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Effects of Vertically Ribbed Surface Roughness on the Forced Convective Heat Losses in Central Receiver Systems

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Abstract. External receiver configurations are directly exposed to ambient wind. Therefore, a precise determination of the convective losses is a key factor in the prediction and evaluation of the efficiency of the solar absorbers. Based on several studies, the forced convective losses of external receivers are modeled using correlations for a roughened cylinder in a cross-flow of air. However at high wind velocities, the thermal efficiency measured during the Solar Two experiment was considerably lower than the efficiency predicted by these correlations. A detailed review of the available literature on the convective losses of external receivers has been made. Three CFD models of different level of detail have been developed to analyze the influence of the actual shape of the receiver and tower configuration, of the receiver shape and of the absorber panels on the forced convective heat transfer coefficients. The heat transfer coefficients deduced from the correlations have been compared to the results of the CFD simulations. In a final step the influence of both modeling approaches on the thermal efficiency of an external tubular receiver has been studied in a thermal FE model of the Solar Two receiver.

Keywords: Molten salt, external receiver, convective losses, Solar Two, CFD, FEM

INTRODUCTION

Most recent project development news suggest that central receiver systems using molten salt as a heat transfer fluid will turn to an industry standard in the near future. Molten Salt projects that are currently in the construction or commissioning phase include Crescent Dunes, the 110 MW_{el} plant from SolarReserve (USA) and Abengoa's 110 MW_{el} Planta Solar Cerro Dominador in Chile. These systems use external receiver configurations together with a surround field. External receivers are profiting of lower spillage losses and higher field efficiency compared to a cavity receiver configuration. On the other hand external receivers are directly exposed to the ambient and are thereby more susceptible to forced convective losses.

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DETERMINATION OF FORCED CONVECTION LOSSES

The forced convective heat transfer can be calculated using heat transfer coefficients. These can be computed from experimentally deduced correlations of the Nusselt number or by CFD analysis.

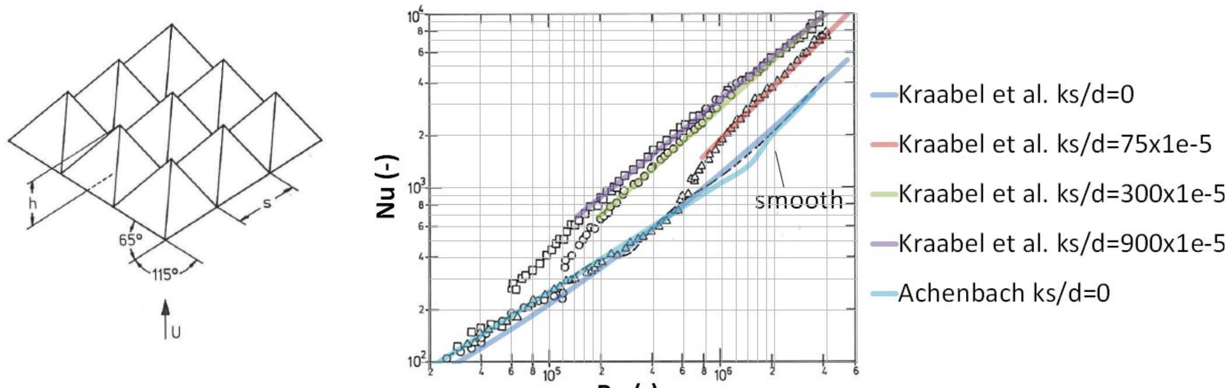
There exist multiple publications on correlations for a smooth cylinder in a cross-flow of air. These were consolidated by Churchill and Bernstein into one single comprehensive equation (1). Already in 1975, Achenbach [2] analyzed the influence of surface roughness on the heat transfer of a cylinder to the cross-flow of air. Based on his results, he deduced a correlation for the Nusselt number as a function of surface roughness. In his experiments the surface roughness was realized as a regular arrangement of pyramids, each having a rhomboidal base, as shown in Fig.1.

