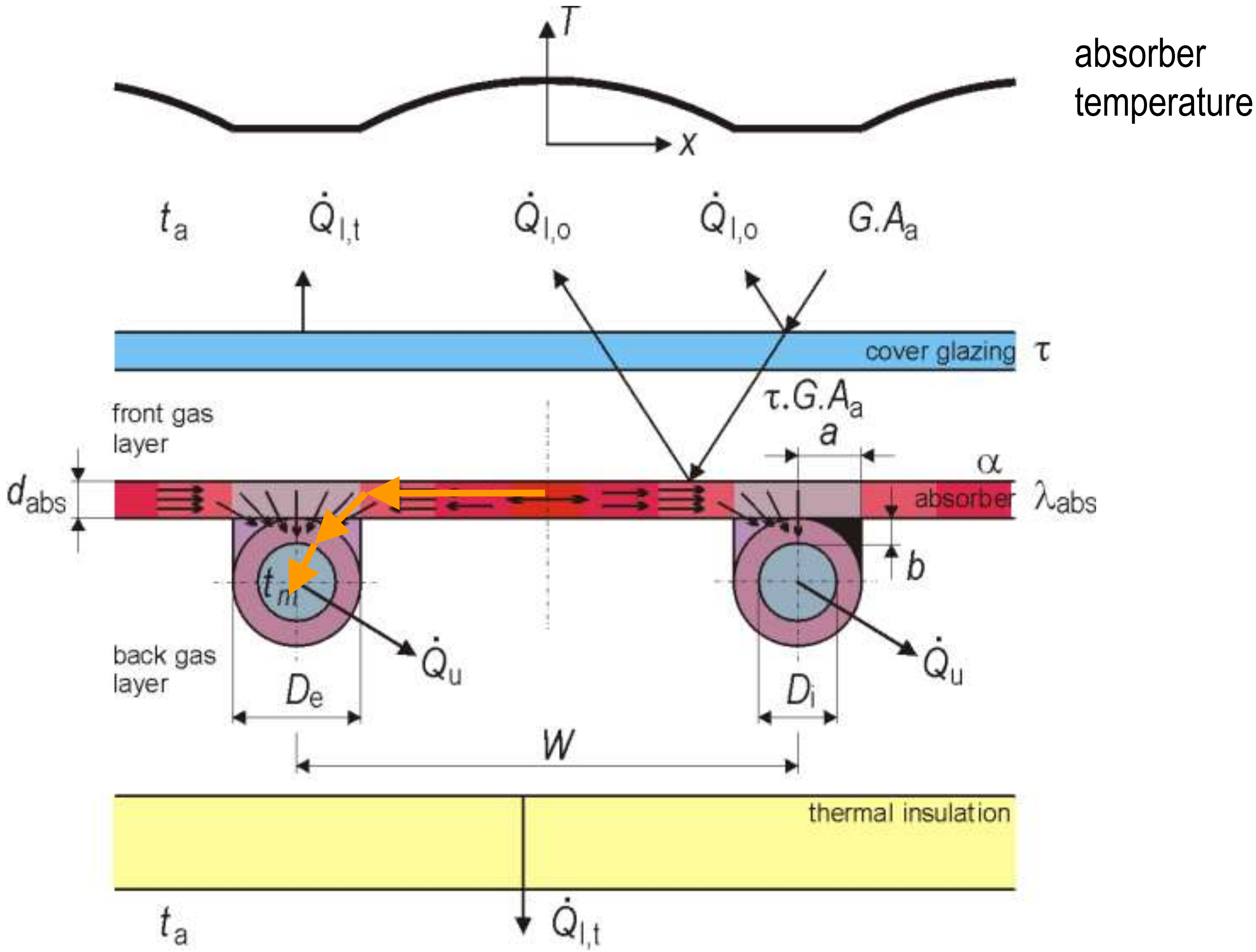
Czech Technical University in Prague

Solar collector energy balance

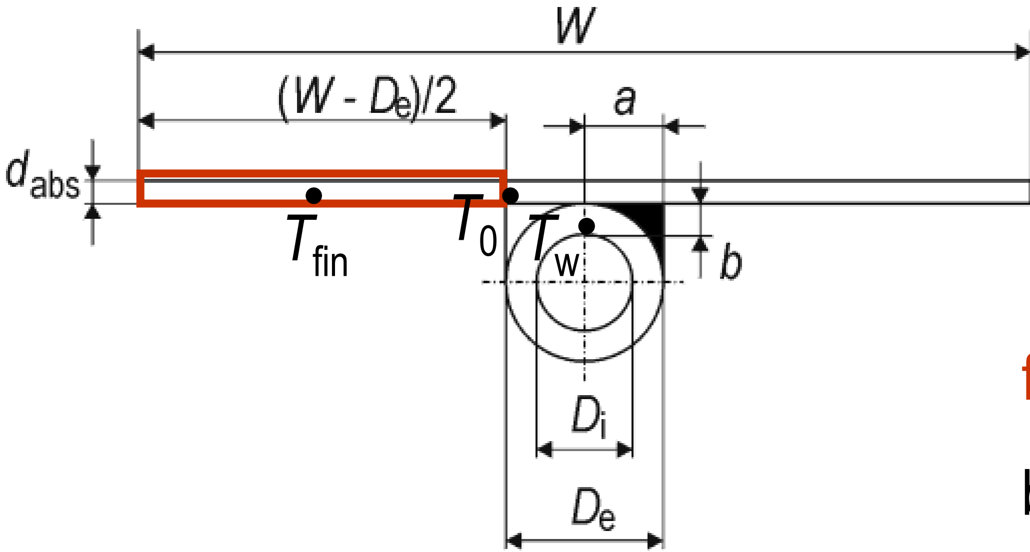
- **External energy balance of absorber**
  - heat flow from absorber surface to ambient environment
  - heat losses
  - quality of solar collector envelope

- **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



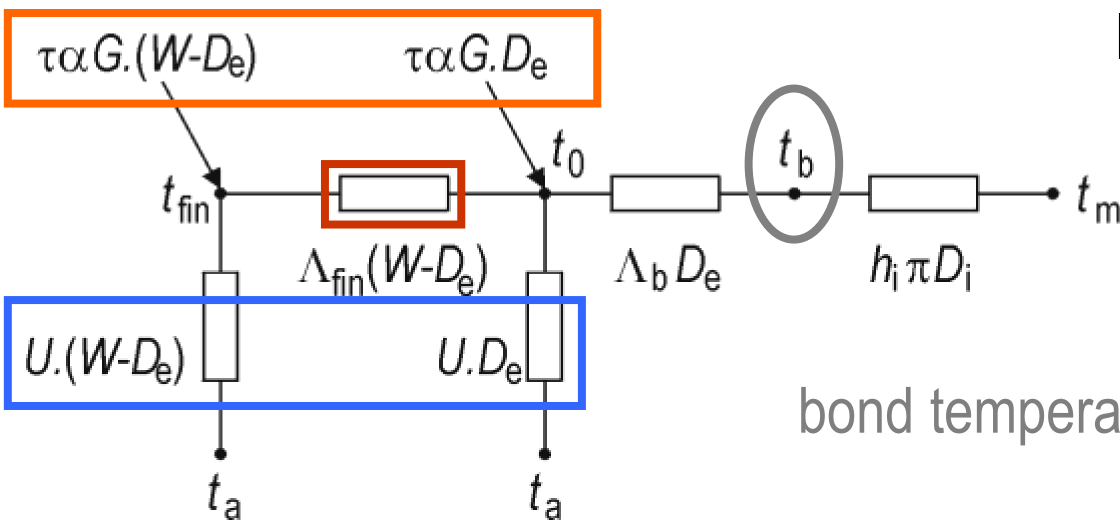
width W = distance between riser pipes

$D_e = 2a = 2 \cdot$ bond width

fin heat transfer Λ_{fin}

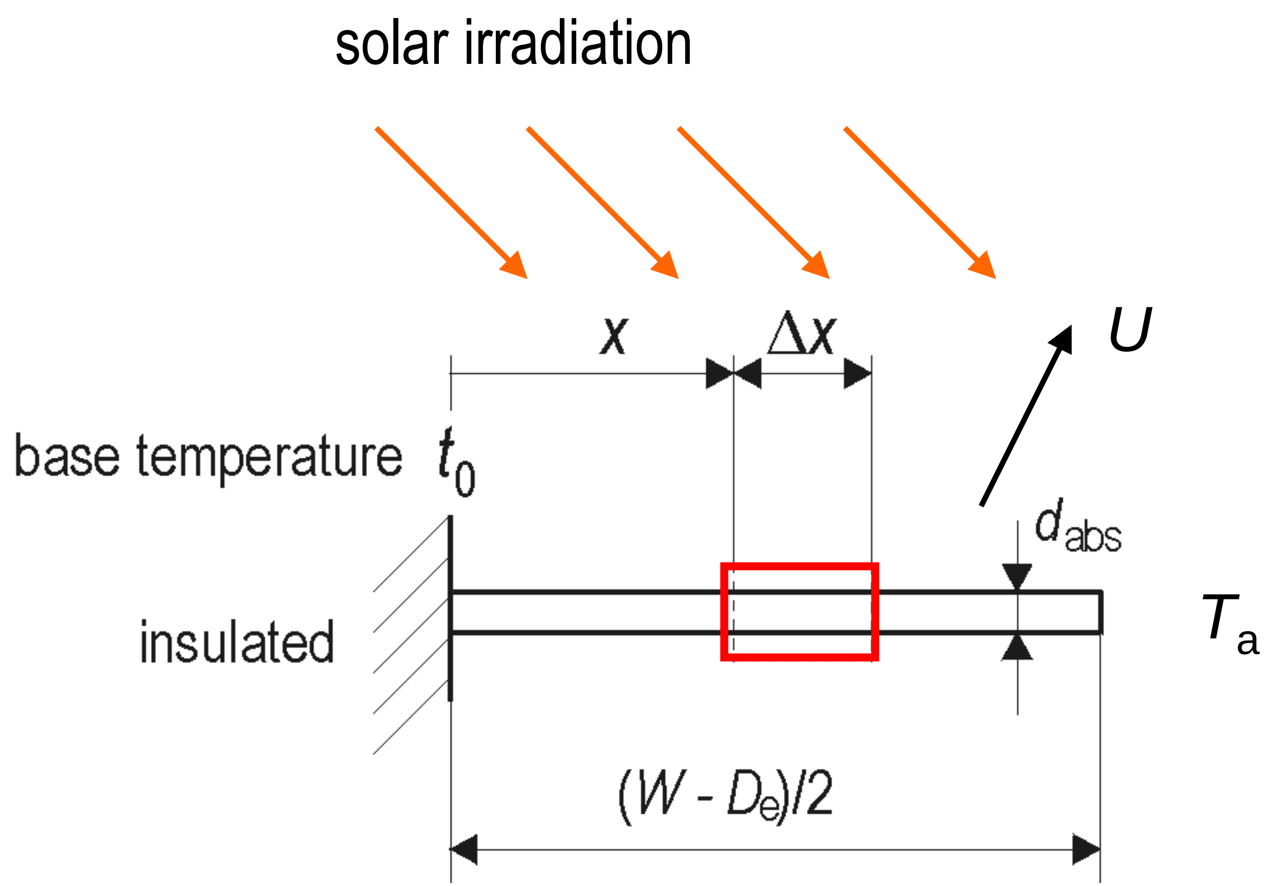
bond heat transfer Λ_b

pipe-fluid heat transfer $h_i \, \pi \, D_i$



bond temperature $t_b = t_w$ (wall temperature)

