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Scaling effects of a novel solar receiver for a micro gas-turbine based solar dish system

Authors: Lukas Aichmayer^a, Jorge Garrido^a, Björn Laumert^a

^a Department of Energy Technology, Royal Institute of Technology, SE-100 44 Stockholm

Contact Details:

Lukas Aichmayer: lukas.aichmayer@energy.kth.se

Jorge Garrido: jorge.garrido@energy.kth.se

Björn Laumert: bjorn.laumert@energy.kth.se

Corresponding Author: Lukas Aichmayer, +46 (0) 8 790 7467

Abstract

Laboratory-scale component testing in dedicated high-flux solar simulators is a crucial step in the development and scale-up of concentrating solar power plants. Due to different radiative boundary conditions available in high-flux solar simulators and full-scale power plants the temperature and stress profiles inside the investigated receivers differ between these two testing platforms. The main objective of this work is to present a systematic scaling methodology for solar receivers to guarantee that experiments performed in the controlled environment of high-flux solar simulators yield representative results when compared to full-scale tests. In this work the effects of scaling a solar air receiver from the integration into the OMSoP full-scale micro gas-turbine based solar dish system to the KTH high-flux solar simulator are investigated. Therefore, Monte Carlo ray-tracing routines of the solar dish concentrator and the solar simulator are developed and validated against experimental characterization results. The thermo-mechanical analysis of the solar receiver is based around a coupled CFD/FEM-analysis linked with stochastic heat source calculations in combination with ray-tracing routines. A genetic multi-objective optimization is performed to identify suitable receiver configurations for testing in the solar simulator which yield representative results compared to full-scale tests. The scaling quality is evaluated using a set of performance and scaling indicators. Based on the results a suitable receiver configuration is selected for further investigation and experimental evaluation in the KTH high-flux solar simulator.

1. Introduction

Hybrid solar micro gas-turbines (MGTs) show potential to supply controllable power on demand to households in remote areas by using solar energy in combination with a back-up fuel (such as locally derived biodiesel). Additionally, the high exhaust temperature of the MGT opens up the possibility of supplying additional services, such as heating, cooling and water purification, through the use of polygeneration technologies [1]. Therefore, the EU-funded Optimised Microturbine Solar Power system (OMSoP) project [2] was established aiming at demonstrating a 3-10 kW_{el} stand-alone

hybrid MGT solar dish system. In a first phase a solar-only system has been installed at the ENEA Casaccia Research Center and commissioning tests have been successfully performed. So far, the tests have been conducted below nominal operation conditions.

In general, interest in using parabolic dish concentrators arises due to the potentially high concentration ratios and high optical efficiencies which allow to reach high receiver temperatures effectively [3] [4]. Solar dish concentrator installations of various sizes and shapes have demonstrated peak flux levels ranging from 1-12 MW/m² [5] [6] [7] [8] [9]. Previous efforts on the power conversion unit for dish systems have mainly focused on the integration of Stirling engines [10] [11] as compared to this work where a MGT is integrated. Previous studies of volumetric air receivers for gas-turbine applications have focused on designs for tower-based systems [12] [13] [14] [15] [16] which are not directly applicable for parabolic dish units. Furthermore, these receivers are typically designed for large-scale applications, with sizes ranging from 250 kW_{el} [17] up to around 25 MW_{el} [18] as compared to the OMSoP MGT dish system with an electrical output in the range of 3-10 kW_{el}. For low power ranges (15-300 kW_{el}) Rosa do Nascimento [19] presents a comprehensive summary on MGTs.

As the solar receiver needs to operate under high temperatures and high fluxes it is one of the key component in such concentrating solar power (CSP) systems. In earlier work [20] a new receiver concept was designed for the integration into a MGT solar dish with boundary conditions derived from the Eurodish concentrator [7]. In this work the concept is adapted to fit the specific OMSoP boundary conditions [21] [22]. The development process of solar receivers for CSP plants often involves the testing of a scaled version in a laboratory environment with controllable testing conditions and reduced system complexity as compared to the full-scale system. In recent years, high-flux solar simulators (HFSSs) have gained importance for the development of CSP components and systems as they offer a fully controllable and constantly available testing environment at lower costs than using a full-scale solar platform [23] [24]. A recent review of solar simulators [25] showed that 26 HFSSs of various designs were in operation in late 2016 with 7 of them capable of providing a radiative power of more than 15 kW and 3 capable of providing more than 30 kW at the focal plane. In early 2017 another HFSS

was completed offering radiative power of up to 310 kW [26]. With the vast majority of the operational HFSSs delivering a radiative power below 30 kW [25] the main challenge of HFSS testing is caused by the fact that the radiative power delivered is lower than the power available at a full-scale CSP system. Additionally, differences of the peak flux and the spatial flux distribution can appear between HFSSs and full-scale systems. These differences result in different temperature and stress profiles within the solar receiver. The same issue arises for the evaluation of receivers in the scaling-up process of a CSP plant when building the actual system is impractical due to long lead times, too costly and/or impossible as the final design is not fully established and changes are anticipated. The need of laboratory tests and scaling is not only limited to solar dish systems but can also be found in tower based CSP plants. Even the most powerful HFSS in operation is not able to provide the full power needed for large utility-scale receivers with radiative power requirements in the order of 500 kW to several MW [12] [13].

In light of the above, the main objective of this work is to determine a systematic scaling methodology for solar receivers and their radiative and power cycle boundary conditions to guarantee that experiments performed in the controlled environment of a HFSS yield representative results when compared to full-scale tests. To achieve this goal, the effects of scaling are investigated comparing the integration of a scaled solar receiver into the KTH HFSS and a full-scale receiver integrated into the OMSoP solar dish system which provides different flux and radiative power. For an objective evaluation the scaling quality is assessed using a set of performance and scaling indicators.

2. Solar receiver and integration

2.1. Solar receiver concept

In this work a receiver concept developed in previous work [20] is adapted to operate under the measured OMSoP solar dish MGT boundary conditions. It then serves as the basis for the investigation of scaling effects between the full-scale receiver for the integration in the OMSoP solar dish and a scaled version for testing in the KTH HFSS. The receiver concept is outlined in Figure 1 which is based around a porous absorber (red) that is placed behind a transparent glass window (yellow) to separate the working fluid from the exterior while allowing the concentrated solar radiation to pass through. By absorbing the radiation within its volume the absorber heats up which in turn heats the air passing through the absorber. Preheated air enters the receiver at the side, is distributed in the inflow mixing box, is redirected towards the glass window cooling it, and finally passes through the absorber as indicated by two exemplary streamlines (black dotted line). The design outlet temperature for air exiting the receiver at the back is 800°C or higher. The exact specifications of the reference receiver and its performance are presented in section 5.

2.2. Receiver integration

The reference receiver is directly integrated in a recuperated MGT power conversion cycle whereas the receiver for the KTH HFSS is coupled with a gas-turbine simulator which allows recreating the boundary conditions present across the operating range of the MGT cycle and offers full controllability which simplifies the system control.

Figure 1: Solar receiver concept (based on [20])

Figure 2 shows the OMSoP solar dish with an outer diameter of 11.73m [27] located at the ENEA Casaccia Research Center close to Rome which provides the concentrated solar radiation input to the full-scale receiver whereas Figure 3 shows the lamp array of KTH HFSS which provides the radiative power input to the scaled receiver. The solar simulator [23] [24] is based around a circular array of 12 high-power xenon-arc lamps fitted with Fresnel lens concentrators and is currently being used for testing and development of solar receivers. It offers a fully controllable and constantly available testing environment. This innovative Fresnel lens design was chosen over the more conventional approach of using ellipsoidal reflectors to reduce costs as these reflectors are not commercially available, as well as to avoid lifetime issues previously encountered [28] and to avoid the magnification effect of the ellipsoidal reflector causing the focal spot size and the optical efficiency to be highly dependent on the size of the emitter arc [29].

Figure 4 presents the integration of the solar receiver with the solar dish and the MGT in a recuperated power conversion cycle. Ambient air is compressed in the compressor to 3 bar before being preheated in the recuperator to 540°C at the design operating point. Solar power provided by the dish is used to heat

up the air to 800°C in the solar air receiver before being expanded in the turbine. The air exiting the turbine is sent to the recuperator to pre-heat the compressor air.