He discovered that the surface roughness noticeably influences the Nusselt number for flows in the critical and supercritical flow range. In the subcritical flow range the Nusselt correlation for a smooth cylinder and the experimental results were in good agreement. He published the following set of equations for the prediction of an average Nusselt number as a function of the Reynolds number and of the surface roughness.

$$\frac{k_s}{D} = 75 \cdot 10^{-5}: \quad \begin{array}{ll} 10^4 < Re_D \leq 10^6 & \text{smooth cylinder correlation} \\ 10^6 < Re_D \leq 4 \cdot 10^6 & Nu_D = 2.57 \cdot 10^{-3} Re_D^{0.98} \end{array} \tag{1}$$

$$\frac{k_s}{D} = 300 \cdot 10^{-5}: \quad \begin{array}{ll} 10^4 < Re_D \leq 3 \cdot 10^5 & \text{smooth cylinder correlation} \\ 3 \cdot 10^5 < Re_D \leq 4 \cdot 10^6 & Nu_D = 0.135 \cdot Re_D^{0.89} \end{array} \tag{2}$$

$$\frac{k_s}{D} = 900 \cdot 10^{-5}: \quad \begin{array}{ll} 10^4 < Re_D \leq 3 \cdot 10^5 & \text{smooth cylinder correlation} \\ 3 \cdot 10^5 < Re_D \leq 4 \cdot 10^6 & Nu_D = 0.0455 Re_D^{0.81} \end{array} \tag{3}$$



(a) (b)

FIGURE 1. Achenbach correlations (a) Roughness pattern used to deduce the Achenbach correlations [1]
(b) Nusselt number as a function of Re and roughness parameter k_s/d, Pr=0.72 (black Symbols: experimental results from Achenbach: x: smooth; Δ: $k_s/d=75x10^{-5}$; o: $k_s/d=300x10^{-5}$; ☐: $k_s/d=900x10^{-5}$)

The following correlations have been published by Siebers and Kraabel [2] for the estimation of the natural (4) and the mixed heat transfer coefficient (5) of a cylinder in the cross-flow of air.

$$N_{uH} = 0.098 \left(\frac{\pi}{2}\right) Gr_H^{\left(\frac{1}{3}\right)} \left(\frac{T_W}{T_\infty}\right)^{-0.14} \tag{4}$$

$$\bar{h} = \left(\overline{h_{nc}^{3.2}} + \overline{h_{fc}^{3.2}}\right)^{\left(\frac{1}{3.2}\right)} \tag{5}$$

In order to account for the geometry of the external receiver, Siebers and Kraabel suggested using the Nusselt correlations published by Achenbach. Thereby the influence of the vertical receiver tubes is modeled by the means of a surface roughness k_s/d.

Where k_s is the radius of a single receiver tube and D the diameter of the cylinder. Compared to the results published by Achenbach, Siebers and Kraabel increased the range of validity of Achenbach's correlation for supercritical flow to lower Reynolds numbers.

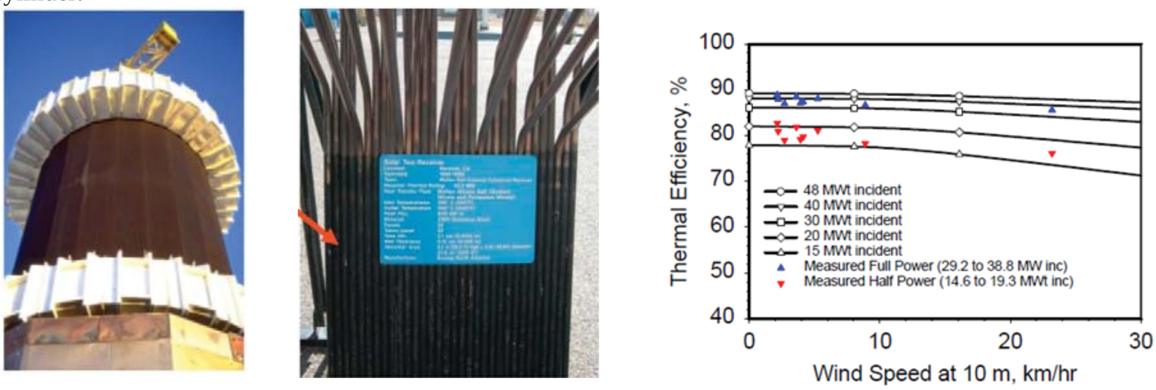
They also added a correlation for high Reynolds numbers based on the assumption that there exists an upper limit for the effect of surface roughness and that for a lower surface roughness this limit will be reached at higher Reynolds numbers. For cases with subcritical flow, where surface roughness does not noticeably influence the Nusselt number, they recommend using Churchill and Bernstein's correlation for a smooth cylinder in a cross-flow of air.