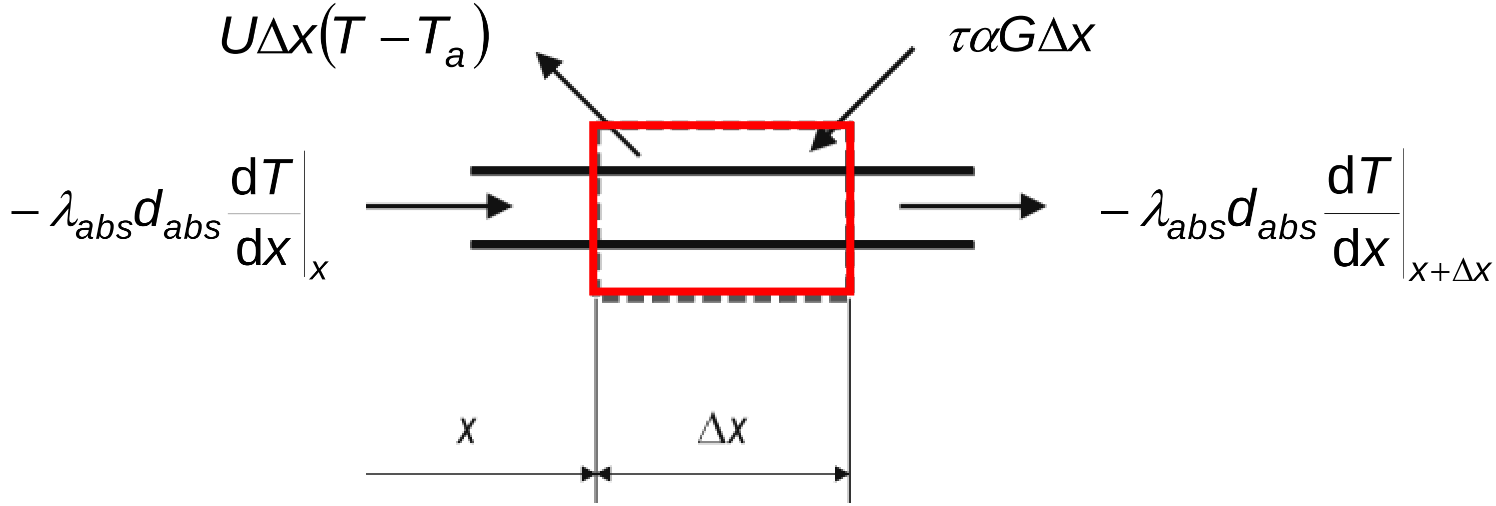
Energy balance of absorber fin



case of standard fin efficiency concept but with solar irradiation

known from air-water heat exchangers theory

Energy balance of absorber fin element



derivation of temperature distribution between tubes (in absorber fin)

$$\tau\alpha G\Delta x - U\Delta x(T - T_a) + \left.\left(-\lambda_{abs} d_{abs} \frac{\mathrm{d}T}{\mathrm{d}x}\right)\right|_x - \left.\left(-\lambda_{abs} d_{abs} \frac{\mathrm{d}T}{\mathrm{d}x}\right)\right|_{x+\Delta x} = 0$$

yields in solution of differential equation of 2nd order

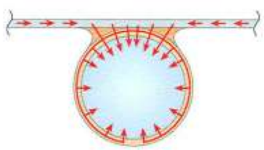
$$\frac{d^2T}{dx^2} = \frac{U}{\lambda_{abs} d_{abs}}\left(T - T_a - \frac{S}{U}\right)$$

Fin efficiency *F*

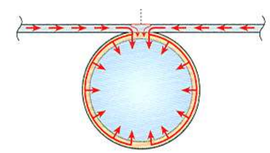
standard fin efficiency
(rectangular profile)

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

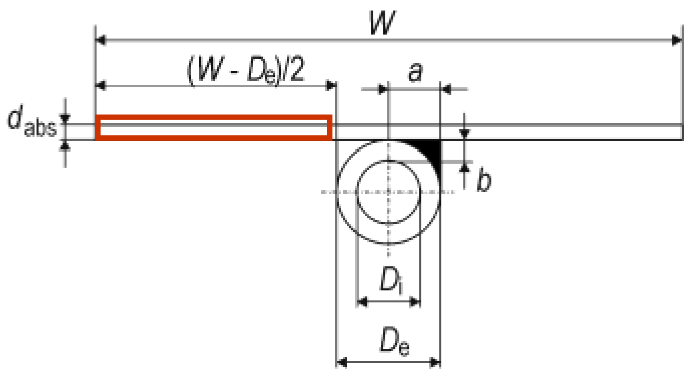
$(W - D_e)/2$ or $(W - 2a)/2$... active fin length

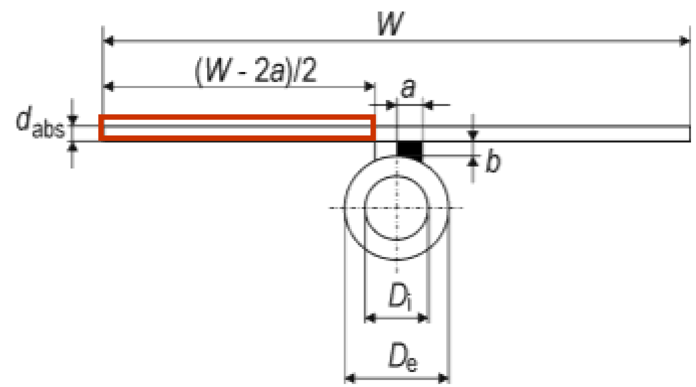


soldering



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 improportional

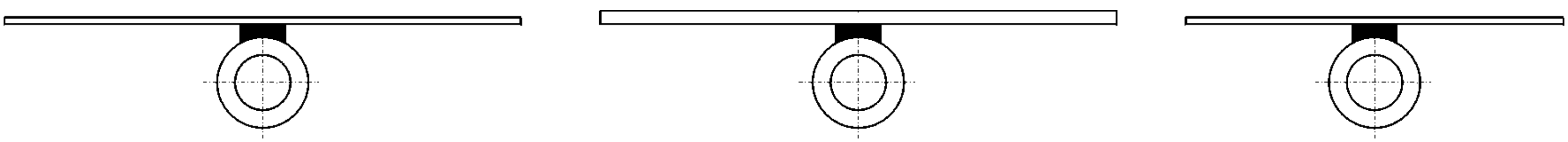
higher fin efficiency = higher heat removal from absorber

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 $\boldsymbol{U}$ **= 4 W/m²K**



aluminium (240 W/mK)

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

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

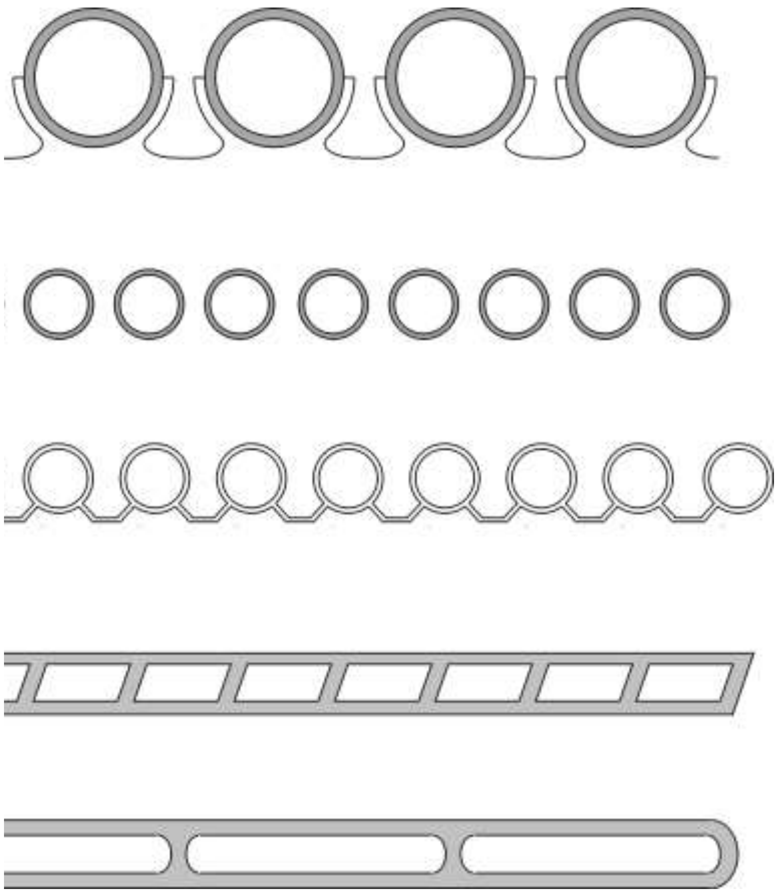
steel (80 W/mK)

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

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

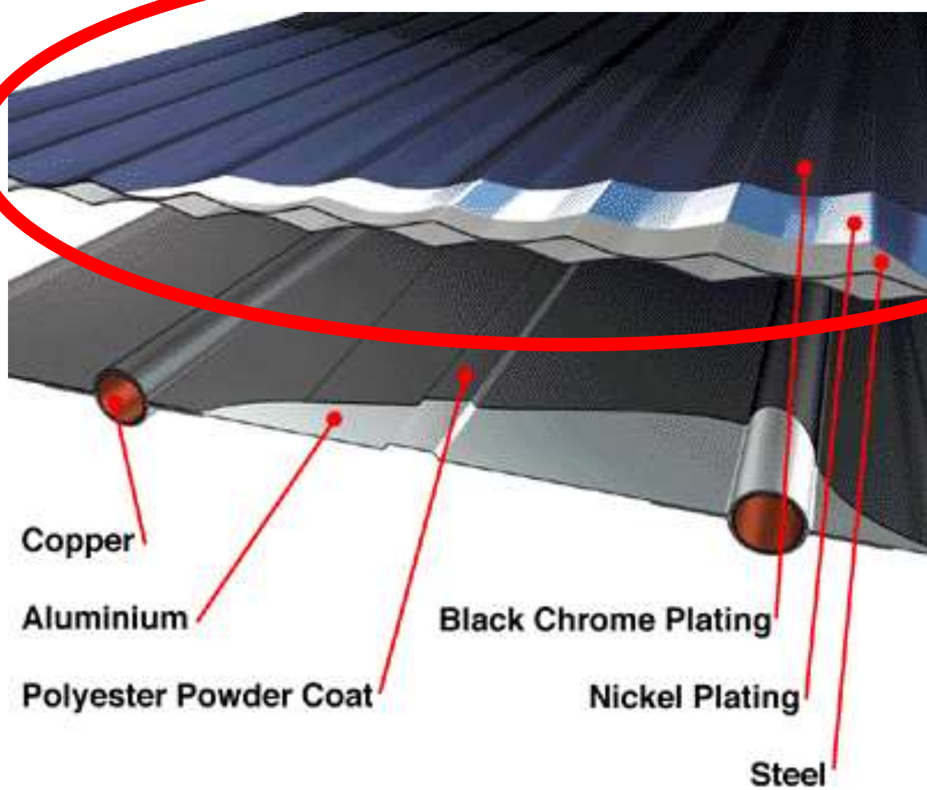
EPDM (0.14 W/mK) d_{abs} = 2.0 mm, W^* = 6 mm

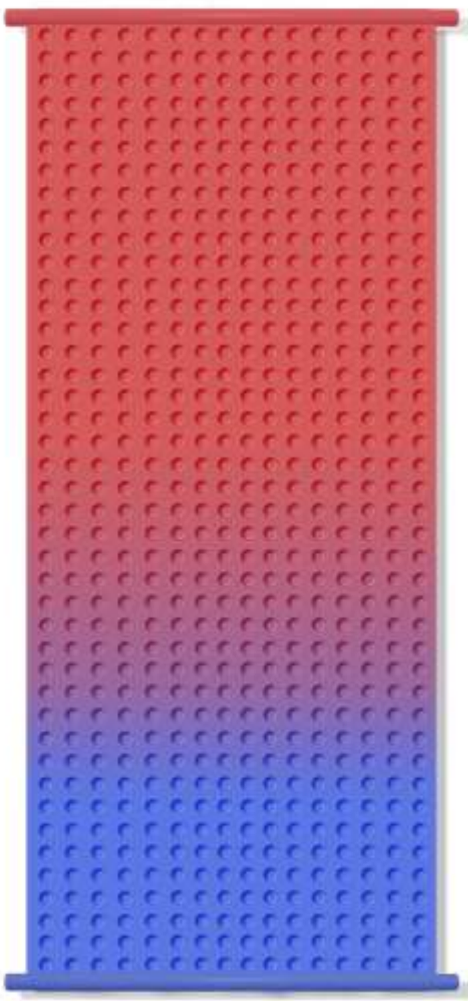
Plastic absorbers



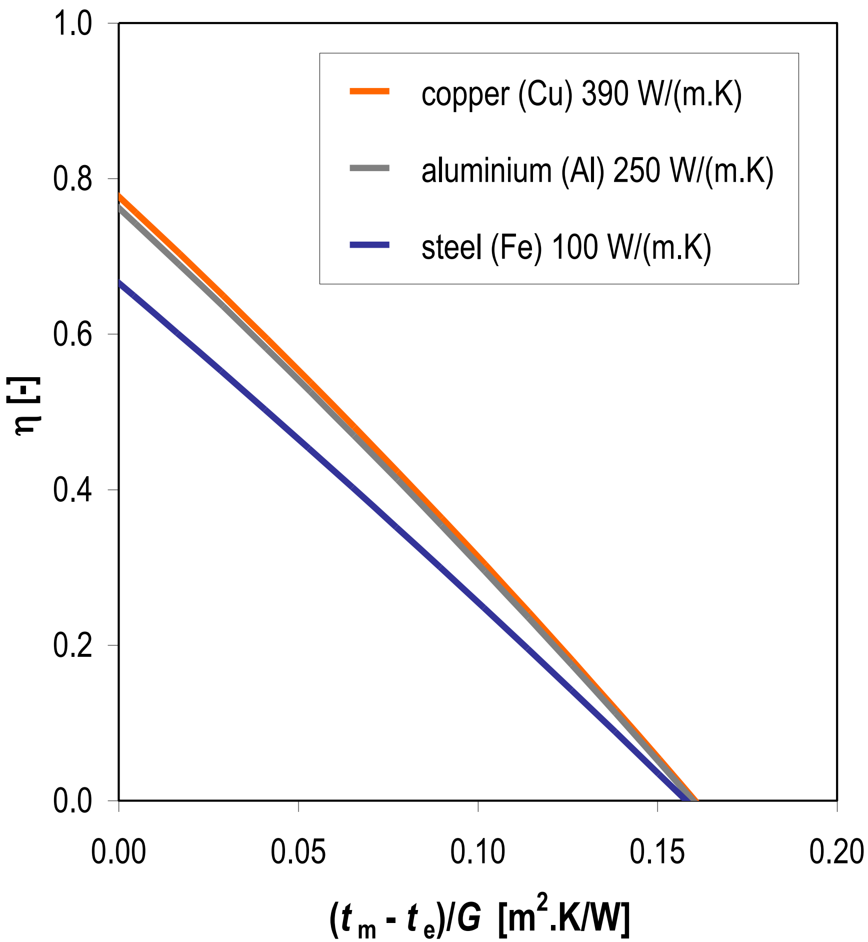
Best absorbers – fully wetted metal sheet