Figure 2: OMSoP solar dish

Figure 3: KTH solar simulator

In order to obtain relevant test results of the scaled solar receiver both the radiative and power cycle boundary conditions need to be recreated. The xenon-arc lamps of the HFSS provide the radiative input to the solar receiver that is placed in the proximity of the focal point. Instead of coupling the solar receiver directly to a MGT a gas-turbine simulator has been implemented which allows recreating the operating range of the MGT power conversion cycle. The gas-turbine simulator ensures constant and controllable boundary conditions while reducing the complexity and facilitating system control.

Figure 5 presents the integration of the solar receiver within the HFSS and the gas-turbine simulator which is based on a combination of a mass flow controller (MFC) and a choked nozzle capable of recreating the off-design behavior of a MGT. This combination allows the pressure level between the mass flow controller and the nozzle (where the receiver is placed) to change with varying mass flow similarly to the MGT operating characteristic [30]. Compressed air supplied by a 40 bar air compressor with a maximum continuous capacity of 100 g/s is sent to the air quality control devices (AQC) where it is filtered by a grade 70C filter and the pressure is reduced to 16 bar before entering the Bronkhorst mass flow controller with an operating range of 0-400 m_n^3/h ($\approx$ 0-143.7 g/s). The exit pressure varies between 1 and 4 bar depending on the mass flow and air temperature. At the nominal operating point the receiver pressure is 3 bar. A custom built 60 kW electrical air heater featuring 4 Leister LE 10000 DF-R HT units placed inside individual pressure vessels is used to pre-heat the compressed air to 540°C before entering the solar receiver. Radiative power supplied by the solar simulator lamps increases the temperature to 800°C before being expanded in the choked nozzle and being dissipated into the ventilation system.

Figure 4: OMSoP solar dish layout

Figure 5: Solar simulator layout

Choked flow [31] is a compressible flow effect where the mass flow through a constriction such as a nozzle is limited by the exit velocity when it reaches sonic conditions (for adiabatic conditions). At choked conditions the mass flow through the nozzle can only be increased by increasing the upstream density. For this work, the advantage of using a nozzle operating at choked conditions is that the pressure in the ventilation system downstream of the nozzle does not affect the upstream behavior provided that the critical pressure ratio is exceeded. Assuming ideal gas behavior, a critical upstream pressure of 1.85 bar needs to be exceeded for an upstream air temperature of 800°C and a downstream pressure in the region of 1 bar.

Figure 6 outlines the radiative boundary conditions at the focal plane provided by the KTH HFSS (red line) and the solar dish concentrator of the OMSoP project (blue line) used for the scaling analysis in this work. A more detailed description of the flux characterization is presented in section 4. It can be seen in Figure 6 that the peak flux of the solar dish concentrator is in the region of 1.4 MW/m² which is significantly lower compared to the solar simulator peak flux in the region of 7 MW/m² designed to match modern dish systems. On the other hand, when looking at the mean flux within a realistic aperture diameter of 200 mm it can be seen that the solar dish concentrator delivers a larger radiative power. For the two testing platforms investigated the radiative power delivered at the focal plane within an aperture diameter of 200 mm is 25.3 kW and 17.6 kW, respectively.

Figure 6: Rotational symmetric flux at the focal plane

3. Analysis process

The analysis process applied in this work is outlined in Figure 7 and is split in three main parts: the optical evaluation of the two testing platforms, the receiver evaluation using a coupled CFD/FEM-approach as well as the scaling evaluation.

The optical evaluation is based on Monte Carlo ray-tracing routines calibrated with experimental data of the solar dish concentrator and the HFSS and yields a bundle of rays at the focal plane of the solar receiver which is used as the input for the receiver evaluation. These rays are used in order to calculate the radiative boundary conditions for the coupled CFD/FEM-analysis of the receiver whereas thermal radiation is accounted for separately. This approach improves the radiative boundary

conditions compared to previous work where an analytic approach assuming parallel light rays was used [20]. Additionally, the power cycle boundary conditions of both testing platforms and a set of design parameters are used as input. The coupled CFD/FEM-analysis yields a set of performance indicators (as defined in section 6.1) which are passed on to the scaling evaluation.

Within the scaling evaluation the scaling quality is assessed using these performance indicators in combination with additional results of the coupled CFD/FEM-analysis which yields a set of scaling indicators. These scaling indicators are used as input for an optimization process which alters selected receiver design parameters and the receiver performance is evaluated again. Finally, the optimization results are used to calculate a Pareto-optimal frontier which allows the selection of a number of receiver designs which ensure that testing results obtained in the KTH HFSS are representative compared to tests in the full-scale solar dish system

Figure 7: Analysis process

4. Optical evaluation

To evaluate the receiver performance precise knowledge of the radiative boundary conditions of the solar dish and the HFSS is imperative. For accurate calculations the 3-dimensional flux distribution around the focal point is required which is later used to calculate the heat sources providing power to the solar receiver. In practice this 3-dimensional flux profile is created by

measuring the 2-dimensional flux distribution at planes with varying axial offset from the focal plane.

In this work experimental data from the solar dish and the HFSS is used to validate ray-tracing simulations which are then applied to calculate the heat input to the solar receiver. Figure 8 shows the schematic overview of the validation process. The experimental flux profiles at different planes are compared with the flux profiles obtained by MATLAB-based ray-tracing routines based on the Monte Carlo method. An optimizer varies a number design parameter to minimize the error between the measurements and simulations described in detail in the calibration section 4.3. As a result, the position and direction vector of all rays at the focal plane of the solar dish and the HFSS are created and later used in the receiver evaluation.

Figure 8: Schematic overview of the optical evaluation

4.1. Experimental setup

Experimental campaigns have been carried out to characterize the performance of the OMSoP solar dish and the KTH HFSS.

4.1.1. Solar dish

Montecchi et al. [22] characterized the solar dish performance installed at the ENEA Casaccia Research center using the VISdish tool for canting and shape-measuring of solar-dish facets in combination with the ray-tracing tool SIMULDISH to predict the flux distribution around the focal spot. During the experimental campaign the surface error of 149'736 measurement points on the dish was determined in horizontal and vertical direction. These surface error measurement results were combined with the experimental reflectance characterization of the solar dish mirror to predict the flux distribution around the focal point with the ENEA tool SIMULDISH. The results of this simulation are presented in the following sections.

For this work the probability density function (PDF, cp. [32]) of the surface error was determined for both the zenith and azimuth angle based on the experimental data provided by Montecchi et al. [22]. The results for 200 bins are shown in Figure 9 with the distribution of the zenith angle error at the top

and the azimuth angle error at the bottom. The zenith angle error PDF presents a typical shape (cp. [33]) with a characteristic peak around 5 mrad. Moreover, it can be seen that the azimuth angle error PDF is relatively constant over the entire angular range. For the MATLAB based ray-tracing used in in this work the azimuth angle error PDF is therefore considered to be constant.

Figure 9: Probability density function of the surface error [22]

4.1.2. Solar simulator

The characterization of the HFSS was performed by point-wise flux measurements and calorimetric power measurements [34]. The point measurements were taken using water-cooled thermopile flux sensors placed behind a water-cooled copper shield mounted on a 3 axis linear unit as shown in Figure 10. Two flux sensor with different measurement ranges were used to ensure relevant results at high and low flux levels. In order to get as close to a point measurement as possible the exposed area of the flux sensor was minimized by placing the sensor behind a well-defined aperture with a diameter of 5 mm. Reducing the radiative power to the sensor also reduces the risk of overheating. The aperture was manufactured in a conical way with an opening angle of 90° to allow all the rays to enter the aperture as the HFSS features a 90° rim angle.

Calibration of the sensors was performed using the spectral properties of the xenon-arc light source, the back-reflector inside the lamp, the Fresnel lens, and the absorption surface of the thermopile flux sensor as described in previous work [34] [35] [36].

The experiments were performed at 3 planes with an axial offset of 0, 30, and 60 mm from the focal plane away from the lamps, respectively. Additionally, the radiative power was measured at 5 different planes with an axial offset of 0, 10, 30, 60, and 100 mm from the focal plane away from the lamps, respectively. These calorimetric measurements are based around a water-cooled copper plate of 300x300 mm coated with Pyromark 2500 with spectrally highly constant absorption

properties. The results of the point-wise and calorimetric characterization are presented in the following sections.

Figure 10: Point measurements thermopile flux sensor

4.2. Ray-tracing

In order to calculate the radiative heat input to the solar receivers investigated ray-tracing models of the OMSoP solar dish and the KTH HFSS were developed and implemented using MATLAB based routines. A dedicated 3-dimensional Monte Carlo approach (cp. [37] [38] [39] [40]) was chosen to reduce computing time down to the order of minutes allowing optimization loops which require thousands of evaluations as compared to commercially available ray-tracing software tools with significantly longer calculation times.

