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Theoretical and experimental investigations regarding open volumetric receivers of CRS

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Abstract

Concentrated sunlight is absorbed in solar thermal power plants by heat resistant absorbers and converted into usable heat which is transferred to a carrier medium. In solar tower power plants such as the plant in Jülich porous absorbers can reach temperatures up to 1000 °C and higher. At this power plant air as heat transfer medium is sucked in through the absorber and heated up to about 700 °C. The absorber is composed of highly porous ceramic or metal wire structures. The SIJ investigates the optimization of solar absorption and the convective heat transfer to the air using thermo and fluid mechanical calculations. In such simulations the key quantities are the penetration depth of solar radiation κ and the volumetric heat transfer coefficient α_v, which indicates how much energy - depending on the volume and temperature difference - is transferred by convection between solid and fluid. The attenuation of the radiation into the depth of the absorber is described generally by an exponential function with parameter κ. This is accompanied by heat transfer to the structure. Existing models of the key quantities have been validated by experimental data.

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Keywords: CRS; CSP; open volumetric receiver; heat transfer coefficient; solar radiation; extinction coefficient

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1. Introduction

Open volumetric receivers are structures absorbing solar radiation, to be used in solar thermal power plants. They consist of porous materials, wire meshes or wire grids (see Fig.1) etc. made of solids resisting high temperatures. The generated heat is transferred to a fluid passing through the more or less open or porous structural material.

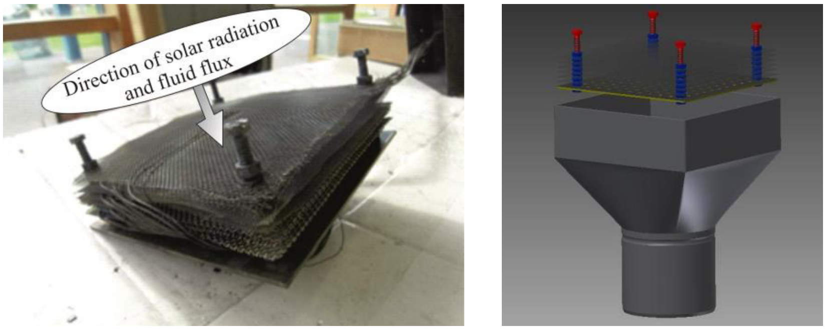

Fig. 1. Receiver structure with wire grid, several thermocouples are integrated inside for temperature measurements (left). The wire structure is installed in a receiver cup (right)

The following models are presented for the analysis of volumetric receivers with emphasis on wire structures. The basic concepts can be found in [1], [2], [3], [4], [5]. In [1] and [2] general radiation modeling aspects in porous structures are considered, in [3], [4] and [5] receiver related explanations are given.

Nomenclature

κ	extinction coefficient	1/m
α_v	volumetric heat transfer coefficient	$W/m^3/K$
T_S	solid temperature	K
T_F	fluid temperature	K
K_n	radiation in anti-fluid direction in front of the nth layer	W/m^2
K_o	backscattered radiation of first layer/ absorber	W/m^2
$I_{(x)}$	radiation at position x	W/m^2
I_0	radiation on first layer/ absorber	W/m^2
I_n	radiation in fluid direction in front of the nth layer	W/m^2
d	diameter	m
Po	porosity	-
σ	Stefan- Boltzmann constant	$W/(m^2\ K)$
$\dot{M}$	fluid mass flux	$kg/m^2/s$
cp	heat capacity of the fluid	J/kg/K
ξ	dimensionless coordinate	-

2. Radiation layer model

In order to calculate the thermal performance, a quasi-homogeneous layer model was developed describing radiation and convection heat transfer (Fig. 2). The layer-related parameters, such as the extinction coefficient κ and the volumetric heat transfer coefficient α_v can be chosen independently of the chosen layer thickness δ, which has practically no influence on the numerical simulations. This thickness parameter determines more or less only the numerical resolution.