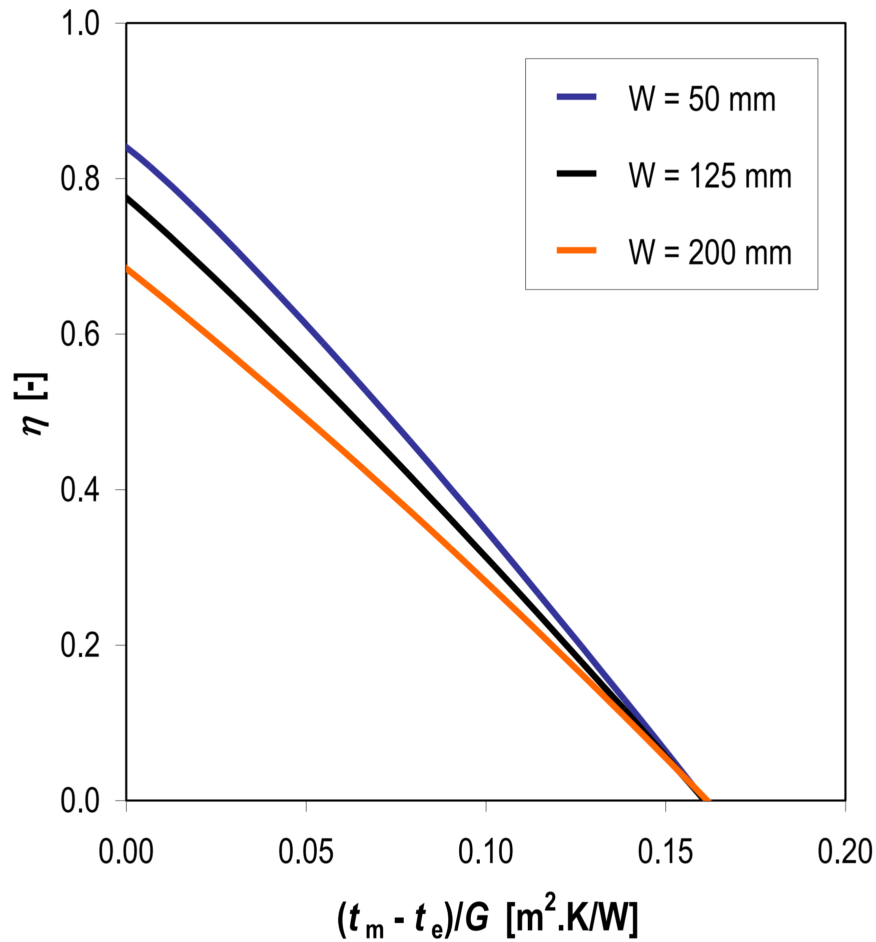
$F' = 1.0$





Influence of material and geometry





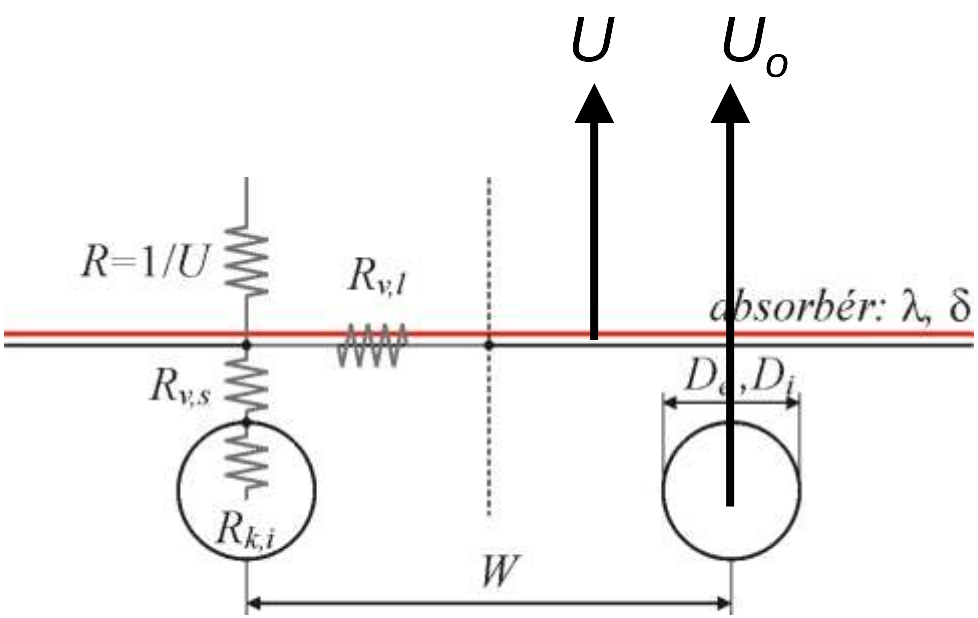
Efficiency factor *F'*

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

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

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

collector efficiency

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

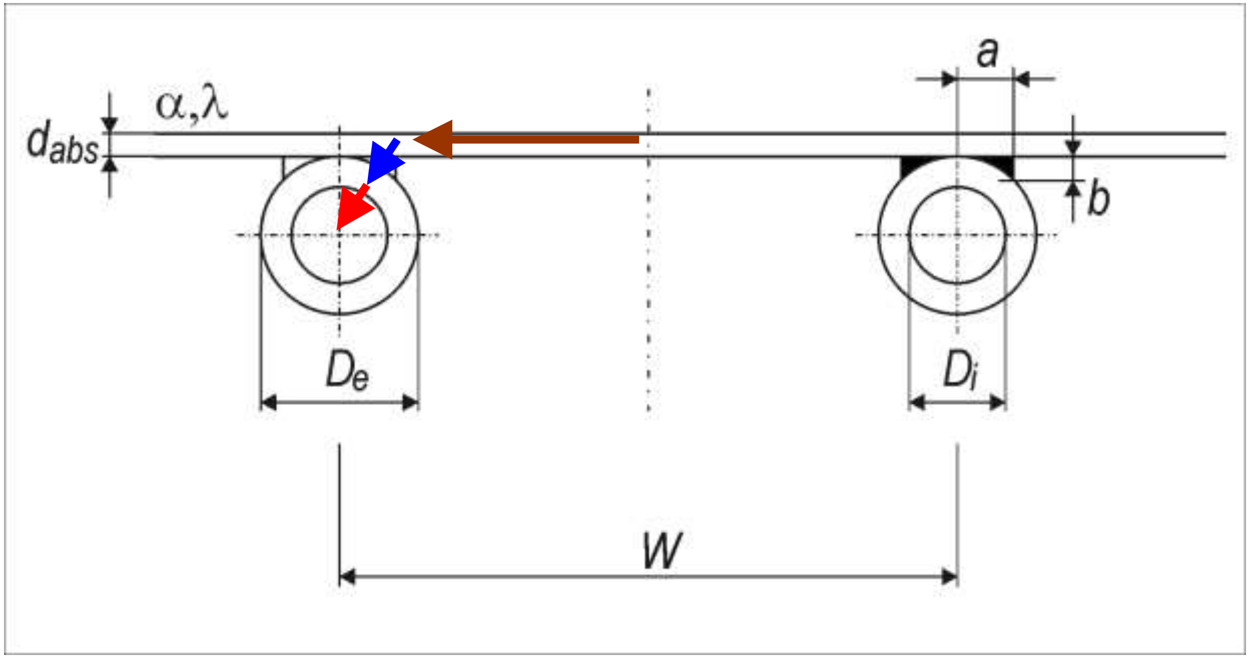
analogy to experimentally obtained efficiency curve reported by testing institutes

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

Efficiency factor *F'*

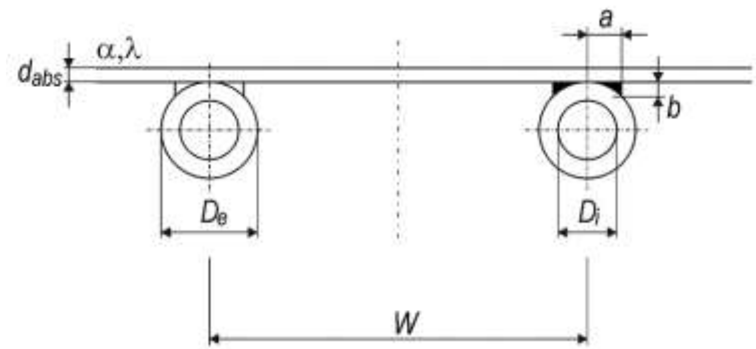
$$F' = \frac{1/U}{W\left[\frac{1}{U[2a+(W-2a)F]} + \frac{1}{C_b} + \frac{1}{h_{fi}\,\pi D_i}\right]}$$

function of fin efficiency *F*

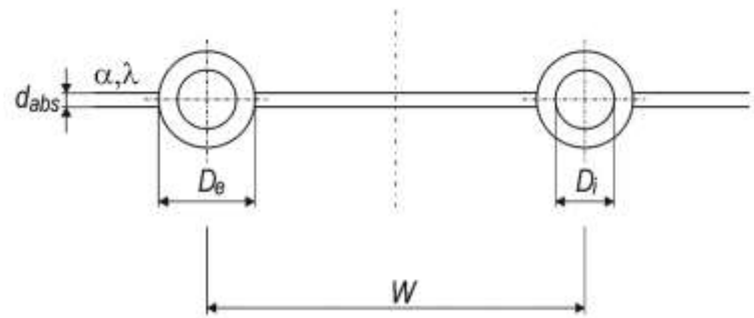


Efficiency factor – absorber designs

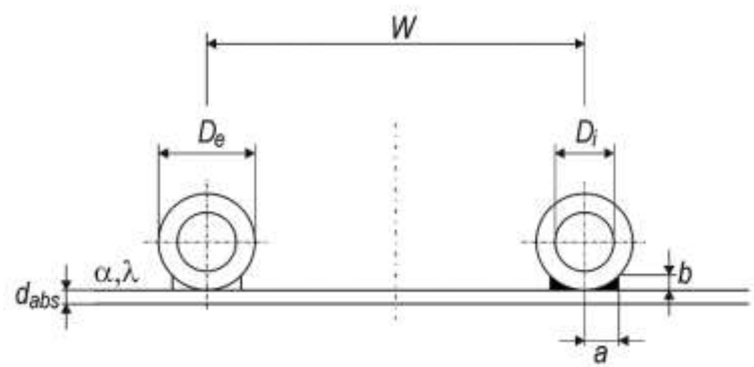
$$F' = \frac{1/U}{W\left[\dfrac{1}{U[2a+(W-2a)F]} + \dfrac{1}{C_b} + \dfrac{1}{h_{fi}\,\pi D_i}\right]}$$



$$F' = \frac{1/U}{W\left[\dfrac{1}{U[2a+(W-2a)F]} + \dfrac{1}{h_{fi}\,\pi D_i}\right]}$$

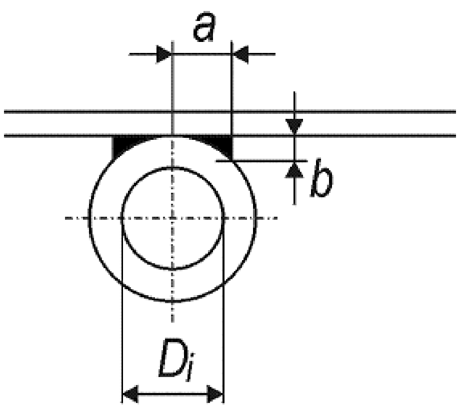


$$F' = \frac{1}{\dfrac{WU}{h_{fi}\,\pi D_i} + \dfrac{1}{\dfrac{2a}{W} + \dfrac{1}{\dfrac{WU}{C_b} + \dfrac{W}{(W-2a)F}}}}$$