However, the approach of Siebers and Kraabel does neither account for the effect of the corners of adjoining panels, nor for the vertically ribbed surface roughness pattern, as found in external tubular receivers. Another concern is the influence of the variable fluid properties and spatially variable boundary conditions on pure forced convection. For these reasons, Siebers and Kraabel ascribe a relatively high uncertainty to their approach.

In 2012 Christian et al. published his results of a CFD model, which was used to evaluate and characterize the convective heat losses from the Solar Two receiver. He showed that the mixed convective losses were ~5-10% higher for a distributed heat flux compared to a uniform heat flux on the receiver. However the receiver was modeled as a smooth surface in this study [4].

THE SOLAR TWO RECEIVER EXPERIMENT

Because of their commercial purpose, only little performance data exists about the receiver technology used in recent molten salt projects. One well documented example of such an external receiver is the demonstration plant Solar Two, which was operated during the late 90's by the Department of Energy in Daggett, California. The receiver of the Solar Two plant served as a basis for the modeling in this paper. At the Solar Two receiver, the molten salt was heated inside 24 absorbing panels, each composed of 32 parallel tubes, mounted in a vertical fashion. The panels were arranged in a polyhedron shape, thereby approximating a cylinder.



(a) (b) (c)

FIGURE 2. The Solar Two experiment (a) receiver [1] (b) panel (c) Efficiency as a function of wind speed [1]

The efficiency of the receiver was measured using the so called "power-on" method [1]. With this method the measured receiver efficiency can be monitored within an uncertainty range of -2.5 to +1.4% [1]. Receiver efficiencies up to 88.8% were measured at the Solar Two receiver test facility. Combined radiative, convective and conductive thermal losses were measured between 2.7 and 3 MW_{th} [1]. Unfortunately no isolated data of convection losses is available from the Solar Two receiver test.

The recommendations for the calculation of convective losses of Siebers and Kraabel were used in the performance calculation of the Solar Two receiver (3). During non-windy test conditions the simulated and measured receiver conditions showed very good agreement. Nevertheless it must be mentioned, that for low wind speeds, the convective losses represent only a small portion of the losses, which is lower than the measurement error range mentioned above. During windy conditions the measured receiver performance showed deviations greater than 20% from the simulated performance. The receiver efficiency during windy conditions was identified as one of the potentially most significant uncertainties in performance modeling. [1]

CFD-MODELING

Modelling Approach

To analyze the influence of the absorber tubes on the convective heat transfer, a detailed CFD model is necessary. Even though computational power increased in the last years, a full three-dimensional model considering the real surface would still lead to unrealistically large models. The Solar Two receiver is built by 24 panels each with 32 absorber tubes, resulting in total 768 absorber tubes. In order to achieve a satisfying resolution of the boundary layer, the mesh close to the walls must be very fine. Indeed, the distance between the first fluid node and the wall has a significant influence on the accuracy of the heat transfer calculation. Due to expansion ratio limits of 1:20, the mesh density cannot grow very fast. As a consequence a mesh for the full receiver would need hundreds of millions of elements. In order to avoid unrealistic computational efforts, three different models where used within the presented analysis:

1. 3D complete receiver with absorbers modeled as a flat 24 corner polygon, insulation and target.
2. 2D slice of the receiver with absorbers modeled as a 24 corner polygon.
3. 2D slice of the receiver with absorbers modeled with ribbed surface representing the absorber tubes.

The first model was used to analyze the free convection heat transfer coefficient and to compare them with correlation (4). Furthermore this model was used to compare the results (wind cases) with the results from the second model. This comparison is necessary as the 2D slice models implicate that the absorber is an endless long polygon. The limited length of the receiver and the real geometry of the insulation may influence the flow field. The third model is used to simulate the heat transfer coefficients for the ribbed surface of the receiver. The models are shown in Fig 3.