A layer of thickness δ is a cell filled with a solid and a fluid both exchanging heat. The sources of the generated heat are radiation flows absorbed and reflected by the solid structure.

There are three types of equations characterizing the layer system. They are given in sub chapters 2.2 – 2.4 for time independent stationary conditions. In sub chapter 2.1 the construction of the layer model is discussed.

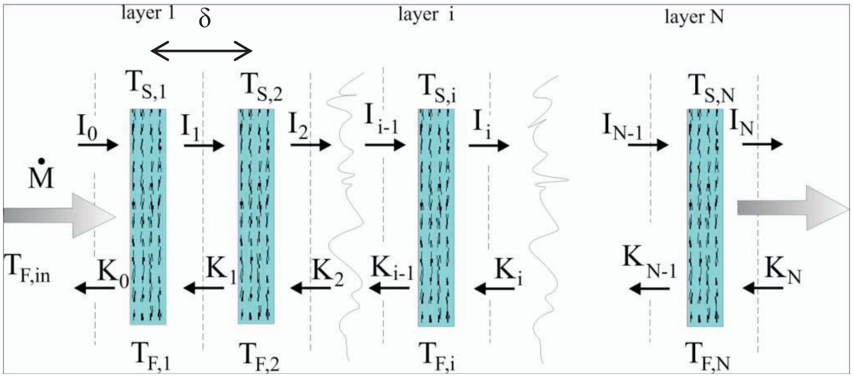

Fig. 2. Layer model of an open volumetric receiver. Radiation fluxes I and K are shown. Radiation, convection and conduction influence the fluid and solid temperatures T_F and T_S

Radiation flux quantities I and K represent both solar and thermal radiation, whereby at any layer i I_i represents the total flux density of radiation directed into the structure (from left to right in Fig. 2) at the right side of layer i, and K_i represents the total flux density of radiation directed towards the surface (from right to left in Fig. 2) at the right side of layer i.

2.1. Geometrical transformation

The real volumetric wire mesh absorber is characterized by properties such as absorber wire diameter and porosity making up a diffuse wire structure as shown in Fig. 3.

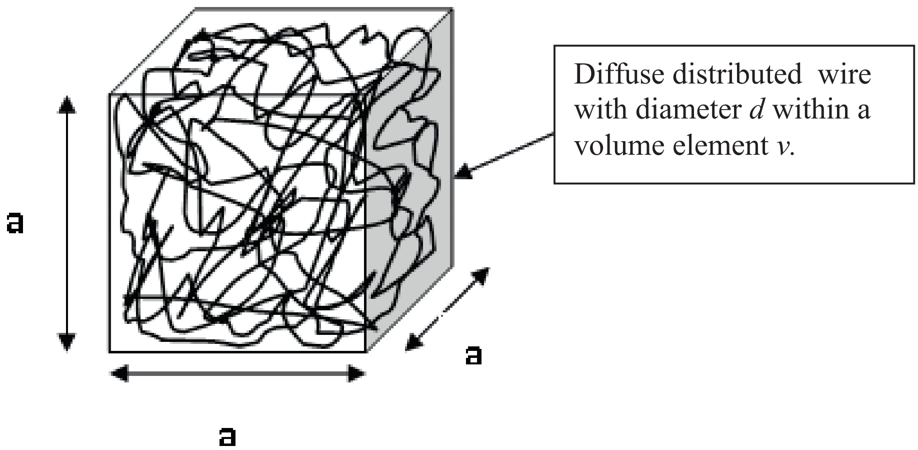

Fig. 3. Diffuse wire arrangement

In order to calculate the physical heat transport and radiation effects within such a random structure, it was replaced by a well-defined three-dimensional equally distributed wire arrangement within the absorber structure maintaining the original wire volume (see Fig. 4) [6].