Bond conductance

bond conductance – estimated from geometry and quality of contact



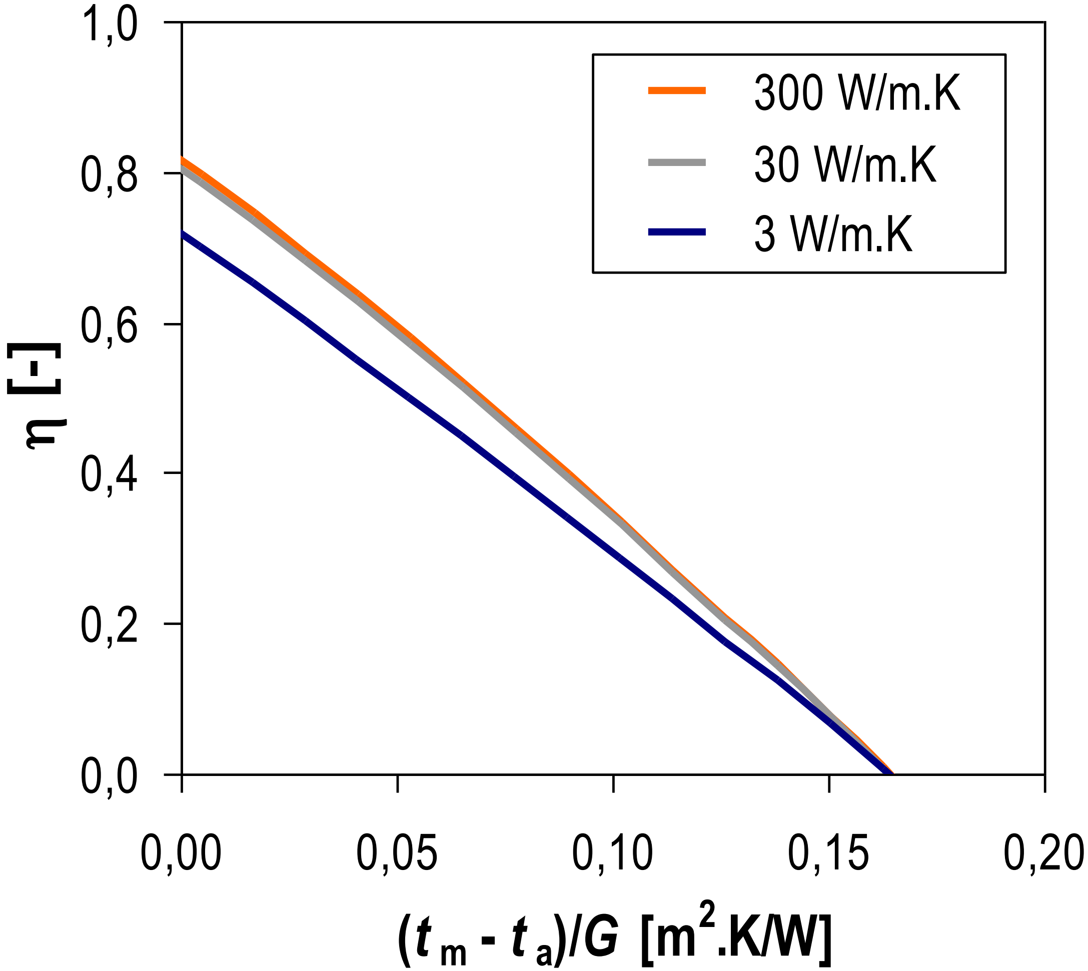
$$C_b = \frac{\lambda_b a}{b}$$

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

bond conductance > 30 W/mK needed

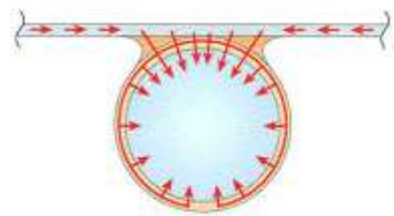
Bond conductance

for C_b < 30 W/mK significant reduction of collector performance

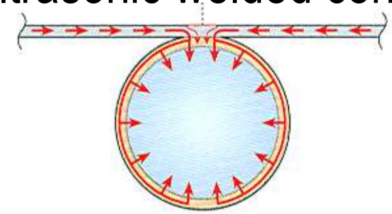


practically no difference between high and middle quality bonds

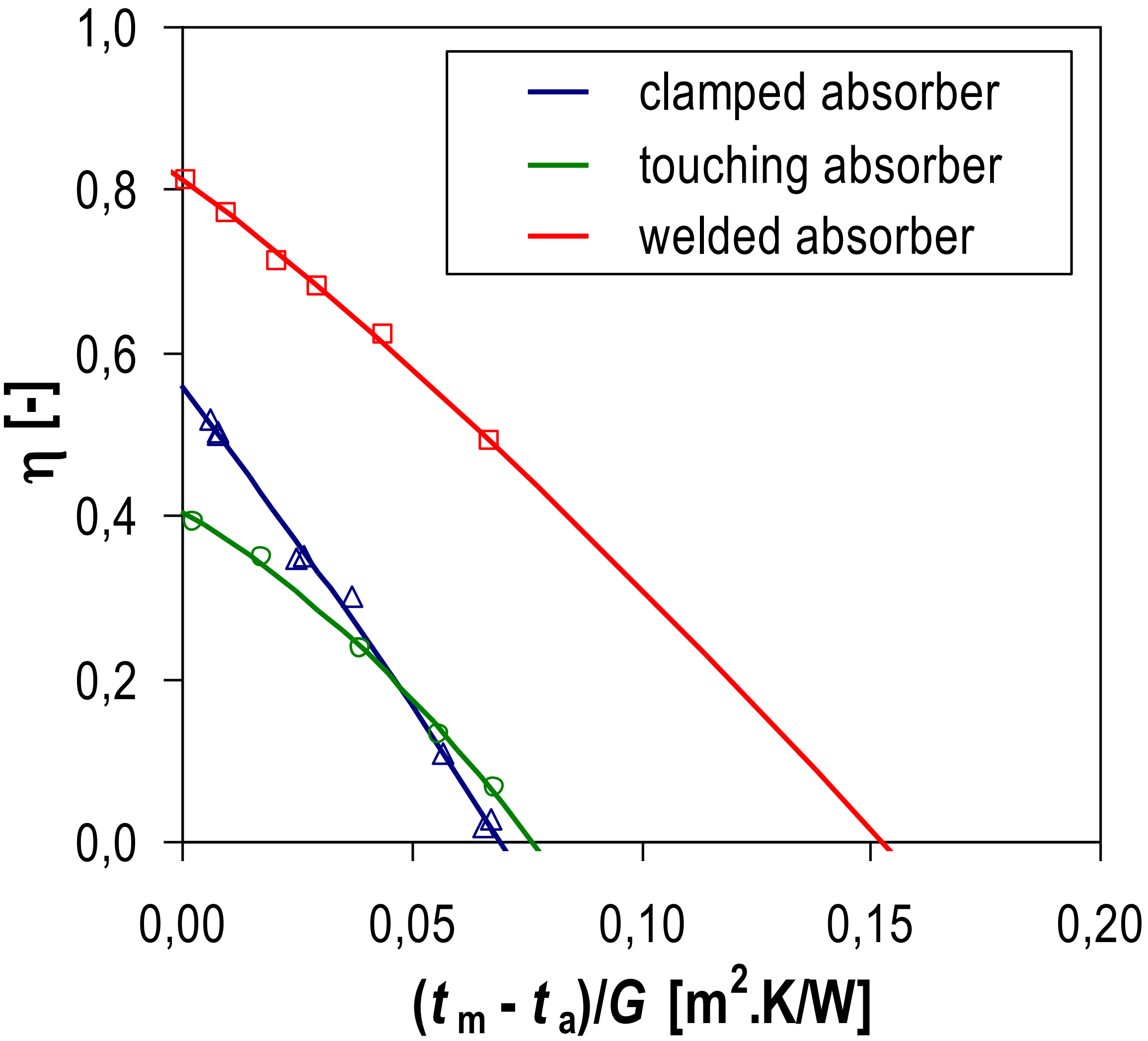
brazed contact



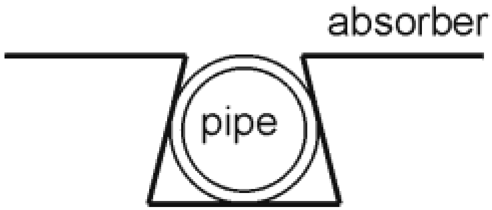
ultrasonic welded contact



Bond conductance: testing



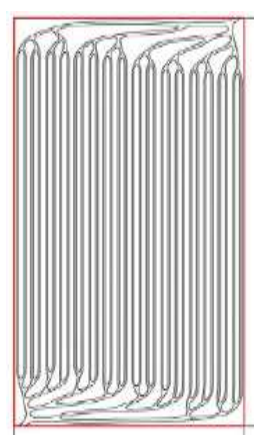
clamped bond

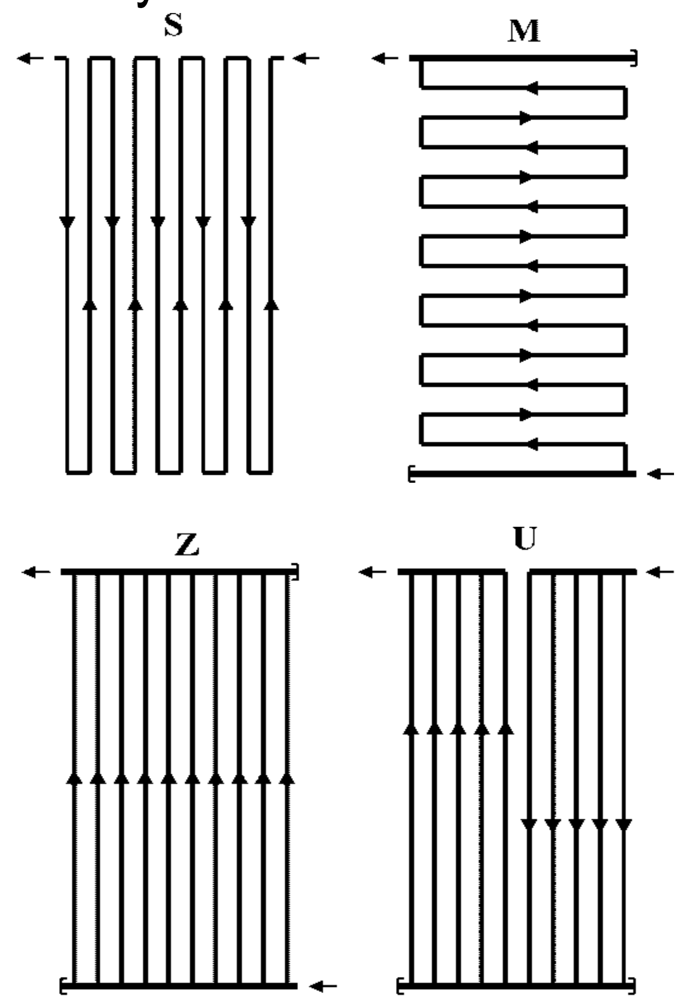


Convection heat transfer inside pipes

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

$$h_{\mathrm{fi}} = \mathrm{Nu}_{\mathrm{D}} \cdot \frac{\lambda_{\mathrm{f}}}{D_{\mathrm{i}}}$$

Nusselt number in circular pipes

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

$$\mathrm{Nu_D} = \frac{48}{11} = 4.364$$ (Shah)

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

$$\mathrm{Nu_D} = \begin{cases} 1.953 \cdot L^{*-1/3} & L^* \le 0.03 \\ 4.364 + \dfrac{0.0722}{L^*} & L^* > 0.03 \end{cases}$$ (Shah)

$$L^* = \mathrm{Gz}^{-1} = \frac{L / D_\mathrm{i}}{\mathrm{Re_D} \cdot \mathrm{Pr}}$$ Gz ... Graetz number