4.2.1. Solar dish

In order to match the ray-tracing and the experimental results 10 million points have been stochastically distributed at the dish surface. From each of these points 3 distinct reflections originate accounting a total of 30 million rays.

For each point investigated the ideal reflection vector $\vec{o}_{ideal}$ that passes through the focal point FP is calculated (shown in Figure 11) and the surface error is determined for each individual ray using the experimentally determined PDFs. For the zenith error angle the PDF shown in Figure 9 (top) is used whereas for the azimuth error angle a constant PDF was considered. First, the surface error is calculated in spherical coordinates (zenith angle error $\Delta\theta$ and azimuth angle error $\Delta\phi$) and then the error contribution due to the Sun's angular spread is added. Finally, the result is converted into Cartesian coordinates and the direction vector of the ray determined. The gray lines in Figure 11 represent 3 distinct reflections whereas the dotted black lines and circle illustrate a cone containing the possible range of deviations after the reflection. This includes the deviations due to the surface error and the sun spread. To calculate the flux distribution at any given plane an orthogonal virtual multi-CCD detector is implemented.

In the ray-tracing implemented in this work the dish surface error is determined stochastically according to the PDFs presented in Figure 9 whereas Montecchi [22] considers the measured values of the surface error for each point individually as obtained during the measurement campaign.

Figure 11: Ray-tracing solar dish

The ray-tracing is performed for a parabolic dish concentrator with an outer diameter of 11.73 m and a focal length of 7.04 m. To make space for the pedestal and the arm bearing the receiver and MGT the dish concentrator was constructed with an inner hole with a diameter of 2.12 m and a 30° wedge was removed [27].

4.2.2. Solar simulator

The ray-tracing routines implemented for the HFSS are based around the ray-tracing of three lamps located in the first quadrant of the HFSS (cp. [23] [24]). This approach is chosen in order to capture the specific characteristics of the HFSS while at the same time reducing computational time. Ray-tracing is performed for these three lamps and then duplicated for the other 3 quadrants to account for all 12 lamps of the HFSS. Figure 12 shows the layout of a single lamp/lens-combination, where the xenon-arc bulb (red) emits light which is reflected at the parabolic back reflector (blue) and concentrated by the silicon-on-glass Fresnel lens (magenta) onto the focal point FP. In this work, the xenon-arc bulb is modelled as a line source of the xenon-arc between the cathode and anode which are 7.5 mm apart. The arc PDF is characterized by the 1-dimensional brilliance distribution along the center line (based on [41]). Applying this line source model reduces the computational time significantly compared to using a 2-dimensional brilliance distribution and modelling the entire bulb structure which allows the application in the optimization process.

First, initial points are created on the Fresnel lens of each lamp assuming an ideal point light source with constant radiance and an ideal parabolic back reflector without any defects. An exemplary ideal light path is indicated by the dotted line in Figure 12 passing through the point F_1 at the Fresnel lens. Then, in order to account for the line source characteristic of the lamp an axial offset Δz_{arc} is stochastically added to each ray at its origin according to the brilliance distribution PDF. As a result

of this deviation the rays impinge onto the Fresnel lens at point F_2 at a distance Δr from the ideal point F_1 as shown by the solid light path in Figure 12.

Refraction inside the Fresnel lens is based on Snell's law assuming an infinitesimally thin lens and only a single refraction in order to simplify the model. To capture the spectrally dependent refraction each incoming ray is split into 10 rays of equal power. Therefore, the light spectrum impinging onto the lens is split into 10 segments of equal power based on the spectral properties of the xenon-arc light source and the back-reflector [35] [36]. For each of these segments the weighted average refractive index is calculated based on fused silica glass and the refraction is determined for each ray leaving the Fresnel lens.

Figure 12: Schematic layout ray-tracing lamp

In order to match the ray-tracing and the experimental results a stochastic error is introduced after the refraction at the Fresnel lens. For each ray leaving the Fresnel lens a zenith angle error angle is added based on a PDF discretized into 30 segments for angles between 0.5 and 80 mrad while assuming a constant azimuth angle error PDF. To keep the complexity at a reasonable level the same PDF is applied to the 3 lamps investigated. Moreover, the axial offset of the xenon-arc bulb Δz_{bulb} and the individual misalignment towards the focal point of the three lamp/lens-combinations simulated are considered. It was found during the experimental campaign that the flux distribution is highly sensitive towards these parameters. The individual misalignment from the ideal focal point mainly impacts the flux distribution at planes behind the focal plane and was considered for all 3 lamps modelled. Changes in the bulb offset strongly influence the peak flux of each individual lamp and consequently the combined peak flux and a single value for the 3 lamps modelled was considered. Additionally, the radiative power leaving each of the 12 individual lamps was assumed to be the same with a value determined in the calibration process.

4.3. Calibration

Both the OMSoP solar dish and the HFSS ray-tracing were calibrated against measurement results. Ray-count sensitivity studies were performed to ensure that the results are stochastically reliable.

4.3.1. Calibration solar dish

The MATLAB based Monte Carlo ray-tracing for the solar dish is calibrated using the experimental characterization of the solar dish surface error and the complete ray-tracing work undertaken by Montecchi [22]. The calibration was implemented for a direct normal irradiance (DNI) value of 800 W/m² which is the design value for the OMSoP solar dish installed at the ENEA Casaccia Research Center.

The power at the solar dish focal plane was chosen as the main calibration parameter which was determined by Montecchi to be 54.5 kW for the chosen DNI. During the calibration the effective reflectivity of the dish concentrator was adjusted to 73.7% in order to match the power at the focal plane. This effective reflectivity accounts for the reflectivity of the panels and the spacing between them.

In Figure 13 the solar dish results of the complete ray-tracing performed by Montecchi [22] (denoted by the solid lines) are compared with the MATLAB based ray-tracing developed in this work (denoted by the dashed lines). It can be seen that the flux profiles of both simulations for planes with an axial offset of 0, 30, and 60 mm to the focal plane match very well. Applying the MATLAB based ray-tracing yields a radiative power of 54.5 kW at the focal plane (within a diameter of 500 mm) and a radiative power of 25.3 kW inside an aperture diameter of 200 mm.

Figure 13: Rotational symmetric flux at the different planes of the solar dish

4.3.2. Calibration solar simulator

The HFSS ray-tracing is calibrated using the flux measurements taken with the water-cooled thermopile flux sensor and the calorimeter. To match the measurement results and the ray-tracing the MATLAB genetic optimization algorithm *ga* [42] is used to minimize the error between the ray-tracing and the measurements. As the algorithm requires thousands of evaluations the amount of initial points on the Fresnel lens was limited to 300 thousand resulting in 3 million rays per lamp and a total of 9 million rays for the lamps modelled of the HFSS.

For the optimization the 30 parameters of the discretized stochastic error function applied to the lamp/lens-combination, the axial offset of the xenon-arc bulb (1 parameter), and the individual misalignment towards the focal point (3 parameters) are used. As the objective function the sum of the absolute error between the measurement points and the ray-tracing in the three planes is used denoted δ in Equation (1). E_m and E_{RT} denote the radiative flux obtained by the point measurement and the ray-tracing, respectively, r the radius, z the planes with different axial offset from the focal plane fp , r_{ap} the aperture diameter of the solar receiver, and $E_{m,ct}$ the measured flux at the center line. The ratio at the end of equation is used to increase the weight of the error at planes further away from focal plane as these planes present flux levels that are a closer match to levels encountered at the focal plane of the solar dish.

$$\delta = \sum_{z=1}^3 \sum_{r=0}^{r_{ap}} |E_m(z, r) - E_{RT}(z, r)| \cdot \frac{E_{m,ct}(fp)}{E_{m,ct}(z)} \quad (1)$$

Additionally, the results of the caloric power measurement are used to calibrate the power of the HFSS ray-tracing. The results of the optimization procedure are presented in Figure 14 and Figure 15. For the results presented the optimizer determined a lamp offset from the ideal focal point of 2.74 mm, 0.14 mm, and 0.13 mm for the 3 lamps, respectively, and a bulb offset of 0.46 mm from the ideal focal point of the back reflector. The optimizer also showed that an average radiative power of 1.45 kW leaving each individual lamp is needed to match the radiative power measured at the focal plane.