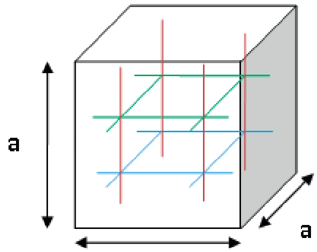

Fig. 4. Uniform arrangement of wire pieces to simplify calculations

The attenuation of the radiation depending on the absorber depth is described by an extinction coefficient κ (see eq. 1). The extinction coefficient κ depends on the geometry of the wire structure and can be expressed as a function of porosity *Po* and wire diameter *d (*eq. 1) in the case of a regular wire arrangement. Calculation details are found in [4]. It is difficult to quantify the differences between uniform and the diffuse wire arrangement. First measurements show differences in the range of 20%.

$$\kappa = -\frac{2}{d} \cdot \sqrt{\frac{4 \cdot (1-P_0)}{3 \cdot \pi}} \cdot ln(1 - \sqrt{\frac{4 \cdot (1-P_0)}{3 \cdot \pi}}) \qquad (1)$$

To calculate the radiant flux density *I(x)* at a certain position *x*, the extinction coefficient κ is used (see eq.2).

$$I(x)=I_0 \times e^{-\kappa \times x} \qquad (2)$$

2.2. Radiation flow equations

Furthermore, the radiation flux density is related to the wire temperature at layer i, $T_{S,i}$, by the following equations (3), where the first equation represents the flux situation at the right-hand side of layer i, and the second equation represents the flux situation at the left-hand side of layer i.

$$I_i\text{-}B \times I_{i\text{-}1}\text{-}R \times K_i=(1\text{-}B\text{-}R) \times \sigma \times T_{si}^{\ 4} \qquad (3)$$

$$K_{i\text{-}1}\text{-}B \times K_i\text{-}R \times I_{i\text{-}1}=(1\text{-}B\text{-}R) \times \sigma \times T_{si}^{\ 4}$$

The parameter *B* characterizes the transmission property of the layer (*B* is computed based on eq. 2 as $B = e^{-\kappa \times \delta}$), *R* is the reflectivity, $T_{S,i}$ is the solid temperature at layer i, and σ = 5.64e-08 W/m²/K^4. Spectral and directional radiation properties are not considered. Concentrated solar light and heat radiation are simply given in an integrated way as an "energy flux density (W/m²)". In case that *B* and *R* are not very different for light and heat radiation the assumption is reasonable. The fluid contribution to radiation processes is neglected.

2.3. Equations describing the heat exchange between solid and fluid

$$\dot{M} \times c_p \times (T_{F,\, i\text{-}1}\text{-}T_{F,i}) + \alpha_v \times \delta \times (T_{s,i}\text{-}T_{F,i})=0 \qquad (4)$$

$\dot{M}$ is the fluid mass flux density (kg/m²/s), c_p the specific heat of the fluid, T_F and T_S are the fluid and solid temperatures, respectively. α_v denotes the volumetric heat transfer coefficient (W/m³/K), which is given by the product of internal surface per volume and a surface related heat transfer coefficient.
The Nusselt number is calculated for wire pieces along the air flow [7] and for cross flow [8].

Firstly the individual heat transfer coefficients for cross (α_{cross}) and parallel (α_{par}) flow are calculated. The mean heat transfer coefficient can be calculated as the weighted average of α_{cross}. and α_{par} with weighting factors α_{abs}, w_{abs}, and x_{abs}, representing the fractions of wire in their respective orientation.

$$\alpha_{mean}=\frac{(\alpha_{abs}\times\alpha_{cross}+w_{abs}\times\alpha_{cross}+x_{abs}\times\alpha_{par})}{\alpha_{abs}+w_{abs}+x_{abs}} \tag{5}$$

To obtain the volumetric heat transfer coefficient α_v, α_{mean} (W/m²/K) has to be multiplied by the surface area density A_v (m²/m³).

2.4. Equations giving the radiation – solid interaction

$$\left(I_{i-1}-K_{i-1}\right)-\left(I_i-K_i\right)-\alpha_v\times\delta\times\left(T_{S,i}-T_{F,i}\right)+\frac{\lambda_{eff}}{\delta}\times\left(T_{S,\,i+1}+T_{S,i-1}-2\times T_{S,i}\right)=0 \tag{6}$$

The last term in equation (6) considers conduction between the solid parts of layers via an effective heat conductivity λ_{eff}.