Nusselt number in circular pipes

- **turbulent flow**

$$\mathrm{Nu_D} = A\,\mathrm{Re_D^{\mathit{m}}}\,\mathrm{Pr}^{n}$$ (Dittus-Boelter, Colburn, Sieder-Tate)

$$\mathrm{Nu_D} = \frac{(f/8)(\mathrm{Re_D} - 1000)\,\mathrm{Pr}}{1 + 12.7\,(f/8)^{1/2}\left(\mathrm{Pr}^{2/3} - 1\right)}$$ (Gnielinski)

where $f = (0.79\ln \mathrm{Re_D} - 1.64)^{-2}$ for smooth pipes

for more models see Reference handbook to KOLEKTOR

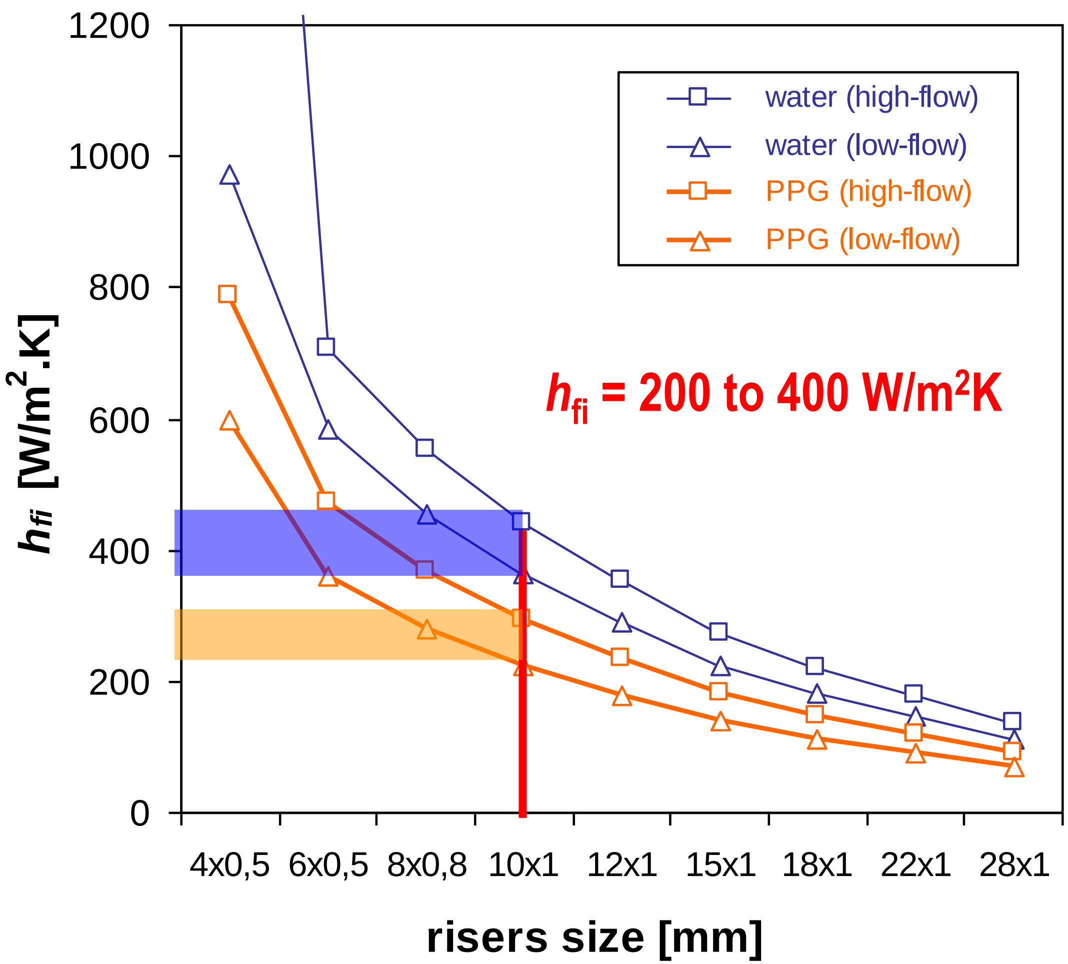
Convection heat transfer inside pipes

- **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



high flow: 0.025 kg/s.m^2

low flow: 0.005 kg/s.m^2

Influence of riser pipe diameter

$$F' = \frac{1/U}{W\left[\dfrac{1}{U[2a+(W-2a)F]} + \dfrac{1}{C_b} + \dfrac{1}{h_{fi}\,\pi D_i}\right]}$$

$$\frac{1}{h_{fi}\pi D_i} = \frac{D_i}{\mathrm{Nu}\lambda_f}\,\frac{1}{\pi D_i} = \frac{1}{\mathrm{Nu}\lambda_f \pi}$$

term is **dependent directly on Nu**

laminar flow

$$\mathrm{Nu_D} = f(\text{constant}) = 4.364$$

$$\mathrm{Nu_D} = f(L^*)$$

thermal diffusivity

$$L^* = \frac{L}{D_i}\,\frac{\nu}{u.D_i}\,\frac{\alpha}{\nu} = \frac{L.\alpha}{D_i^2}\,\frac{\pi D_i^2}{4\dot{V}} = \frac{\pi L\,\alpha}{4\dot{V}}$$

term $\dfrac{1}{h_i \pi D}$ 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 λ = 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

	$\frac{1}{U \cdot [D_e + (W - D_e) \cdot F]}$	$\frac{1}{C_{spoj}}$	$\frac{1}{h_i \cdot \pi \cdot D_i}$
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 m, fin width W = 110 mm, absorber thickness d = 0.2 mm, copper λ = 350 W/m.K, pipe D_e/D_i = 10/8 mm

bond conductance C_b = 200 W/K

U = 4 W/m²K

	$\frac{1}{h_i \cdot \pi \cdot D_i}$	F' U = 4 W/m²K
h_i = 400 W/m².K	0.099	0.919
h_i = 600 W/m².K	0.066	0.931

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

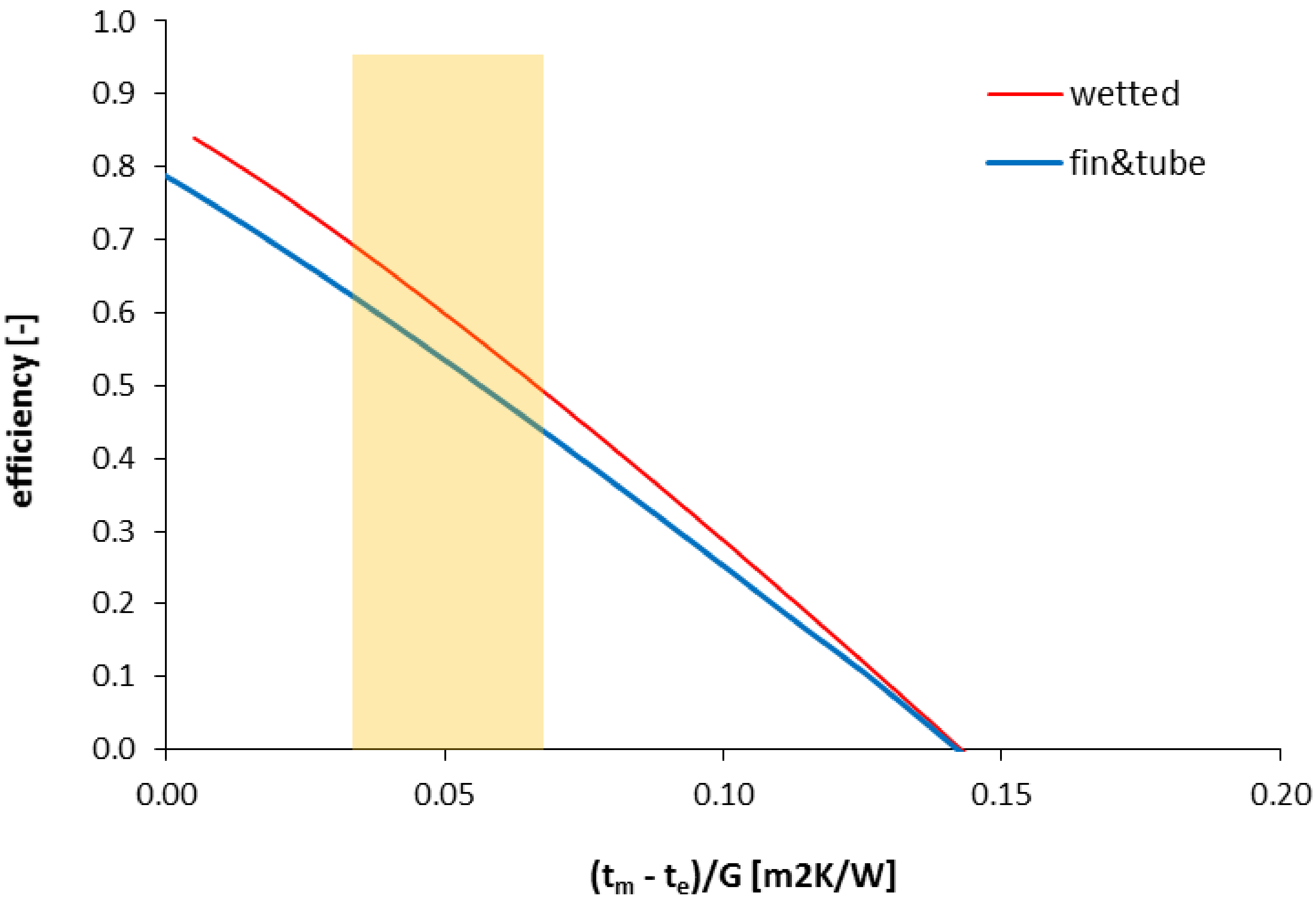
- **fully wetted absorber from steel**
  - type of bond middle, pipes D_e/D_i = 6/5 mm, bond a/b = 3/1 mm
  - material steel, thickness 0.5 mm
  - bond conductivity 100 W/mK
  - number of channels 145 pcs (not 150 pcs) ... **W = 6.01 mm > D_e**