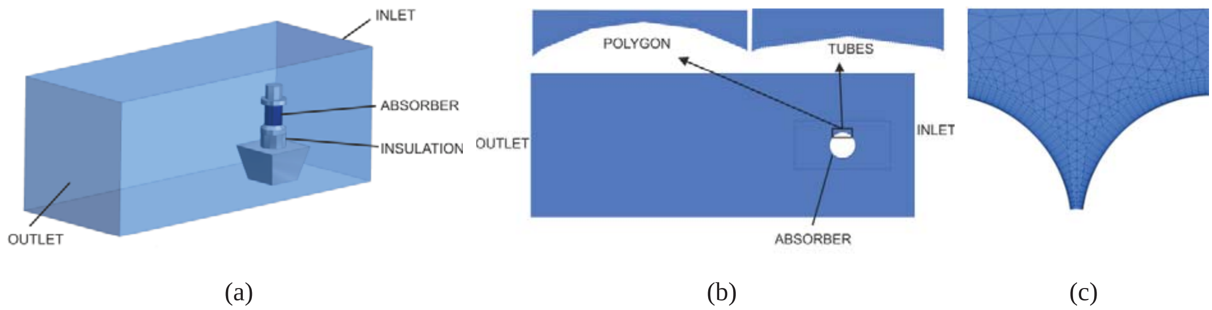


FIGURE 3. CFD models (a) 3D complete receiver; (b) 2D slice model; (c) Boundary layer mesh

Model Description

At the inlet the air velocity is defined normal to the surface. In order to obtain a numerically stable model, the outlet is modelled as an open boundary, thus allowing fluid to flow inwards and outwards. All other faces of the ambient air are modelled with a symmetry boundary. The Shear Stress Transport (SST) turbulence model was used. In order to capture the Kármán vortex street, the model was solved in a transient fashion. Because the heat transfer of forced convection is mainly influenced by the velocity and less by temperature, the absorber tube walls are modelled with a fixed temperature of 500°C. At this point it must be emphasized that the aim of the CFD model is not to determine the heat losses of the receiver. The purpose of the CFD model is the deduction of the Nusselt number for different wind speeds. The thermal losses of the receiver are calculated using a thermal FEM model, utilizing the resulting Nusselt numbers to compute the convective heat loss.

Quality Assurance

The quality of the CFD simulations was ensured by the following best practice guidelines. The modeled size of the surrounding ambient air volume was increased until the influence on the results could be neglected (<0.1%). The mesh independence of the results was checked by a mesh study, comparing the results of three different meshes. An error of 0.25% is accepted in the used mesh densities. The distance of the first fluid node to the wall was chosen to be 1.5e-5m to guarantee an average dimensionless wall distance (y^+) < 1 at highest wind velocity.

THERMAL MODELING

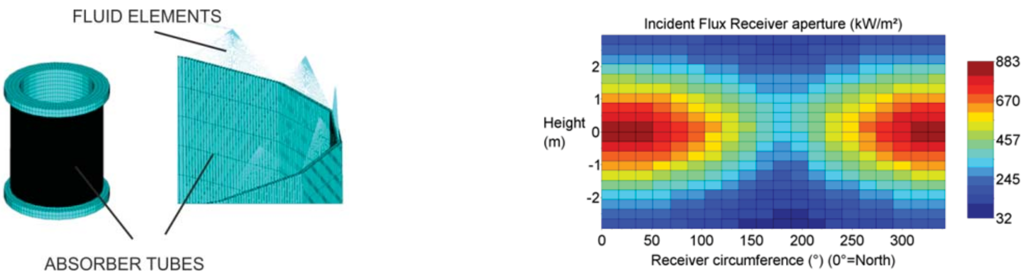
Model Description

A three-dimensional thermal FE- model of the absorber tubes and of the insulation was used to analyze the wind influence on the receiver efficiency. The model considers the main thermal boundary conditions, such as absorbed solar radiation, convective heat transport to the solar salt and the thermal losses by radiation, conduction and convection to the ambient. The thermal model was used to validate the heat transfer coefficients deduced from the CFD simulations by simulating three measurement points of the solar two experiments. Furthermore the model was used to compute the influence of high wind velocities at design point on the thermal efficiency of the Solar Two receiver.