The unknown quantities in equations 3-6 are the radiation flux densities and the temperatures of fluid and solid.
The solution of the nonlinear equation system can only be obtained by numerical methods.

3. Simplified model

In case that the entering solar induced flow is much greater than the thermal radiation at the solid surfaces based on their temperatures, it can be shown that the radiation flux follows an exponential function [2]. The parameter in the exponential function is the extinction coefficient κ (see eq. 1 and 2). From that it follows that the solar radiation can be considered as a heat source in the solid structure. The heat source density *r(x)* can be correlated with the incident solar radiation flux I_{net}. All the incoming and not reflected radiation energy is found in the heat source [9].

$$I_{net}=\int_0^\infty r_0\times exp(-\kappa\times x)\times dx \tag{7}$$

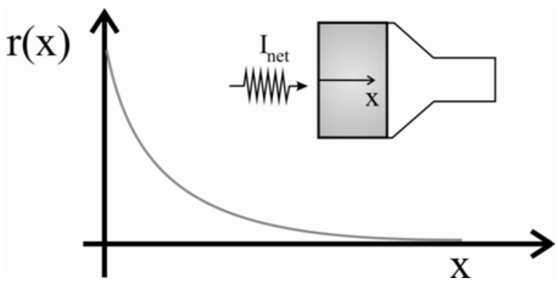

Fig. 5. Heat source density $r(x) = r_0\times exp(-\kappa\times x)$, the parameter r_0 is obtained via eq. (7): $r_0 = \kappa\times I_{net}$

This leads to a simplified model which can be treated analytically giving a relevant insight into the behavior of an open volumetric receiver depending on structural parameters and operational conditions. The used variable $\xi=\kappa\times x$ is a dimensionless length quantity into the receiver.

$$\kappa \times \dot{M} \times c_p \times \frac{dT_F}{d\xi} = \alpha_v \times (T_S - T_F) \tag{8}$$

$$\kappa^2 \times \lambda_{eff} \times \frac{d^2 T_s}{d\xi^2} = -r_0 \times exp(-\xi) + \alpha_v \times (T_S - T_F)$$

In case of negligible conductivity ($\lambda_{eff} \approx 0$) the differential equation system (8) can be solved directly from:

$$\kappa \times \dot{M} \times c_p \times \frac{dT_F}{d\xi} = r_0 \times exp(-\xi) \tag{9}$$

The analysis of real wire-based receiver structures shows that the omission of conduction is a reasonable assumption (see e. g. [3]). The following fluid and solid temperature distributions can be derived.

$$T_F(\xi) = T_{F,\,in} + \frac{r_0}{\kappa \times \dot{M} \times c_p} \times (1 - exp(-\xi)) \tag{10}$$

$$T_S(\xi) = T_{F,\,in} + \frac{r_0}{\kappa \times \dot{M} \times c_p} + \frac{r_0}{\alpha_v} \times (1 - \frac{\alpha_v}{\kappa \times \dot{M} \times c_p}) \times exp(-\xi) \tag{11}$$

A principal graphical representation of equations 10 and 11 is shown in Fig. 6. The interesting observation is here that dependent on the magnitude of fluid flow, two different temperature curves labeled '*a*' and '*b*' occur. In case the heat transfer conditions inside the receiver and the penetration depth of radiation is high (it means a low extinction coefficient) the outer receiver surface shows a lower temperature than the receiver parts inside (curve b).