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



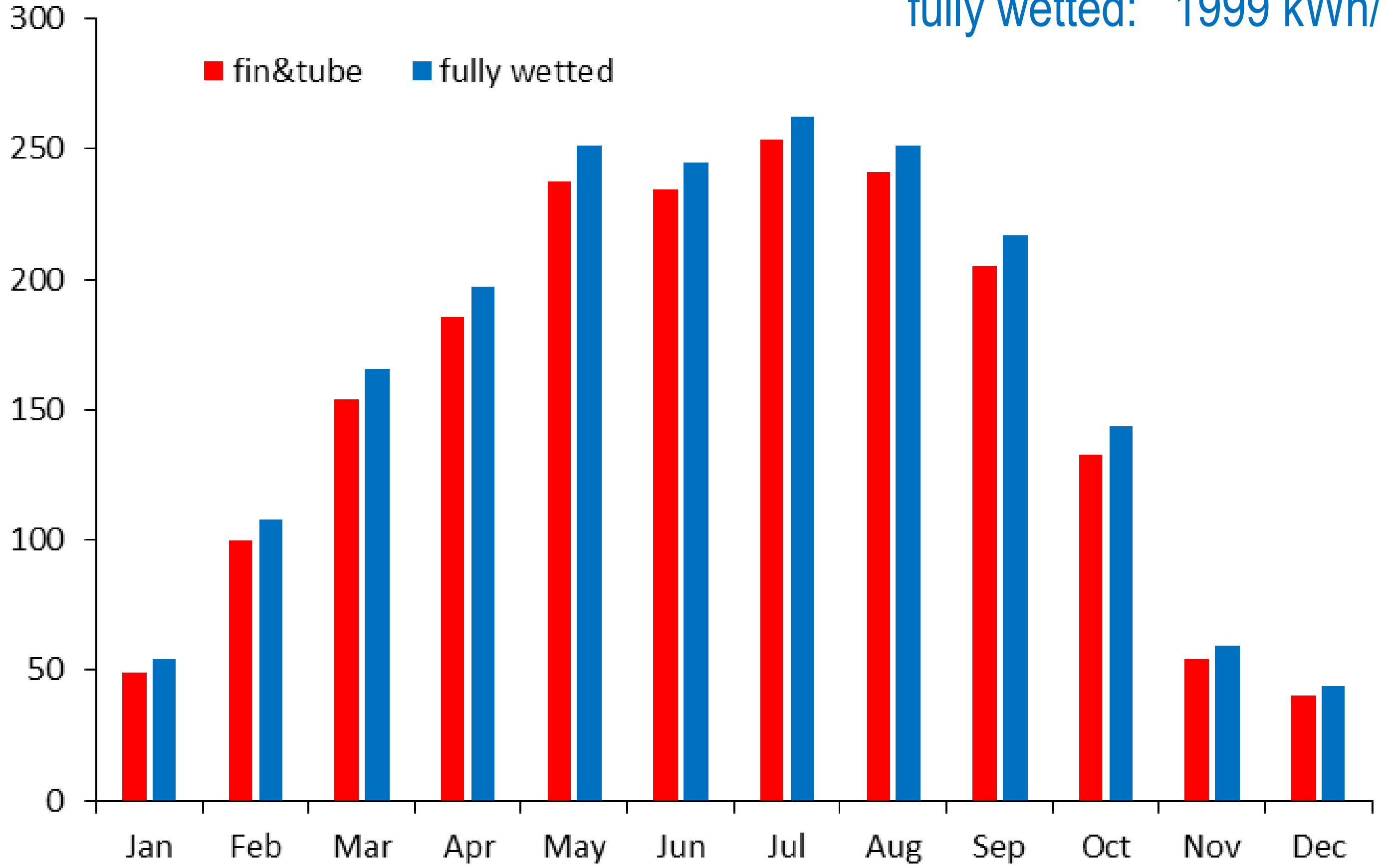
Fully wetted absorber vs. fin&tube

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

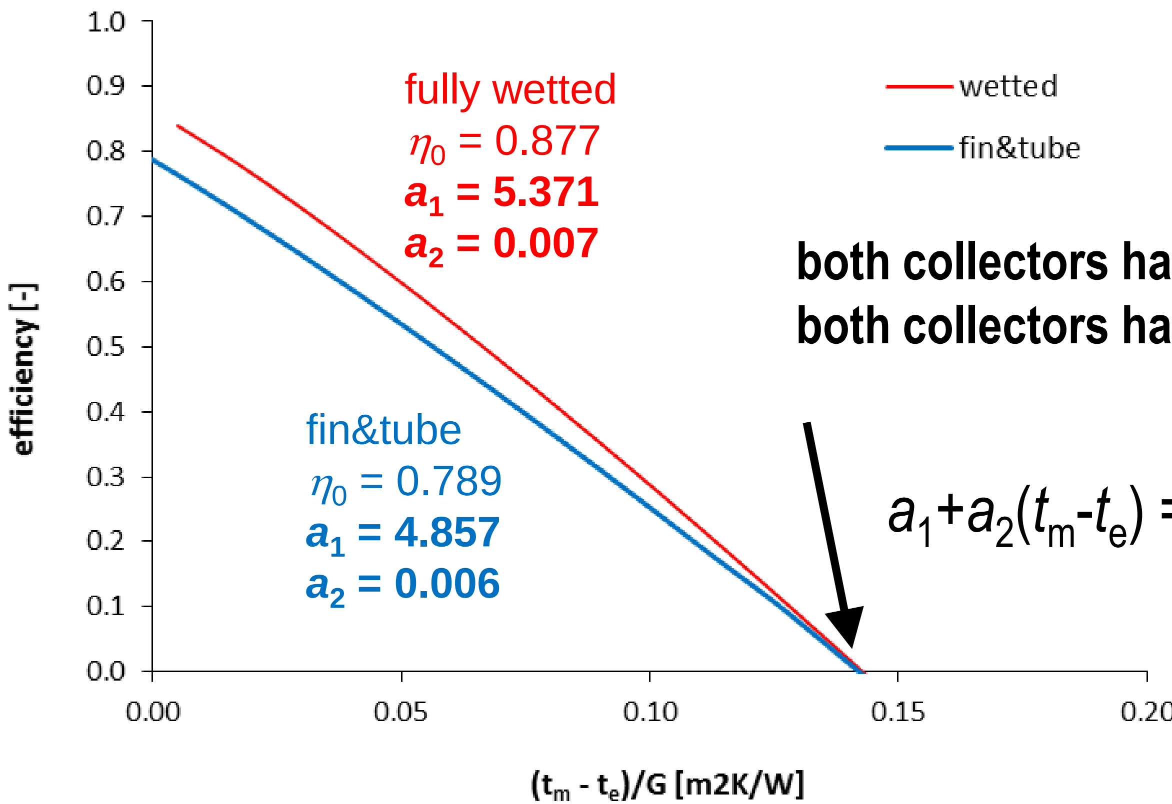


Fully wetted absorber vs. fin&tube

- 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!

$$a_1+a_2(t_m-t_e) = F'U$$

$F'=0.996$

$F'=0.901$

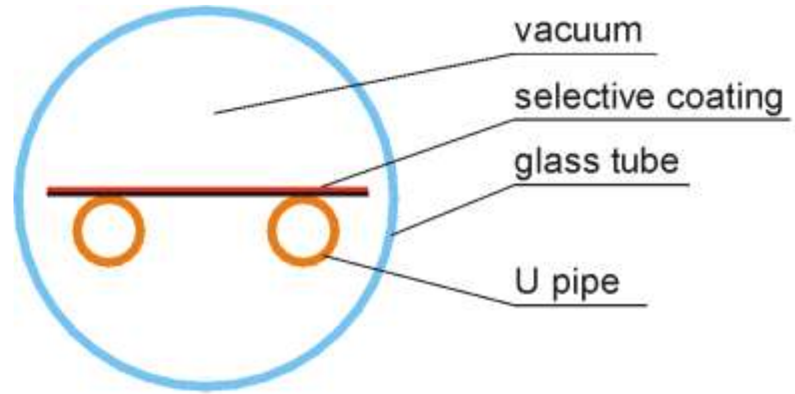
$U=5.39\ \mathrm{W/m^2K}$
for given ΔT

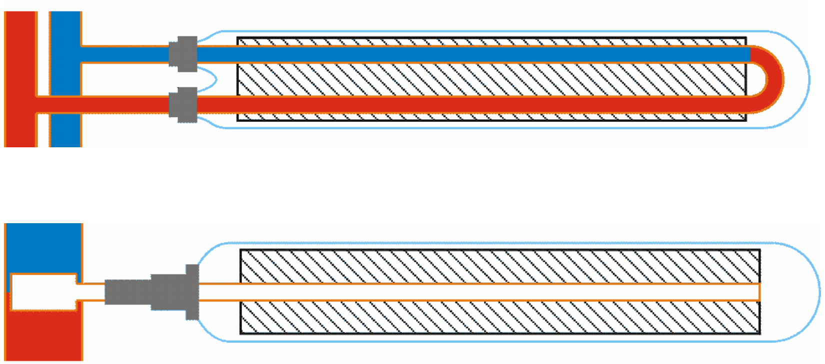
coefficients η_0, a_1, a_2 are one complex and cannot be separated

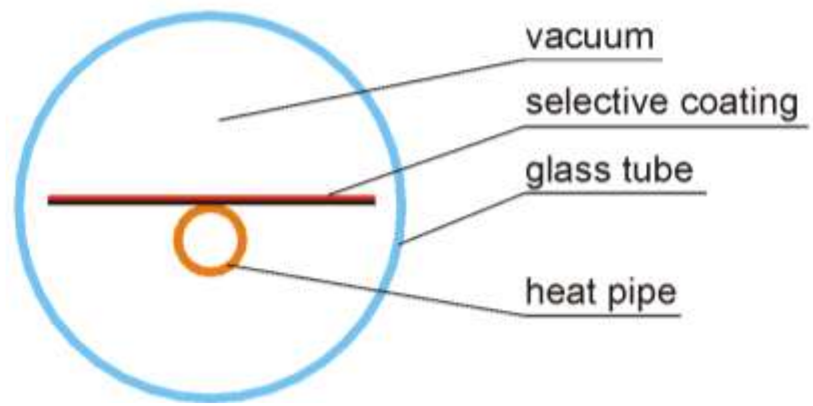
Vacuum tube collectors x efficiency factor

Single glass vacuum tube / flat absorber

- direct flow
- concentric direct flow
- heat pipe







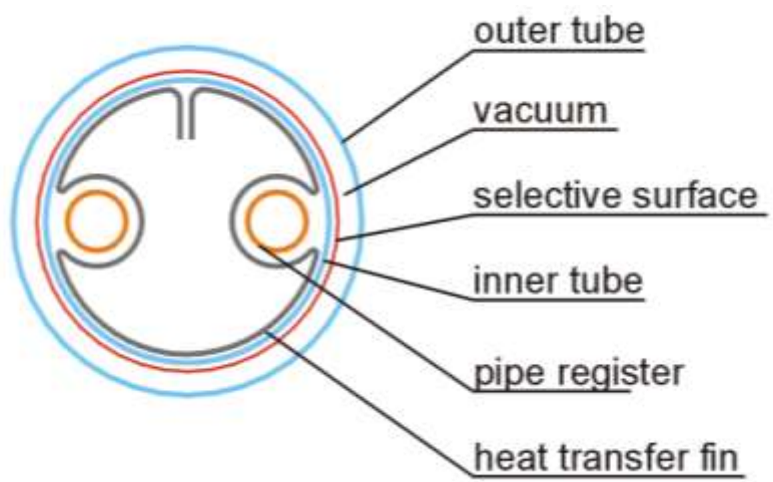
firm metal contact absorber-pipe provides high *F'*

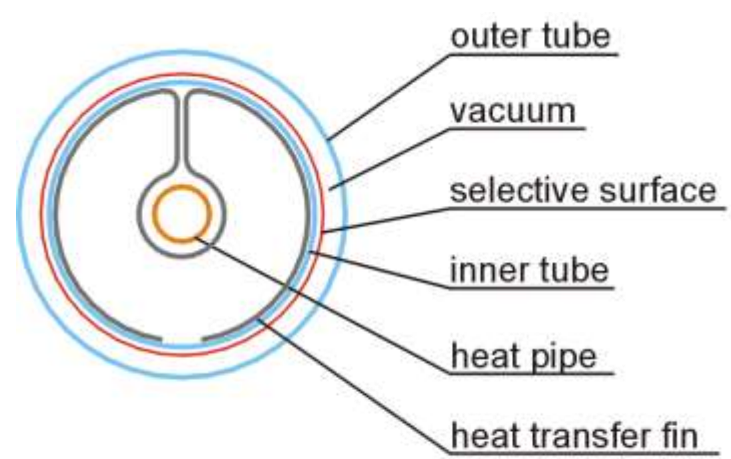
Vacuum tube collectors x efficiency factor

Double-glass (Sydney) vacuum tube

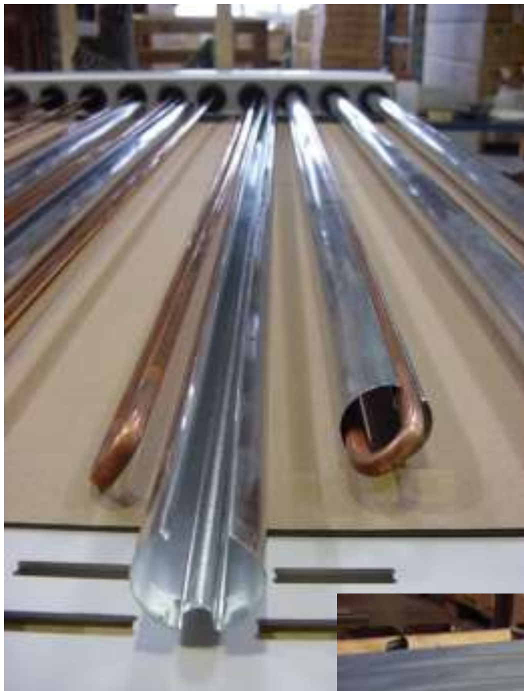
- all glass concentric tube (Dewar)
  - cylindric absorber (internal) glass tube
  - cover (external) glass tube
  - vacuum in space between
- **absorber coating** applied on exterior surface of internal glass tube
- conductive heat transfer fin in contact with interior surface

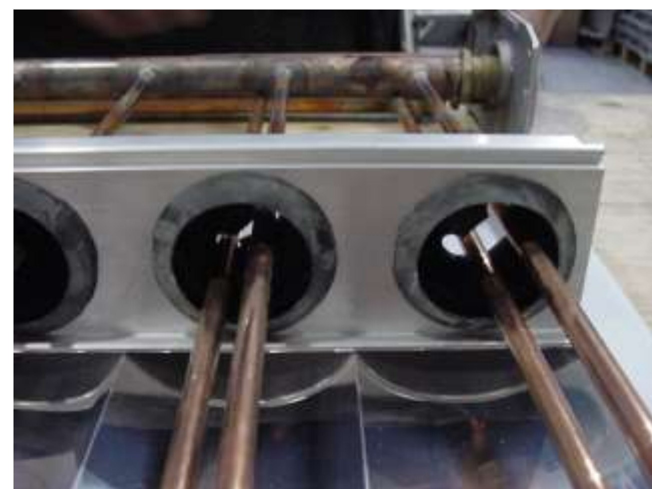
problematic contact absorber-pipe ...