Figure 14 shows the results of the point measurements performed with the thermopile flux sensors of the HFSS (denoted by the solid lines) which are compared to the results of the ray-tracing routines implemented (denoted by the dashed lines). It can be seen that there is good agreement between the measurements and ray-tracing for the planes with an axial offset of 0 and 30 mm to the focal plane. In the 60 mm offset plane small peaks at a radius of 17.5 mm and 45 mm can be observed in the ray-tracing which have not been captured by the point measurements. These peaks are consistent with the peaks expected due to the individual lamps with a tilt angle of 17° and 37° of the inner and outer lamp array, respectively.

Figure 14: Rotational symmetric flux at the different planes of the HFSS

Figure 15: Radiative power and flux as a function of the focal plane offset and aperture diameter

In Figure 15 the flux at the center line of the receiver and the maximum flux for different focal plane offsets are shown and denoted by the black solid and black dashed line, respectively, with values shown on the axis to the left. For receiver focal planes close to the HFSS focal plane the maximum flux occurs at the center line of the receiver. For offsets larger than 35 mm the contribution of the individual lamps gets visible and the maximum flux occurs away from the center line.

Furthermore, the power received within a specific receiver aperture D_{ap} calculated by the ray-tracing is shown in Figure 15 for different focal plane offsets as indicated by the red solid lines with their values shown on the axis to the right. The calorimetric measurements are shown by the blue squares with an uncertainty of approximately $\pm 500W$ and compared to the

calculated power within a square with a side length of 300 mm as indicated by the blue solid line also on the axis to the right. It can be seen that there is good agreement between the measurements and ray-tracing. For the relevant flux levels above 1.5 MW/m^2 which corresponds to a receiver focal plane offsets of up to approximately 60 mm the relative error remains below 2% which is within the uncertainty of the measurements. At the receiver focal plane offset of 100 mm the relative errors increases to 3.4%.

4.4. Rays on focal plane

Based on the calibration results the position and direction vector of all the rays at the focal plane were determined and stored to be used in the receiver evaluation. In order to increase the quality of the heat source calculations inside the solar receiver the amount of rays was increased compared to the calibration step.

5. Receiver evaluation

The solar receiver evaluation is based around a coupled CFD/FEM analysis which uses input from the optical evaluation. It yields a set of performance indicators which are used in the scaling evaluation.

5.1. Coupled CFD/FEM analysis

The receiver performance as well as material temperatures and material stresses for all critical components are evaluated in the multi-physics software tool COMSOL using a 2-dimensional rotational symmetric coupled CFD/FEM approach as described and validated in the previous work [20]. In this work the heat source calculations are based around the 3-dimensional ray-tracing routines calibrated with measurement results which yield 2-dimensional rotational symmetric heat sources as compared to the analytic approach applied in the previous work.

5.2. Power cycle boundary conditions

The OMSoP design point operating conditions are used as the power cycle boundary conditions for the solar receiver installed in the solar dish and are shown in Table 1. The ray-tracing models presented in section 4.2 show that the solar dish delivers 25.3 kW of radiative power inside a aperture diameter of 200 mm. Therefore, a receiver efficiency of approximately 80% is sufficient to heat the air from 540°C to 800°C .

Cycle operating conditions		
Air mass flow $\dot{M}$	[g/s]	70
Receiver inlet temperature T_{in}	[$^\circ\text{C}$]	540
Receiver outlet temperature T_{out}	[$^\circ\text{C}$]	800
Receiver operating pressure p_{rec}	[bar]	3.0

Table 1: Solar dish power cycle boundary condition

In order to account for the different radiative boundary conditions between the solar dish and the solar simulator the

power cycle boundary conditions need to be adjusted. For this study the fluid inlet and outlet temperatures as well as the operating pressure are kept constant while the air mass flow is adjusted.

5.3. Radiative boundary conditions

In order to evaluate the receiver performance, the power input to the system needs to be determined. Absorption of concentrated radiation within the porous absorber constitutes the main power input while the absorption inside the glass window and on other surfaces inside the receiver are significantly smaller. To couple the absorption of concentrated radiation with the other heat transfer phenomena (conduction, convection, and thermal radiation) a volumetric source term is used in the heat balance equation. In this study a Monte Carlo approach using the path length method [43] is employed to calculate the volumetric heat source contrary to the analytic approach in the previous analysis [20]. The path of each individual ray is traced inside the solar receiver starting from the front surface of the glass window as shown in Figure 16. Depending on the starting point and direction vector the ray passes through the glass window (denoted g) and through the cavity (denoted c) before entering the absorber (denoted a) or impinging on a secondary structure (denoted o).

For each ray the end point is calculated and all its power contributes to the incremental heat source associated with its position. In this evaluation five possible light paths are considered (R_1 - R_5) which contribute to 5 different heat sources: the heat source within the glass window $\dot{q}_{v,g}$, the volumetric heat source inside the porous absorber $\dot{q}_{v,a}$, the surface heat source at the absorber limiting wall $\dot{q}_w$, the heat source of rays outside the absorber aperture $\dot{q}_{v,o}$, and the heat source of rays passing the absorber $\dot{q}_{v,p}$. The indices i and j in Figure 16 denote the discretization parameters of the heat sources in radial and axial direction, respectively.

Figure 16: Absorber light paths

5.3.1. Glass window heat source

The heat source inside the glass window $\dot{q}_{v,g}$ is calculated using Equation (2) where E denotes the flux distribution at the receiver front plane calculated by the ray-tracing routines described in section 4.2 and $\bar{\alpha}_{tot}$ the averaged total absorption of all the rays passing through the glass window.

$$\dot{q}_{v,g}(r) = E(r) \cdot \bar{\alpha}_{tot} \quad (2)$$

For this work Suprasil CG a high purity synthetic fused silica material manufactured by Heraeus was chosen. Due to its high transmittance from the deep UV to the near IR and excellent physical properties [44] [45] the axial influence of the absorption within the glass window is negligible.

The absorption as well as the reflection and transmission are calculated individually for each ray impinging onto the glass window and are averaged at the end. The influence of averaging these properties is negligible for a sufficiently large number of rays. To obtain the absorption coefficient in Equation (2) the behavior of all rays impinging onto the front surface of the glass window is evaluated. In this work two reflections are considered: one on the front surface and one at the back surface as shown for an exemplary ray R_3 in Figure 16. For two reflections the total reflection ρ_{tot} , the total absorption α_{tot} , and total transmission τ_{tot} through the glass for each individual ray can be calculated using Equations (3), (4) and (5) (cp. [46]). The base 10 for the extinction coefficient was chosen as the optical data available uses this base [44] [45].

$$\rho_{tot} = R + R \cdot (1 - R) \cdot 10^{-2 \cdot K_{ext} \cdot s} \quad (3)$$

$$\alpha_{tot} = (1 - R) \cdot (1 - 10^{-K_{ext} \cdot s}) + R \cdot (1 - R) \cdot (10^{-K_{ext} \cdot s} - 10^{-2 \cdot K_{ext} \cdot s}) \quad (4)$$

$$\tau_{tot} = (1 - R)^2 \cdot 10^{-K_{ext} \cdot s} \quad (5)$$

In these equations R denotes a single reflection at the boundary between air and glass (or vice-versa), K_{ext} the extinction coefficient of the glass, and s the path length of a single ray from one side of the glass window to the other.

As incidence angles reach up to 53° their influence on the reflection has to be taken into account. The reflection R is calculated for each individual ray using the Fresnel equations for unpolarized light [47] as shown in Equations (6) where the index *per* and *par* indicate perpendicular and parallel polarized light, respectively. In Equations (7) and (8) the index *i* and *t* denote incident and transmitted properties, respectively, n the refractive index, and θ the angle between the light ray and the surface normal $\vec{n}$ of the glass window. The transmitted angle θ_t is calculated using Snell's law.

$$R = \frac{1}{2} \cdot (R_{per} + R_{par}) \quad (6)$$

$$R_{per} = \left(\frac{n_i \cdot \cos(\theta_i) - n_t \cdot \cos(\theta_t)}{n_i \cdot \cos(\theta_i) + n_t \cdot \cos(\theta_t)} \right)^2 \quad (7)$$

$$R_{par} = \left(\frac{n_i \cdot \cos(\theta_t) - n_t \cdot \cos(\theta_i)}{n_i \cdot \cos(\theta_t) + n_t \cdot \cos(\theta_i)} \right)^2 \quad (8)$$

To accurately calculate the reflection, absorption and transmission at the glass window the spectral dependency of the extinction coefficient and the refractive index of the glass window need to be considered as the spectral distribution of the of the light impinging is not uniform. In this work this is done by calculating the weighted average of the extinction coefficient and the refractive index according to the SolarPACES guidelines [48] inside the wavelength range of 200 to 2500 nm. By using the weighted average of the refractive index it is possible to determine the reflection R for each ray with an error of less than 0.015% as compared to calculating the reflection for each individual wavelength and then performing the weighted averaging. Similarly, the error committed by using the weighted average of the extinction coefficient instead of calculating the internal transmission for various path length is negligible.