Boundary Conditions

Solar Radiation

The solar radiation reflected by the heliostat field on the receiver aperture is not only dependent on the sun's azimuth and elevation angle, but is also influenced by other random optical circumstances such as mirror reflectivity, tracking errors etc. The heat flux absorption capability of a molten salt receiver panel is limited. Also in the case of the Solar Two heliostat field, the maximum deliverable heat flux exceeds this limit. For this reason the peak heat flux is reduced by using aiming points for the heliostat field. This leads to an inhomogeneous and complex heat flux distribution on the receiver. Based on the published data on the Solar Two heliostat field [1], a representative heat flux distribution on the receiver aperture was computed using a ray tracing code. The resulting absorbed solar radiation is iteratively scaled until the outlet temperature of the receiver is reached. Following the assumptions made for the Solar Two Tests, an absorption coefficient of 95% is applied for computing the reflection losses [1].



(a) (b)

FIGURE 4. (a) Thermal FE- model (b) Flux map

Heat Transfer to Fluid

The heat transfer to the fluid is modeled using one-dimensional fluid flow elements allowing mass and thermal transportation. Consequently the fluid temperature is depending on the heat transferred to the fluid. The fluid elements are located within every absorber tube and are thermally connected to the inner surface of the tube. The heat transfer coefficients are calculated corresponding to the mass flow in the tube using the Gnielinski Nusselt correlation [5]. For the computation of the fluid properties the temperature-dependent correlations published in [1] are used. The fluid flow through the 24 receiver panels is modelled according to the receiver flow path shown in [1]. Perfect thermal mixing of the fluid is assumed inside the headers.

Thermal Radiation

The thermal radiative exchange between the absorber tubes and the insulation as well as the radiation to the ambient is considered by the radiosity method. The view factors were calculated separately for each panel to reduce computational efforts. The emissivity of both the absorber tubes (ε_{IR}=0.83) and the insulation (ε_{IR}=0.84) as well as the ambient temperature was modelled according to the numbers published in [1].

Convection Losses

The convection losses are considered on half of the tube surface (ambient side) using a heat transfer coefficient and the ambient temperature from [1]. The heat transfer coefficients deduced from the Nusselt number correlations from the literature [3] are considered to be homogenous over the circumference and height of the absorber surface. Great care must be taken when using Nusselt correlations from the literature. The correlation given by Achenbach uses the diameter of the macroscopic cylinder as a characteristic length. However in the FEM-Model the heat transfer coefficient is applied to the elements of the absorber tubes. Therefore the values of the heat transfer coefficients of Achenbach have to be divided by a factor of $\frac{\pi}{2}$. The heat transfer coefficients deduced from the CFD simulations vary over the receiver circumference depending on the wind direction. For the comparison of the simulated thermal losses with the measured thermal losses of the Solar Two receiver, the wind direction was modelled in agreement with the published test data.

RESULTS

CFD Simulation

Comparison of Complete 3D Model with 2D model

Because the modelling of the absorber tube area by a complete three-dimensional model would be to complex, a two-dimensional model was used to calculate the heat transfer to the ambient. However this approach implicates an infinitely long absorber. For this reason, the influence of this modelling error is analyzed by comparing the two-dimensional model with neglected tube surface (CFD model 2) with the complete receiver model (CFD model 1).

It was found, that the heat transfer coefficient of the three-dimensional complete receiver model is smaller than for the two-dimensional model. The difference is nearly linearly dependent on wind speed (5% at 5 m/s up to 15% at 20m/s). This could be explained by the influence of the real geometry of the receiver on the flow field. While the two-dimensional model shows a distinctive Kármán vortex street, which is typical for cylindrical structures, the complete receiver model shows an overflow of the fluid at the top of the receiver tower. In addition, the pyramidal structure at the bottom of the receiver influences the flow field at the absorber area. Both effects lead to a reduction of the transition Kármán vortex street near the top and the bottom of the absorber (Fig.5). For this reason, the heat transfer coefficients used for the thermal receiver simulations were reduced by the wind velocity dependent function mentioned above. It should be noticed that this effect is probably lower for larger receivers, leading to higher convective losses (as for the smaller Solar Two receiver).