Sydney vacuum tube collectors



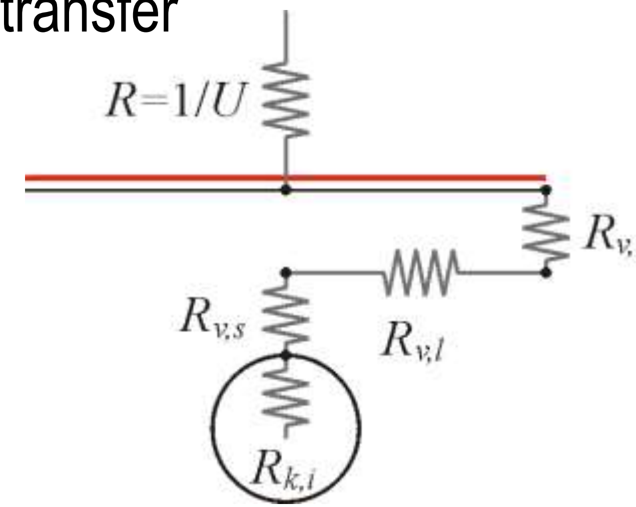


Thermal analysis of direct flow VTC

given by Sydney tube

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

tube collector scheme of heat transfer

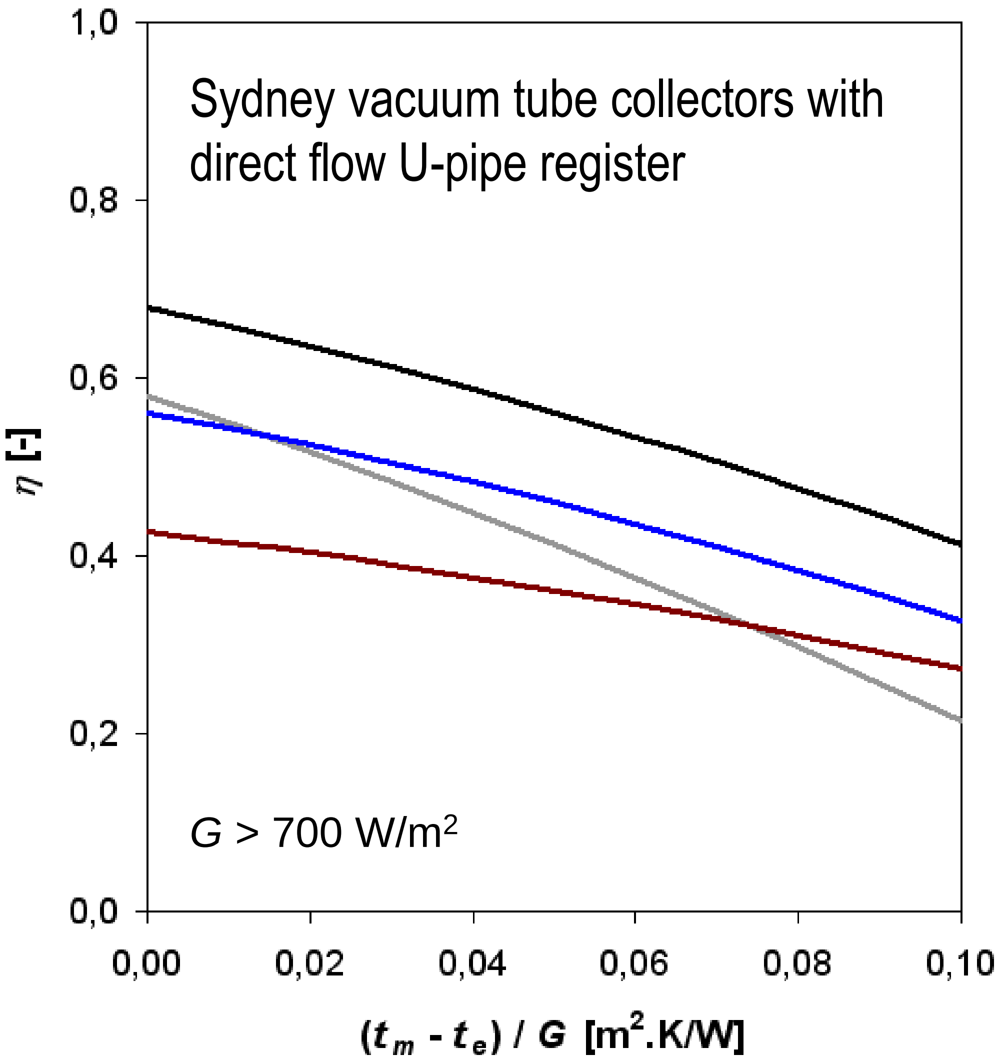


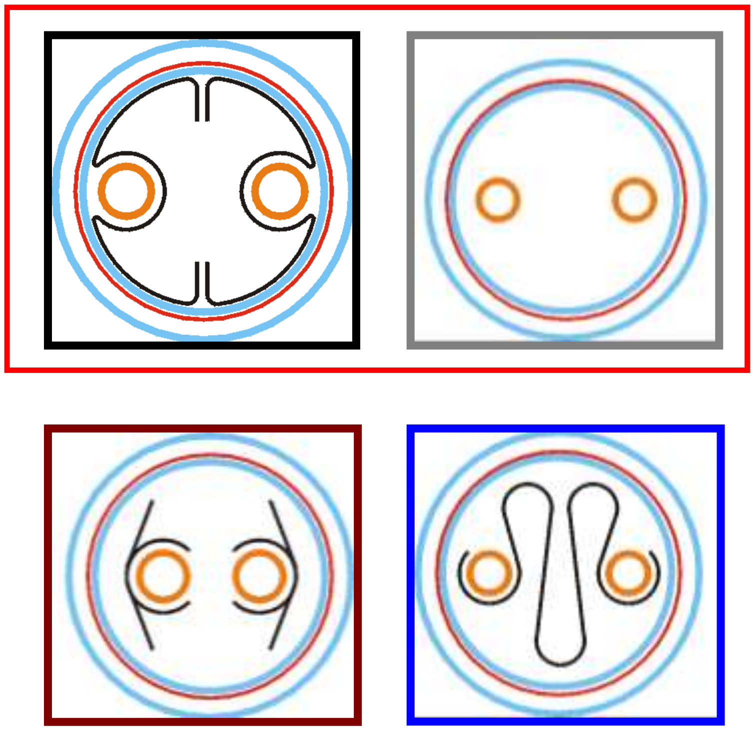
<u>contact fin:</u> short (W), conductive (λ), thick (d)

<u>bond-contact:</u> conductive; absorber tube-fin, fin-pipes

<u>heat removal:</u> laminar / turbulent flow in pipes

Influence of contact fin



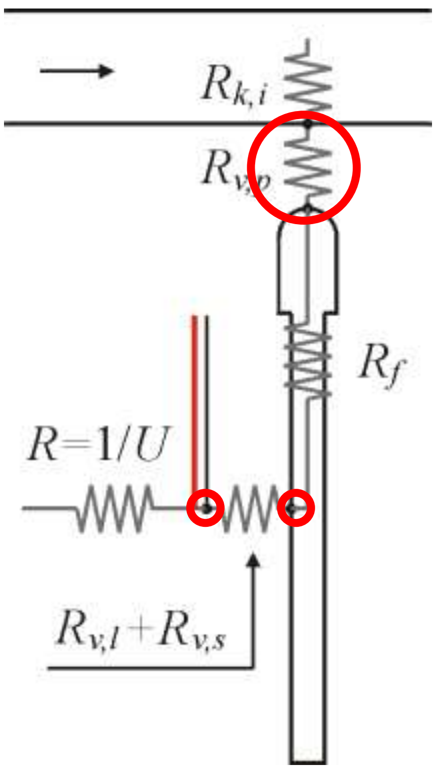


contact fin is a principle element in Sydney collector

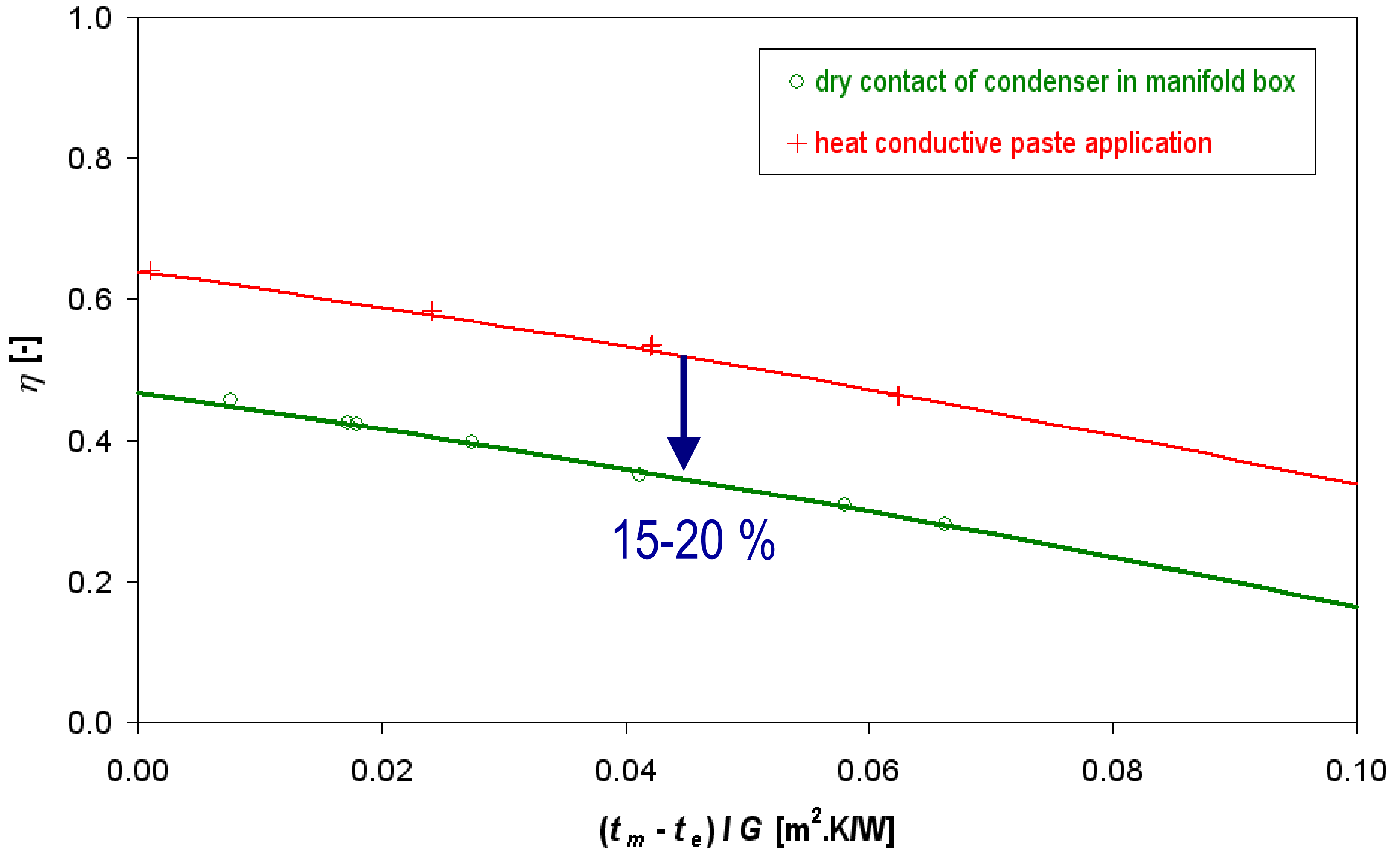
Contact resistance in heat pipe VTC

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

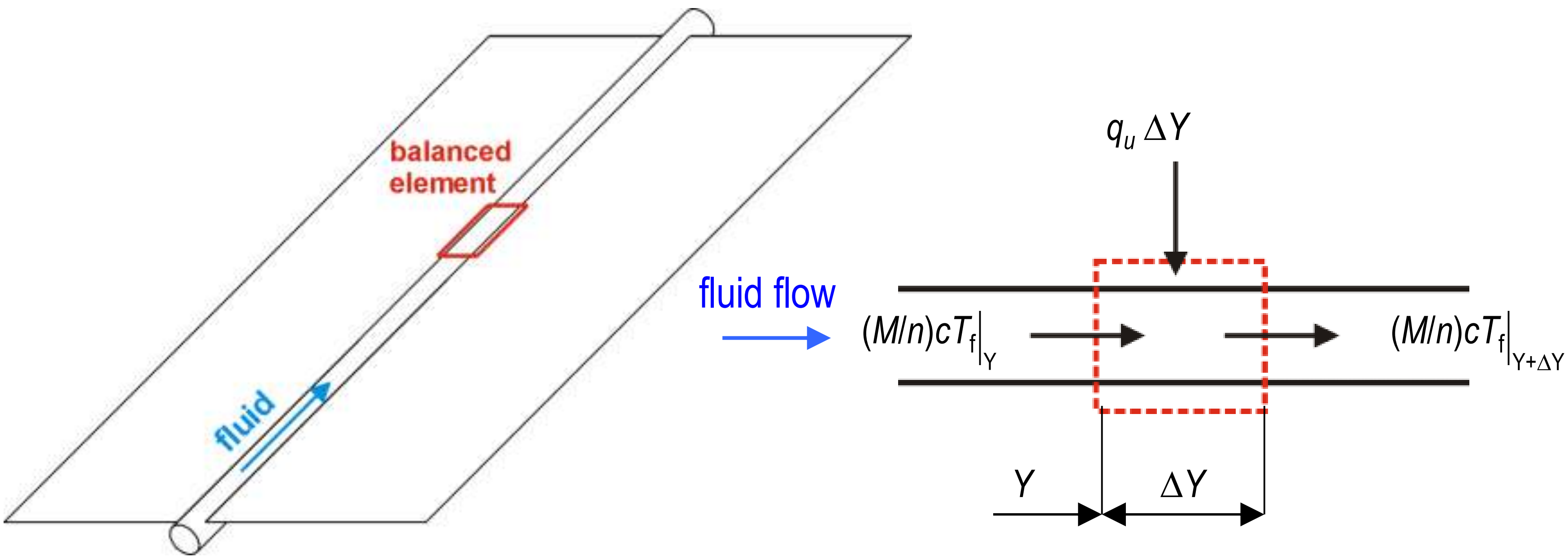
- evaporator (in contact with fin transfering the heat from inner glass tube)
- condenser (placed into manifold box)



Contact resistance in heat pipe VTC



Temperature distribution in flow direction



derivation of temperature distribution between tubes (in absorber fin)

$$\left(\frac{\dot{M}}{n}\right)cT_f\Big|_y - \left(\frac{\dot{M}}{n}\right)cT_f\Big|_{y+\Delta y} + \Delta y q_u^{/} = 0$$

yields in solution of differential equation of 1rst order

Heat removal factor F_R

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

<u>definition:</u> relates the actual collector gain to gain if absorber surface at fluid inlet temperature

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

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

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?