In Equation (9) the weighted averaging of an exemplary variable ψ is shown where ψ_λ denotes the spectral distribution of the variable, i_λ the light source spectral distribution, and $\Delta\lambda$ the constant difference between the wavelength steps.

$$\bar{\psi}_w = \frac{\sum_\lambda \psi_\lambda \cdot i_\lambda \cdot \Delta\lambda}{\sum_\lambda \psi_\lambda \cdot \Delta\lambda} \quad (9)$$

The optical property calculations are based on spectral data for Heraeus Suprasil CG [44] [45] for a 10 mm thick specimen and are presented in Figure 17 denoted by the magenta lines. The dash-dotted magenta line indicates the refractive index, the solid magenta line the transmission, and the dotted magenta line the internal transmission, respectively. The difference between the internal transmission and the transmission is the reflection loss. The extinction coefficient for a base 10 is indicated by the dashed magenta line with the axis shown on the right.

Figure 17: Glass window optical properties

Additionally, Figure 17 shows the normalized spectral distribution of the light source in the HFSS indicated by the black solid line based on the spectral properties of the Xenon-arc light source, the parabolic back reflector, and the Fresnel lens presented in previous work [35] [36]. For the solar dish the spectral light distribution of terrestrial sunlight for an air mass of 1.5 is employed [49] as shown by the solid blue line (normalized). The red line in Figure 17 denotes the normalized thermal spectrum of a black body at 1000°C which is used to determine thermal radiation absorption (section 5.4).

For terrestrial sunlight with an air mass of 1.5 a weighted average of 1.4548 is obtained for the refractive index whereas for the HFSS light source a weighted average of 1.4555 is obtained. Within the significant wavelength range of 200 and 2500 nm the variation of the refractive index of air is negligible [50]. Similarly, the extinction coefficient of the glass window is calculated as the weighted average using the spectral extinction coefficient of Suprasil CG [45] and the spectral radiation data of the solar dish and solar simulator. For a base 10 the extinction coefficient is determined to be 0.04 m⁻¹ and 0.11 m⁻¹ for the HFSS and the solar dish, respectively. Applying Equations (3), (4) and (5) for a 10 mm thick glass window for all rays the averaged total absorption $\bar{\alpha}_{tot}$ used in Equation (2) is close to the absorption of perpendicular impinging light with values in the region of 0.11% and 0.25% for the HFSS and the solar dish, respectively.

5.3.2. Heat sources inside the receiver

After passing the glass window the rays continue inside the solar receiver where a minor part of rays (R_I) passes outside of the absorber aperture D_a and are absorbed at a secondary object, such as the inlet tube wall, contributing to the heat source $\dot{q}_{v,o}$. The remaining rays enter the solar absorber and are absorbed inside the volume, impinge on the wall surrounding the absorber (denoted w), or pass through the absorber depending on the path length inside the porous absorber. This path length of each ray is calculated stochastically using the path length method [43] based on the standard radiative transfer equation which was shown to be applicable in porous solar absorbers in previous work as the conditions for treatment of the absorber as a continuum are fulfilled (see [51]). Scattering effects are neglected as their influence is small due to the relative short heat exchange distance inside the porous absorber [51]. Spectral dependencies inside the absorber are neglected as well to simplify the model and reduce computing time. Applying these assumptions, the path length s inside the porous absorber can be calculated using Equation (10) where χ denotes a random variable between 0 and 1 and K_{ext} the extinction coefficient of the absorber. Rays that are absorbed within the volume contribute to the volumetric heat source $\dot{q}_{v,a}$. The extinction coefficient for open cell materials such as porous absorbers can be calculated using Equation (11) where ϕ denotes the cell diameter, ε_p the porosity, and φ a material constant, which takes a value of 4.8 for silicon carbide [52].

$$s = -\frac{\log(1 - \chi)}{K_{ext}} \quad (10)$$

$$K_{ext} = \frac{\varphi}{\phi} (1 - \varepsilon_p) \quad (11)$$

Rays that impinge on the side wall of the absorber are either absorbed (R_2) or reflected (R_3) calculated using Equation (12) where χ denotes a random variable between 0 and 1 and α the absorptivity of the wall. For this analysis the austenitic superalloy 253MA is chosen for the metallic components of the receiver due its operating temperature of up to 1150°C, its machinability, weldability, and availability for the manufacturing of a prototype. As no absorptivity values are available for this superalloy it is estimated by the total hemispherical emissivity of fully oxidized Inconel 718 for temperature in the region of 1000°C as 90% [53] [54]. Rays that are absorbed on the absorber side wall contribute to the surface heat source $\dot{q}_w$ whereas the rays reflected continue their path and are subsequently absorbed inside the volume or pass through the absorber altogether. In this work the absorber thickness is chosen in a way that only approximately 1% of the rays pass through the absorber.

$$\alpha = \begin{cases} 1 & \chi \leq \alpha \\ 0 & \chi > \alpha \end{cases} \quad (12)$$

Rays that make it all the way through the absorber (R_5 and parts of R_3) are finally absorbed by the outlet tube contributing to the heat source $\dot{q}_{v,p}$. Table 2 summarizes the different light paths possibilities considered in this work with the main determining property and the heat source the ray contributes to.

	Case	Prop.	Source
1	Out of absorber aperture	D_a	$\dot{q}_{v,o}$
2	Absorbed on receiver wall	α	$\dot{q}_w$
3	Reflected on receiver wall	$1 - \alpha$	-
4	Absorbed within absorber	s	$\dot{q}_{v,a}$
5	Passing absorber	L_a	$\dot{q}_{v,p}$

Table 2: Solar absorber cases

To simplify the ray-tracing the end position of rays outside of the absorber aperture and the ones that pass through the absorber is not calculated explicitly. It is assumed that the heat sources $\dot{q}_{v,o}$ and $\dot{q}_{v,p}$ act as volumetric heat sources that heat the air before entering the cavity and after exiting the absorber, respectively.

5.4. Thermal radiation treatment

In this work thermal radiation transfer was considered inside the glass window and the porous absorber. Moreover, the radiation exchange in the cavity between the glass window and the absorber front surface including the surrounding metal parts was modeled using a modified COMSOL surface-to-surface

radiation model to capture the volumetric nature of the glass window.

The radiative heat transfer inside the volume of the glass window and the porous absorber is modelled using the P1-approximation [37] [38]. This approach is chosen as it facilitates the coupling to the other heat transfer phenomena present (convection, conduction, and concentrated radiation absorption) and keeps computational time low which is a requirement for the use in an optimization process. Convection between the solid and fluid is modelled using the same correlations [55] as used in previous work.

The heat transfer of thermal radiation inside the absorber is characterized by the scattering coefficient σ_s and absorption coefficient a which are calculated using Equation (13) and Equation (14), respectively, where ε denotes the emissivity, ϕ the cell diameter, and ε_p the porosity. These equations are based on [56] and are modified with a material constant φ which takes a value of 4.8 for silicon carbide [52]. In this work the scattering and absorption coefficient of the porous absorber are considered to be constant with temperature.

$$\sigma_s = (2 - \varepsilon) \cdot \frac{\varphi}{2\phi} \cdot (1 - \varepsilon_p) \quad (13)$$

$$a = \varepsilon \cdot \frac{\varphi}{2\phi} \cdot (1 - \varepsilon_p) \quad (14)$$

For the calculation of thermal radiation heat transfer inside the glass the absorption coefficient is calculated for different solid temperatures (100-1200°C) as the weighted average using Equation (9) applying Planck's law for the spectral emission and Suprasil CG properties. Scattering effects are neglected as their influence is small compared to the effects of absorption [57].

For the radiation exchange between the absorber front surface and the glass window the absorption of thermal radiation within the glass window is calculated using the emission spectrum of a black body at 1000°C (as shown in Figure 17) and the extinction coefficient for Suprasil CG. The relative absorption is calculated for 100 axial elements of 0.1 mm. For the whole thickness of the glass the total reflection, total absorption, and total transmission is calculated to be 3.85%, 50.78%, and 45.37%, respectively, applying Equations (3)-(8). As the distance between the glass window and the absorber is only 10 mm an incidence angle of 0° was considered and 2 reflections were modelled.