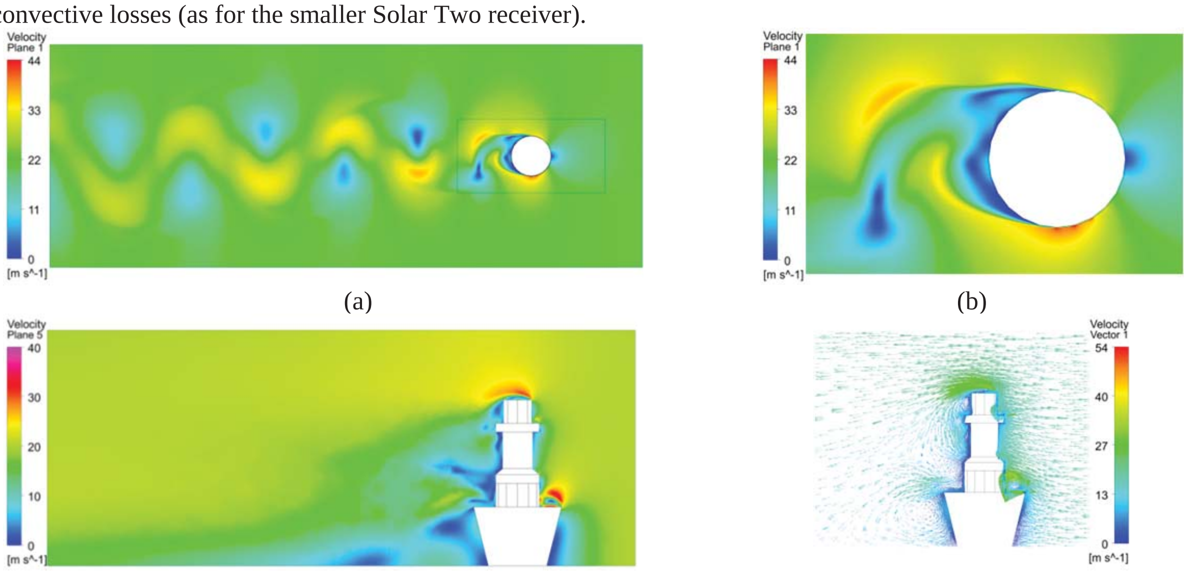


FIGURE 5. CFD results (a,b) Kármán vortex street of 2D slice model (c) 3D complete model velocity field (d) velocity vectors

Comparison of 2D Models (Polygon/Tubes)

The comparison of the two-dimensional model considering the tube surface with the two-dimensional model not considering the tube surface showed that the heat loss is significantly higher for the case with consideration of the tube surface for wind speeds greater than 5m/s. This is because a greater surface area participates in the heat exchange. At a wind velocity of 20 $\frac{m}{s}$ the Nusselt numbers show up to 20% higher values for the ribbed surface model.

Model Validation

The heat transfer coefficients from Achenbach's correlation, as well as the values deduced from the CFD model were used to simulate three test points of the Solar Two experiments. The three chosen test dates represent a no wind case, a low wind case with 3.4 $\frac{m}{s}$ (at receiver height) and a medium wind case with 8.9 $\frac{m}{s}$ (at receiver height). The mass flow of the molten salt, the ambient temperature and the wind direction were modelled according to the published test data [1]. The measured outlet temperature of the receiver was computed iteratively by scaling the heat flux during the thermal simulations.

Correlation (5) from Siebers and Kraabel [2] was used to model mixed convection. The thermal efficiency was calculated according to the definition given in [1]. Where α is the receiver absorptivity (0.95), $L_{thermal}$ is the thermal loss and P_{abs} is the heat transferred to the molten salt.

$$\eta = \frac{\alpha}{1+\frac{L_{thermal}}{P_{abs}}} \tag{6}$$

The simulated results are in a good agreement with the measurements. In general the CFD based results show higher heat losses than the correlation-based results. The only available test data with higher wind speed (23-Mar-99) shows the highest deviation from the measurements. Indeed, the measured efficiency is given within an uncertainty range of -2.5 to +1.4% [1]. This could explain the deviation of the CFD simulation with the test data. (It seems to be illogical that a higher speed leads to lower losses).