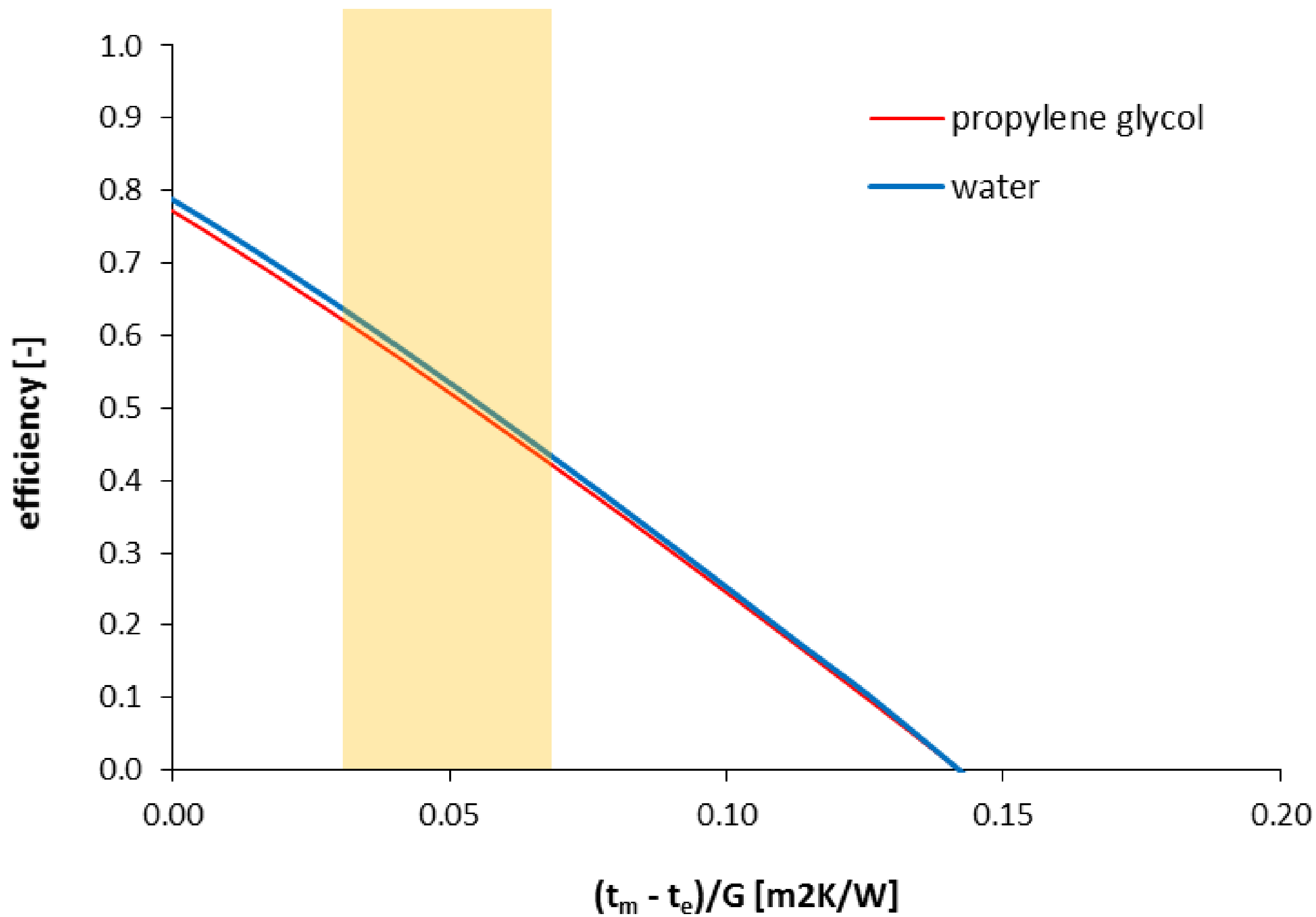
- **propylene glycol / water**
  - mixture 50 % / 50 % for freezing point -32 °C
  - 8x viscous at 20 °C, high influence of temperature, only 2x at 80 °C
  - lower specific heat by cca 25 %
  - worse heat transfer

Water vs. propylene glycol

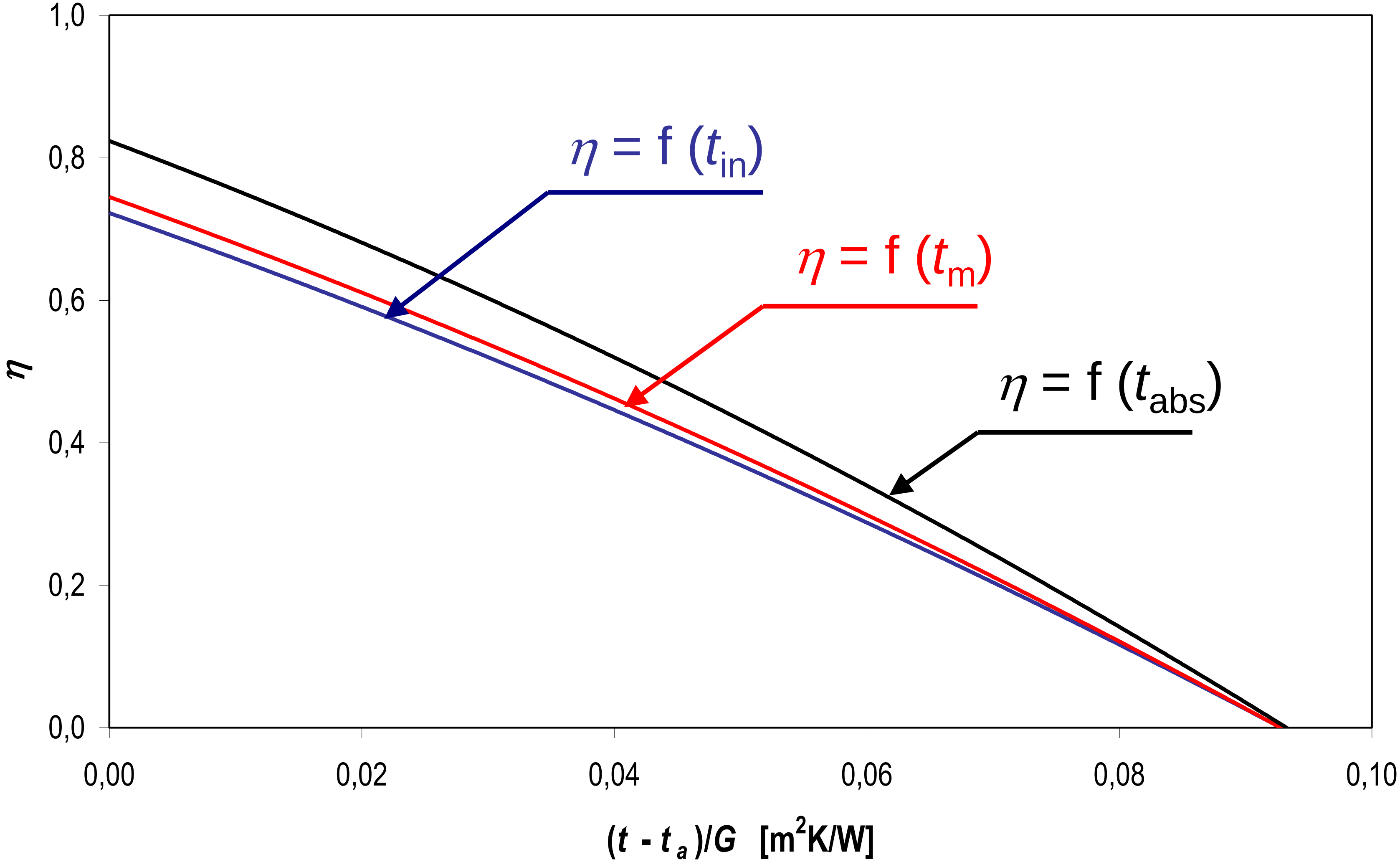
- run KOLEKTOR programme, open **default.kol**
- Absorber card — make the change of fluid
- Calculation card — select Efficiency curve calculation

  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



Efficiency curve



Useful heat output of solar collector

based on absorber temperature, external balance of absorber

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

based on mean fluid temperature, experimental work

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

based on fluid inlet temperature, simulation tools

$$\dot{Q}_u = F_{\mathrm{R}} A_c\left[\tau\alpha G - U\left(t_{in} - t_a\right)\right]$$

Solar collector temperatures

mean absorber temperature

$$t_{\mathrm{abs}} = t_{\mathrm{in}} + \frac{\dot{Q}_{\mathrm{u}}/A_{\mathrm{c}}}{F_{\mathrm{R}}U}(1 - F_{\mathrm{R}})$$

mean fluid temperature

$$t_{\mathrm{m}} = t_{\mathrm{in}} + \frac{\dot{Q}_{\mathrm{u}}/A_{\mathrm{c}}}{F_{\mathrm{R}}U}\left(1 - \frac{F_{\mathrm{R}}}{F'}\right)$$

based on

fluid inlet temperature

fluid output temperature

$$t_{\mathrm{out}} = 2t_{\mathrm{m}} - t_{\mathrm{in}}$$

Solar collector efficiency

based on absorber temperature

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

based on mean fluid temperature

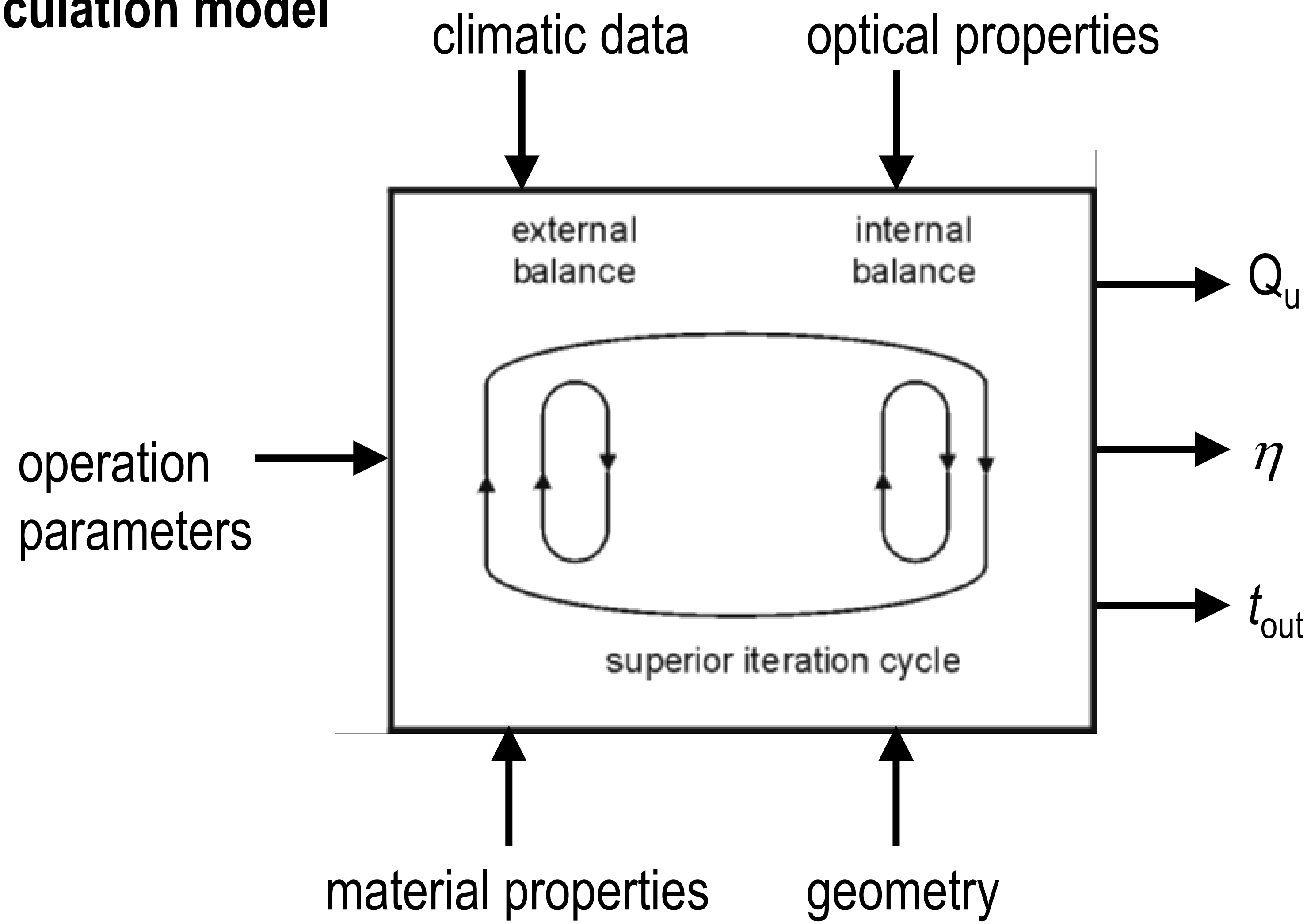
$$\eta = F'\left[\tau\alpha - U\frac{(t_m - t_a)}{G}\right]$$

based on fluid inlet temperature

$$\eta = F_\mathrm{R}\left[\tau\alpha - U\frac{(t_{in} - t_a)}{G}\right]$$

Solar collector performance calculation

calculation model



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