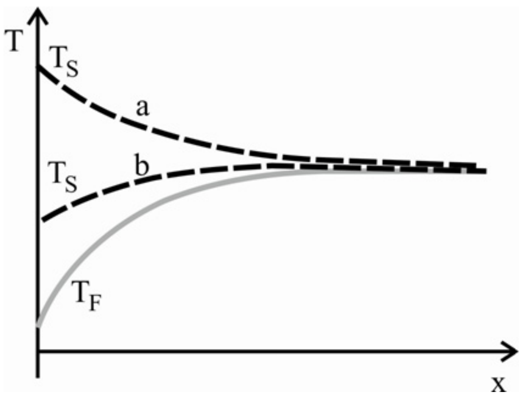

Fig. 6. Fluid temperature T_F and solid temperature T_S inside the receiver (x= ξ/κ); curve a: $\frac{\alpha_v}{\kappa} < \dot{M} \times c_p$;

curve b: $\frac{\alpha_v}{\kappa} > \dot{M} \times c_p$; the curves are constructed with constant $\frac{\alpha_v}{\kappa}$ and constant ratio $\frac{r_0}{\kappa \times \dot{M} \times c_p}$

4. Experimental setups and results

Experimental results of a wire mesh structure test for different mass fluxes are shown in Fig. 7, where the fluid mass flux was varied at constant incident radiation. It is seen that according to Fig. 6 for small mass flow rates the maximum temperature occurs clearly inside the receiver at some distance from the front surface: The 'volumetric effect'.

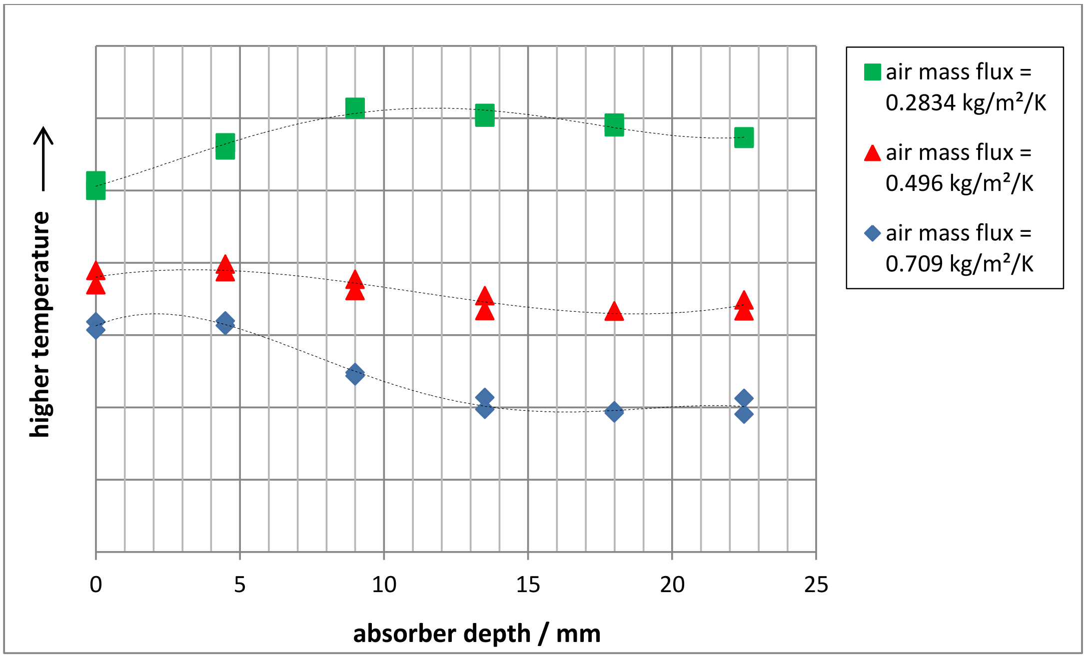

Fig. 7. Measured temperature profile inside a metallic wire absorber structure; it should be remarked that here r_0 was hold constant, the ratio $\frac{r_0}{\kappa \times \dot{M} \times c_p}$ was therefore different and so the air temperature is different for the different curves

The determination of the extinction coefficient κ and volumetric heat transfer coefficient α_v for a given receiver structure is a remarkable challenge. Both parameters influence the "volumetric effect".