6. Scaling evaluation

As stated in section 1, the aim of this work is to investigate the effects of scaling of an exemplary solar receiver from boundary conditions present in the full-scale OMSoP solar dish system to the boundary conditions in the KTH HFSS. The performance of the solar receiver is highly dependent on these boundary conditions. This analysis is done to guarantee that testing results obtained in the KTH HFSS are representative compared to tests in the full-scale solar dish system.

In this work, the Monte Carlo ray-tracing and coupled CFD/FEM routines are linked with a multi-objective optimizer to evaluate the performance of a variety of scaled solar receiver for testing at the KTH HFSS and are compared to a reference solar receiver designed for the integration in the solar dish system. The scaling quality is assessed using a set of scaling indicators.

In this section the receiver performance indicators are defined, the reference receiver design is presented, the receiver parameters that are varied in the optimization process are shown, and concrete scaling objectives and scaling indicators are derived based on a situation analysis.

6.1. Receiver performance indicators

First, to accurately compare different designs performance indicators are defined. In this work the receiver outlet temperature T_{out} , the receiver efficiency η_{rec} , and pressure drop Δp across the receiver are chosen as the main thermal performance indicators. The receiver efficiency is defined as the ratio of the absolute air enthalpy increase inside the receiver and the radiative power available at the receiver aperture (cp. [20]) whereas the air outlet temperature is calculated as the mass averaged mean temperature to account for spatial non-uniformity. When assuming constant specific heat capacity the mean outlet temperature can be calculated using Equation (15) where ρ denotes the local air density, u the local velocity component normal to the outlet, T the local fluid temperature, and A the outlet area.

$$\bar{T}_{out} = \frac{\int_A \rho \cdot u \cdot T \cdot dA}{\int_A \rho \cdot u \cdot dA} \quad (15)$$

Additionally, to ensure that material limits are not exceeded the maximum temperature and stress of both the glass window and absorber are selected as the most critical material performance indicators as identified in previous work [20]. For the glass window the first principal stress is chosen as the critical performance indicator as the compressive strength of silica glass is a factor 10 larger than the bending strength [44]. As was shown in previous work [20] a modified Rankine stress theory can be used for determining the critical stress of porous foam as presented in Equation (16) where σ_I , σ_{II} , and σ_{III} denote the principal stresses.

$$\sigma_{com} = \max(|\sigma_I|; |\sigma_{II}|; |\sigma_{III}|) \quad (16)$$

6.2. Reference receiver

Secondly, to evaluate the scaling quality of various scaled receivers a reference receiver is needed. As the main objective of this work is the investigation of these scaling effects the reference receiver is obtained using an engineering approach rather than an optimization process.

The reference receiver is designed for the integration into the OMSoP solar dish system working at the design operating conditions [21]. At the design point the receiver increases the air temperature from 540°C to 800°C with a mass flow 70 g/s and an operating pressure of 3 bar. In order to reach this temperature increase a receiver aperture in the region of 200 mm is required for the design DNI of 800 W/m². Both component and systems tests are planned at these design conditions.

Additionally, the procurement process for a ceramic open cell foam showed that no products with a porosity larger than 85% can be purchased at reasonable costs. Therefore, a SiSiC absorber with porosity of 85% a cell diameter of 4 mm was chosen for the reference receiver with a thickness of 50 mm so that only approximately 1% of concentrated solar power passes the absorber. For the glass window a fused silica product of 10 mm thickness provided by Heraeus was chosen. In the reference receiver the absorber is placed 10 mm behind the glass window.

Parameter		Previous [20] p1%-c	Current Ref.-receiver
Inlet temp. T_{in}	[°C]	740	540
Outlet temp. T_{out}	[°C]	909	815
	[g/s]	109	70
Air mass flow $\dot{M}$			
Aperture D_{ap}	[mm]	109	200
Cavity width L_c	[mm]	6.2	10
Window thickness L_g	[mm]	5	10
Porosity ε_p	[%]	90.1	85
Cell diameter ϕ	[mm]	4	4
Window temp. T_g	[°C]	862	702
Window stress σ_I	[MPa]	26.4	23.6
Absorber temp. T_a	[°C]	1087	1110
Absorber stress σ_{com}	[MPa]	3.36	5.93
Material utilization	[%]	84.0	79.6
Receiver efficiency η_{rec}	[%]	86.7	86.2
Pressure drop Δp	[%]	1.16	0.09
	[Pa]	3459	280

Table 3: Solar dish reference receiver parameters

Compared to the prototype suggested in previous work [20] the solar absorber is kept straight to simplify the manufacturing and the experimental evaluation. This is possible as the peak flux of the solar dish concentrator is considerably lower compared to the one considered in the preliminary analysis [20] which was derived from the Eurodish concentrator. Table 3 summarizes the parameters and performance indicators of the previous prototype (p1%-c) with a curved absorber front and the current reference receiver with a straight front. The previous prototype was evaluated using the models presented in previous work and different power cycle boundary conditions were applied compared to the current work. Additionally, the radiative boundary conditions were derived from Eurodish experiments [7] and parallel light inside the receiver was

assumed as compared to the current evaluation using ray-tracing routines and the improved modeling approach presented in section 4 and section 5.

For the OMSoP boundary conditions the efficiency of the reference receiver was determined to be 86.2% which is higher than the design goal of 80% as can be seen in Table 3. Thus the receiver is able to raise the air temperature up to 814.8°C as compared to the design goal of 800°C. When comparing the performance of the previous prototype with the current reference receiver it can be seen that the receiver efficiency as well as the material temperatures and stress levels remain quite constant except for the absorber stress. This is due to the fact that the two receivers use absorbers of different porosities. The row below the absorber stress in Table 3 presents the material utilization (occurring stress divided by permissible stress). It can be seen that the material utilization improves from the previous prototype to the current reference receiver. The absorber temperature profile and third principal stress of the reference receiver is presented in the results section 7. The pressure drop on the other hand decreases significantly due to the decreased mass flow and increased aperture diameter.

6.3. Scaling parameters

In order to obtain representative results during the receiver testing in the KTH HFSS, a number of parameters can be adjusted. The main scaling parameters are identified as the power cycle boundary conditions, the radiative boundary conditions as well as the geometrical and material properties of the solar receiver itself. The power cycle conditions can be adjusted more or less freely using the gas-turbine simulator described in section 2. The radiative boundary conditions are mainly determined by the axial position of the receiver in the HFSS and the receiver aperture size. Geometrical and material properties of the receiver and the porous absorber highly affect the fluid flow field and the radiation absorption.

In this work, the inlet temperature T_{in} and outlet temperature T_{out} as well as the operating pressure are kept constant between the two testing platforms in order to keep the thermal performance comparable. To do so the air mass flow rate $\dot{M}$ through the receiver in the HFSS is adjusted for varying radiative power inputs according to Equation (17) where η_{rec} denotes the receiver efficiency, P_{rad} the radiative power available at a specific focal plane offset Δz within the aperture diameter D_{ap} , and h_{in} and h_{out} the specific enthalpy at the receiver inlet and outlet, respectively. For this calculation the scaled receiver efficiency is considered to be constant and equal to the reference receiver efficiency with a value of 86.2% (as shown in Table 3). However, due to small variations in the efficiency the receiver outlet temperature of the scaled system fluctuates around the reference outlet temperature with deviations in the range of ± 5 K.

In order to adjust the radiative boundary conditions the receiver inside the HFSS is placed at different axial positions with varying offset Δz from the focal plane fp as shown in Figure 18. Additionally, varying the receiver aperture D_{ap} of the solar receiver changes the radiative boundary condition. This

affects mainly the ratio between the maximum flux located close to the center line of the receiver and the flux at the side of the receiver.

$$\dot{M} = \frac{P_{rad}(D_{ap}, \Delta z) \cdot \eta_{rec}}{(h_{out} - h_{in})} \quad (17)$$

The geometrical and material properties investigated are explained based on Figure 18 where E_b denotes the impinging concentrated radiation. As was presented in previous work [20] the receiver concept is based around a porous absorber (a) which is placed at a distance L_c behind the glass window (g). To keep the radiation transfer between the absorber and the glass window similar between the solar dish and the HFSS case the distance L_c was kept constant at 10 mm. On the other hand, the distance L_n created by the inner tube supporting the absorber and the glass window was identified to be an important scaling parameter. Varying this distance strongly influences the fluid flow field and the heat transfer inside the porous absorber and thus the material temperatures and stresses.