TABLE 1. Measurements Solar two experiments [1]

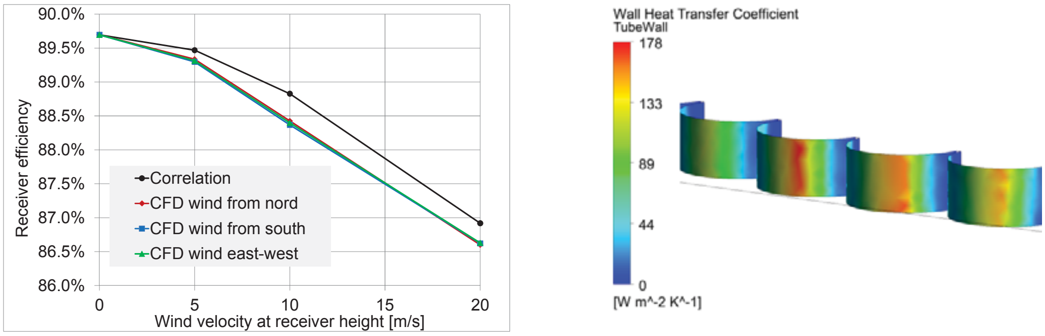
Value	29-Sep-97	5-Mar-99	23-Mar-99
Inlet temperature [°C]	295	308	302
Outlet temperature [°C]	551	564	561
Mass flow [kg/s]	80	81	61
Ambient temperature [°C]	32	16	16
Wind velocity at receiver height [$\frac{m}{s}$]	0.67	3.4	8.9
Wind direction clockwise from north	131	270	263
Heat transferred to salt ($L_{thermal}$) [MW]	31.6	31.5	24.2

TABLE 2. Comparison of measurement and thermal modelling

Value	Measurement	CFD-based	Correlation-based
29-Sep-99			
Thermal loss [MW]	2.27	2.18	2.18
Efficiency [%]	88.8	89.0	89.0
5-Mar-99			
Thermal loss [MW]	3.04	2.44	2.34
Efficiency [%]	86.7	88.4	88.6
23-Mar-99			
Thermal loss [MW]	2.75	2.77	2.58
Efficiency [%]	85.6	85.8	86.3

Receiver Efficiency at Full Load

The CFD results indicate that at higher wind velocities the heat loss may be higher than estimated by correlations. For this reason the thermal FEM model of the Solar Two receiver was used to analyze the receiver efficiency at full load (full mass flow of 101 kg/s, design outlet temperature of 565 °C, ambient temperature of 20°C) for different wind speeds and wind directions. At 20 $\frac{m}{s}$ the CFD based results show an efficiency drop from 89.7% to 86.6%, the correlation-based results estimate instead an efficiency of 86.9%. Even if the CFD results show a strong circumferential dependence according to the wind direction and a very inhomogeneous local heat transfer coefficients at the tube surface (Fig. 6b), the results show that this influences are negligible (Fig. 6 a)regarding the efficiency. In part load situations (clouds, start up) the influence of inhomogeneous local heat transfer coefficients could become critical due to local freezing of the salt.



(a) (b)

FIGURE 6. (a) Receiver efficiency at full load for different wind directions (b) Local heat transfer coefficient

SUMMARY AND OUTLOOK

Three different CFD models and a thermal FEM model were used to analyze the influence of wind velocity and direction on the receiver efficiency of the Solar Two receiver. Main attention was drawn on the influence of the ribbed surface of the receiver. Correlations from the literature were compared with the simulated results. It was found, that the real surface should be taken into account for efficiency calculations at higher wind speeds. Especially during startup and shutdown or during cloud transients the circumferential and local heat transfer coefficients should be taken into account in order to avoid freezing. Further work should be done to validate the model for higher wind velocities as there is no measurement data available. The presented efficiency drop is calculated at the design point so far. Part load situations and the local wind situation should be considered for annual return calculations. For larger receivers the effect of the ends could be lower, resulting in higher convection losses. The presented modelling approach is currently used at the development of a 700 MW_{th} high performance molten salt receiver.

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