4.1. Measurement of extinction coefficients

The transmission coefficient and thereby the extinction coefficient for several wire samples is determined by using a luminance camera and a light plate, shown in Fig. 8.

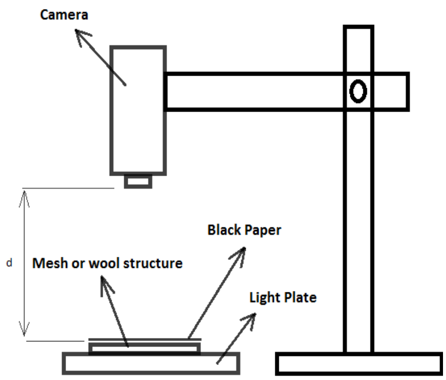

Fig. 8. Experimental setup to determine the extinction coefficient

For different thicknesses of wire meshes and grids the attenuation of radiation was determined. First results confirm eq. 2 in principle. Detailed results will be published in a consecutive paper.

4.2. Measurement of volumetric heat transfer coefficient and assessment of results

Within the framework of the project a test bed for the determination of volumetric heat transfer coefficients α_v of porous structures was developed. The process flow diagram with basic information is shown in Fig. 9.

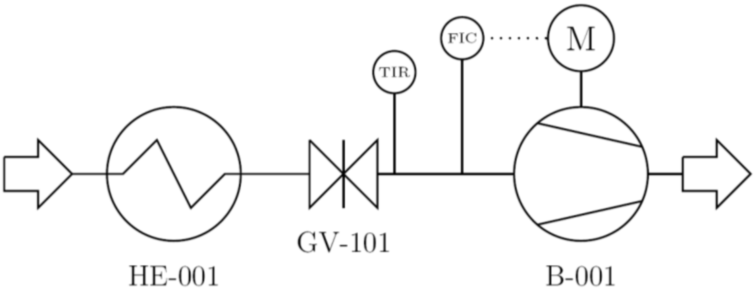

Fig. 9. Process flow diagram.HE-001: Heat Exchanger, GV-101: Gate Valve, B-001: Blower [10]

A transient experiment was chosen: Air is sucked in by the blower (B-001) through the heat exchanger (HE-001) where it is heated up. The stream is guided to the inlet of the gate valve (GV-101) through a piping system. The gate valve offers the opportunity to bring four separate specimens abruptly into the flow.

Hot air flows through the test bench until steady state conditions are reached. As soon as stationary conditions are established a cold specimen is brought into the flow and heated. The signal response (air temperature) is measured at the outlet of the sample as a function of time (see Fig. 11).

The main advantage of this transient experiment is that only one temperature profile has to be measured. The concept of this experiment was chosen as it can be integrated in an existing test bench. The disadvantage of varying flow patterns during the change of two specimens is counteracted by the gate valve construction which allows fast changing times. This ensures that steady state conditions are only slightly disturbed.

A schematic of the system is shown in Fig. 10.

For the determination of the volumetric heat transfer coefficient αv between air and the wire mesh structure a porous system is defined. The resulting system of differential equations, based on the heat balance for the fluid and solid phase, relates the fluid temperature T_F, the solid temperature T_S, to the heat transfer coefficient α_v.

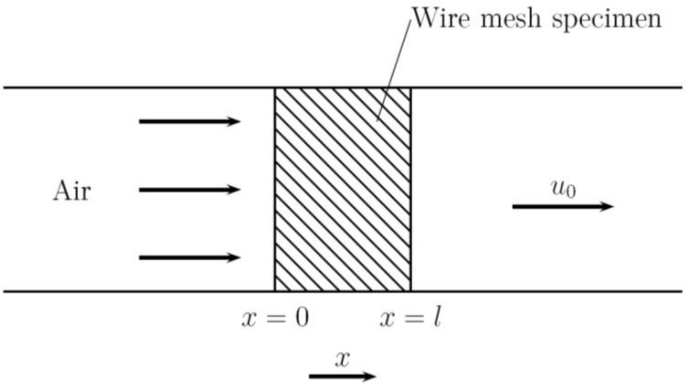

Fig. 10. Schematic of the evaluated system with definition of coordinates [10]