Figure 18: Volumetric solar receiver layout

The material properties of the absorber were kept unchanged (porous SiSiC with an average porosity of 85% and a cell diameter of 4 mm) as well as the thickness of the glass window (10 mm).

6.4. Situation analysis

In a next step, to identify the receiver scaling objectives a pre-study was performed calculating the performance of a 125 receiver designs for the integration in the KTH HFSS varying the three scaling parameters mentioned above randomly within the ranges presented in Table 4.

Parameter		Min.	Max.
Axial offset Δz	[mm]	0	125
Receiver aperture D_{ap}	[mm]	80	250
Inlet restriction L_n	[mm]	1	10

Table 4: Scaling parameter range

The results of the situation analysis are shown in Figure 19 where red markers indicate air properties, black markers

absorber properties (a) and blue markers glass window properties (g), respectively. Furthermore, crosses denote temperatures T , squares the first principal stresses σ_I , and circles third principal stresses σ_{III} . The dashed black and blue lines indicate the permissible temperature and stresses of the absorber and glass window, respectively, whereas the red dashed line denotes the desired air outlet temperature of 800°C. For a SiSiC open cell foam with a porosity of 85% and a cell diameter of 4 mm the permissible stress is 7.44 MPa (cp. [20]) and the maximum continuous operating temperature is 1500°C [58]. The permissible stress limit of a glass window made of Suprasil is 67 MPa and temperatures the maximum continuous operating temperature of 950°C [44].

Figure 19: Performance indicators situation analysis

It can be seen in Figure 19 that the air outlet temperature as one of the thermal performance indicators marked by the red crosses varies from approximately 784°C to 828°C between the different designs which is a maximum deviation of 2.6% from the from the desired outlet temperature. The receiver efficiency as the second thermal performance indicator shows variations in the region of 76% to 90%.

Regarding the material performance indicators 50 out of the 125 designs violate at least one of the permissible limits and are therefore infeasible. Without precise selection of the scaling parameters the critical stress inside the absorber exceeds the permissible limit whereas material temperatures in more than 84% of the cases remain below the permissible limit. Moreover, it can be seen that the compressive stress inside the absorber (indicated by the third principal stress σ_{III}) dominates compared to the tensile stress (indicated by the first principal stress σ_I).

When comparing the permissible stress of the glass window with the first principal stress occurring it can be seen that all cases show a safety margin of at least 45% (factor of safety close to 2). The maximum glass window temperature observed is for 88% of the cases below the permissible temperature limit of 950°C. However, for a robust design Röger et al. [59] suggest

an even lower continuous operating temperature for fused silica in order to avoid degradation due to recrystallization at high temperatures, especially in atmospheres with nucleus-forming contaminants.

6.5. Scaling indicators

Based on the results of the pre-study performed the third principal stress profile inside the absorber was selected as the main scaling criterion. The main objective of the scaling evaluation is therefore to minimize the difference between the absorber stress profile of the scaled solar receiver in the HFSS and the solar receiver mounted in the solar dish system. Additionally, in order to expose the volumetric absorber to a similar peak mechanical load the difference between the maximum stress of both systems is to be minimized.

To mathematically assess the differences between the two-dimensional rotational symmetric stress profiles and to fulfill the scaling objectives three scaling indicators are defined.

- The mean stress surface error $\Delta\bar{\sigma}$ measures the overall scaling quality (Equation (18)).
- The maximum stress surface error $\Delta\sigma_{max}$ measures local deviations which are not captured by the mean value due to the large number of points averaged (Equation (19)).
- The stress point error $\Delta\hat{\sigma}_{pt}$ is used to guarantee similar mechanical loads. It is defined as the absolute error between the minimum third principal stress in the HFSS receiver and the minimum third principal stress in the solar dish receiver (Equation (20)).

$$\Delta\bar{\sigma} = \frac{\sum |\sigma - \sigma_{ref}|}{numel(\sigma_{ref})} \quad (18)$$

$$\Delta\sigma_{max} = \max(|\sigma - \sigma_{ref}|) \quad (19)$$

$$\Delta\hat{\sigma}_{pt} = |\min(\sigma) - \min(\sigma_{ref})| \quad (20)$$

In Equation (18)-(20) bold properties indicate a two dimensional matrix, the index *ref* denotes properties of the reference (solar dish) receiver, and *numel* the total number of elements of the matrix. To compare stress profiles of absorbers of different sizes the results were normalized to the reference diameter of the solar dish receiver.

Additionally, secondary scaling indicators using the volumetric solar absorber material temperature profile instead of the stress profile are defined by replacing σ by T in Equation (18)-(20).

6.6. Optimization setup

The optimization used for the scaling effects evaluation is based around the MATLAB genetic algorithm *gamultiobj* [42] coupled to the MATLAB based ray-tracing and heat source calculation routines and the CFD/FEM-analysis in COMSOL. The evaluation is performed on a Linux-based network utilizing distributing computing.

As optimization parameters the axial offset Δz of the receiver from the HFSS focal plane, the receiver aperture D_{ap} , and the distance L_n restricting the air inlet to the cavity between the glass window and the absorber were chosen. The same parameter range as in the situation analysis (see Table 4) is chosen for the optimization parameters. The scaling indicators described in Equation (18)-(20) are selected as the objective functions.

7. Scaling results

As stated in section 1, the goal of this work is to guarantee that the solar receiver testing results obtained in the KTH HFSS are representative compared to tests in the full-scale solar dish system. Therefore, the ray-tracing and coupled CFD/FEM routines are used to evaluate the performance of an exemplary solar receiver designed for the integration in the solar dish system and compared to the performance of a variety of scaled solar receiver for testing at the KTH HFSS. The scaling quality is assessed using the scaling indicators defined in section 6.5: the mean stress surface error $\Delta\bar{\sigma}$, the maximum stress surface error $\Delta\sigma_{max}$, and the stress point error $\Delta\hat{\sigma}_{pt}$.

7.1. Optimization results

A multi-objective optimization was applied in order to identify suitable receiver configuration for the KTH HFSS that yield representative results compared to tests in the full-scale solar dish system. In this section the result of this multi-objective optimization with 2000 receiver evaluations is presented.

Figure 20: Multi-objective optimization Pareto-surface

Based on the optimization a Pareto-surface for the three scaling indicators is determined and shown in Figure 20. The two objective functions mean stress surface error $\Delta\bar{\sigma}$ and maximum stress surface error $\Delta\sigma_{max}$ are shown on the abscissa and ordinate, respectively, whereas the third objective function

(the stress point error $\Delta\hat{\sigma}_{pt}$) is shown on the z-axis and additionally is indicated by the color bar. For better clarity Figure 20 only presents the Pareto-surface close to the minimum of the objective functions while other solutions exhibit objectives of up to one order of magnitude larger.

As can be seen in Figure 20 the Pareto-optimal solution for three objective functions is characterized by a surface which is difficult to interpret. Therefore, distinct Pareto-optimal frontiers of the first two objective functions (mean and maximum stress surface error) were determined for points below a limit of 25, 10, and 8% of the third objective function (stress point error) as shown in Figure 21 where the color of the line represents the stress point error. Due to this definition it is possible that points seem to be located in front of the Pareto-optimal frontier but with a larger stress point error. Additionally, in Figure 21 all evaluations performed are shown as compared to the Figure 20 where only the Pareto-optimal points are presented. The three selected Pareto-frontiers are also shown in Figure 20 (marked by a black arrow) and it can be seen that they lie on the Pareto-optimal surface.

Figure 21: Multi-objective optimization Pareto-frontiers

7.2. Selected cases

Based on the optimization, three potentially interesting receiver designs lying on the Pareto-optimal frontiers have been selected for further investigation. Cases ‘B’ and ‘C’ are located at the extremes of the 10 and 8% stress point error frontier, respectively, while case ‘A’ is located in the middle of the 25% frontier before the stress point error increases significantly.

7.2.1. Scaling indicators of selected cases

Table 5 summarizes the scaling parameters, as well as the main and secondary scaling indicators of these selected cases. Additionally, in Figure 22 the third principal stress profile is

shown on the left and the temperature profile is shown on the right for the reference receiver ‘R’ and the selected cases ‘A’, ‘B’, and ‘C’. The gray and black crosses in the contour plots denote the minimum third principal stress of the absorber and the maximum absorber temperature, respectively. These are the values used for the calculation of the stress point error $\Delta\hat{\sigma}_{III,pt}$ and the maximum temperature point error $\Delta\hat{T}_{pt}$, respectively. The gray and black plus signs denote the position of the maximum stress surface error $\Delta\sigma_{III,max}$ and the maximum temperature surface error ΔT_{max} , respectively.