Fluid:
$$\frac{\partial T_F(x,t)}{\partial t} = -u_0 \times \frac{\partial T_F(x,t)}{\partial x} - \frac{\alpha_v \times (T_F(x,t) - T_S(x,t))}{\rho_F \times c_{p,F} \times Po} \quad (12)$$

Solid: $$\frac{\partial T_S(x,t)}{\partial t} = -\frac{\alpha_v \times (T_F(x,t) - T_S(x,t))}{\rho_S \times c_{p,S} \times (1-Po)} \quad (13)$$

u_0 is the superficial velocity of the stream of air, ρ_F and ρ_S the densities for the fluid respectively the solid phase, $c_{p,F}$, $c_{p,S}$ the specific heat capacities for both phases and *Po* represents the porosity of the specimen.

As material constants and geometry factors of equation 12 and 13 are known, the heat transfer coefficient can be determined by adapting the calculated temperature field to the measured one. The only variables to be measured in the experiment are the fluid temperatures at the outlet and at the inlet of the specimen and the fluid flow velocity. Further it is necessary to know the initial temperature of the wire specimen at the very beginning.

Based on these values the partial differential equation system can be solved for varying α_v.

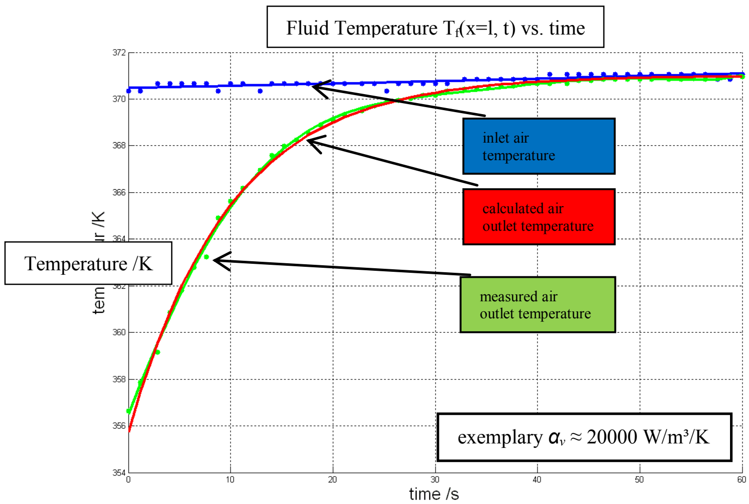

Fig. 11: Exemplary assessment of experimental results of metallic wire specimen

These discrete measurement values are brought into continuous curves by aid of the “Smoothening Spline Function” in Matlab [11]. As result the temperatures can be output at every instant of time. The inlet fluid temperature is used as boundary condition for the solver to calculate the theoretical outlet fluid temperature. The simulated outlet temperature curve is compared with the measured one for a series of different heat transfer coefficients and different initial solid temperatures. The heat transfer coefficient is determined at the simulated curve with best accordance to the measured curve.

This means the shape of the simulated curve is changed by varying the heat transfer coefficient until the sum of squared residuals of both curves shows a minimum (see Fig. 11). In the case of Fig. 11 the volumetric heat transfer coefficient was determined to α_v = 20 kW/(m³ K). Following the laws of similarity such values can by transferred to other geometry and temperature situations.

5. Conclusion

In the paper the key quantities of open volumetric receivers are discussed. They are the penetration depth of solar radiation, the volumetric heat transfer coefficient and the fluid mass flow. A radiation model and a simplified model for dominating light radiation are presented. It is shown that the volumetric effect is seen for the ratio $\alpha_v/(\kappa \times \dot{M})>1$. In typical experiments it is demonstrated that there is no volumetric effect for high air flow rates. First experimental activities were started to determine the basic quantities, volumetric heat transfer coefficient and extinction coefficient.

Acknowledgements

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