		A	B	C
D_{ap}	[mm]	132.96	132.34	132.69
Δz	[mm]	43.97	41.40	41.11
L_n	[mm]	6.383	6.314	6.305
$\Delta\bar{\sigma}_{III}$	[kPa]	396.57	392.42	397.00
$\Delta\sigma_{III,max}$	[MPa]	2.10	2.20	2.36
$\Delta\hat{\sigma}_{III,pt}$	[kPa]	928.74	546.05	449.75
	[%]	15.67	9.21	7.59
$\Delta\bar{T}$	[K]	12.19	13.36	13.87
ΔT_{max}	[K]	47.21	45.21	47.84
$\Delta\hat{T}_{pt}$	[K]	43.55	20.35	15.29

Table 5: Scaling indicators and scaling parameters of selected cases

It can be seen in Table 5 that all selected cases have an aperture diameter in the region of 130 mm, are placed with an axial offset in the region of 40 mm in the HFSS, and have a restricting distance L_n in the region of 6 mm.

Differences of the mean stress surface error $\Delta\bar{\sigma}_{III}$ remain below 1.1% between the three cases whereas the maximum stress surface error $\Delta\sigma_{III,max}$ varies up to 12.4%. This shows the importance of the maximum stress surface error for the evaluation of the scaling quality as the mean error is not able to capture large local deviations. Figure 22 visualizes the difference of the third principal stress σ_{III} between the reference receiver ‘R’ and selected cases. The maximum stress surface error $\Delta\sigma_{III,max}$ which is in the region of 2 MPa occurs close to the absorber side and front surface as indicated by the gray plus signs in Figure 22. Finally, the stress point error $\Delta\hat{\sigma}_{III,pt}$ varies 51.6% between the cases. In Table 5 and Figure 21 the absolute value of the stress point error was normalized using the minimum third principal stress of the reference receiver.

The difference of the mean temperature surface error $\Delta\bar{T}$ remains below 13.8% while the maximum temperature surface error ΔT_{max} varies up to 4.2% and the stress point error varies up to 64.2% between the different cases. Additionally, it can be seen in Table 5 that the maximum temperature point error $\Delta\hat{T}_{pt}$ decreases with decreasing stress point error $\Delta\hat{\sigma}_{III,pt}$ with case ‘C’ showing the smallest temperature point error of all the cases presented.

Figure 22: Absorber third principal stress and temperature

7.2.2. Performance indicators of selected cases

In Table 6 the performance indicators as defined in section 6.1 are presented for the reference receiver 'R' and the three selected receiver cases for the HFSS. It can be seen that the thermal performance indicators outlet temperature T_{out} and

receiver efficiency η_{rec} remain almost constant between the reference receiver and the cases 'A', 'B' and 'C' (variations below 2.2%) whereas the pressure drop across the receiver Δp varies up to 10.2%.

The maximum porous absorber temperature $T_{a,max}$ as one of the material performance indicators remains almost constant too (variations below 3.2%) whereas the minimum third principal

stress $\sigma_{a,min}^{III}$ shows larger variations of up to 15.7%. All cases presented show material stresses below the permissible stress limit of 7.44 MPa (SiSiC foam, $\varepsilon_p = 85\%$, $\phi = 4\text{mm}$, cp. [20]) and temperatures below the maximum continuous operating temperature of 1500°C [58]. The maximum glass window temperature as one of the material performance indicators presents variations of up to 3.9% whereas the maximum first principal stress $\sigma_{g,max}^I$ shows the largest deviations of up to 57%. Again, all cases presented show material stresses below the permissible stress limit of 67 MPa and temperatures below the maximum continuous operating temperature of 950°C [44]. For short term operation the material is able to handle temperatures up to of 1200°C [44].

		R	A	B	C
T_{out}	[°C]	814.78	818.01	818.37	818.34
η_{rec}	[%]	86.19	87.20	87.32	87.31
Δp	[Pa]	280.07	300.33	309.52	308.48
	[%]	0.093	0.100	0.103	0.103
$T_{a,max}$	[°C]	1110.04	1066.49	1089.68	1094.76
$\sigma_{a,min}^{III}$	[MPa]	-5.926	-4.997	-5.380	-5.476
$T_{g,max}$	[°C]	701.51	663.13	671.81	674.35
$\sigma_{g,max}^I$	[MPa]	23.633	10.459	10.362	10.417

Table 6: Performance indicators of selected cases

7.2.3. Relative scaling quality

In a next step, the primary scaling indicators of the selected cases ‘A’, ‘B’, and ‘C’ (presented in Table 5 and Figure 22) are compared with the minimum third principal stress of the reference receiver as the main material performance indicator to assess the relative scaling quality. Similarly, the secondary scaling indicators (also presented in Table 5 and Figure 22) are compared with the maximum absorber temperature of the reference receiver. This comparison is summarized in Table 7.

		A	B	C
$\Delta\bar{\sigma}_{III}$	[%]	6.69	6.62	6.70
$\Delta\sigma_{III,max}$	[%]	35.39	37.13	39.77
$\Delta\hat{\sigma}_{III,pt}$	[%]	15.67	9.21	7.59
$\Delta\bar{T}$	[%]	0.88	0.97	1.00
ΔT_{max}	[%]	3.41	3.27	3.46
$\Delta\hat{T}_{pt}$	[%]	3.14	1.47	1.11

Table 7: Relative scaling indicators of selected cases

It can be seen that the mean scaling error committed is in the region of 6.6%. The maximum stress surface errors of the selected cases are between 35.4 and 39.8% of the absolute value of the minimum third principal stress of the reference receiver. Finally, the stress point error was already normalized before and the difference of the selected cases to the reference receiver was shown to be between 7.6 and 15.7%. Similar behavior can be observed in Table 7 for the normalized secondary scaling indicators (related to the absorber material temperature).

7.3. Final receiver selection

When selecting the most optimal receiver configuration a trade-off has to be made between minimizing the stress point error $\Delta\hat{\sigma}_{III,pt}$ which ensures similar mechanical peak loads and reducing the maximum stress surface error $\Delta\sigma_{III,max}$ which shows the largest deviation compared to the reference receiver. At the same time the mean stress error $\Delta\bar{\sigma}_{III}$ needs to be kept small enough in order to guarantee an acceptable overall scaling quality. For the optimization performed case ‘B’ presents a suitable trade-off between the maximum stress point error $\Delta\hat{\sigma}_{III,pt}$ and the maximum surface error $\Delta\sigma_{III,max}$ with an excellent overall scaling quality and was therefore selected for further investigation and experimental evaluation.

8. Conclusions

A systematic scaling method for solar receivers was presented to guarantee that experiments performed in the controlled environment of HFSSs yield representative results when compared to full-scale tests. In particular, the solar receiver scaling effects of a solar receiver integrated into the full-scale OMSoP solar dish concentrator system and the KTH high-flux solar simulator were investigated.

For the receiver evaluation Monte Carlo ray-tracing routines of the solar dish concentrator and the solar simulator were developed and validated with experimental characterization results. The thermo-mechanical analysis of the solar receiver is based around a coupled CFD/FEM-analysis linked with stochastic heat source calculations in combination with the ray-tracing routines.

For the scaling evaluation a set of performance indicators was defined. The receiver outlet temperature T_{out} , the receiver efficiency η_{rec} , and pressure drop Δp across the receiver were selected as the main thermal performance indicators whereas the maximum temperature and stress of both the glass window and absorber were chosen as the most critical material performance indicators. Additionally, a reference receiver was designed to fit the specific boundary conditions present in the OMSoP solar dish system. The design is based on a concept developed in previous work for different power cycle boundary conditions and radiative boundary conditions derived from the Eurodish concentrator.

In a next step, the scaling parameters of the solar receiver integrated in the KTH HFSS were investigated and 3 scaling parameters were selected: namely the axial offset Δz of the receiver from the HFSS focal plane, the receiver aperture D_{ap} , and the distance L_n restricting the air inlet to the cavity between the glass window and the absorber were selected. It was shown that the axial offset and the receiver aperture diameter are the main parameters influencing the radiative boundary conditions of the receiver whereas the restriction strongly affects the fluid flow field and the heat transfer inside the porous absorber and thus the material temperatures and stresses.

A situation analysis performed allowed to identify the third principal stress profile inside the absorber as the main scaling criterion. The scaling analysis was therefore performed with the

main objective of minimizing the difference between the absorber stress profile of the scaled solar receiver in the HFSS and the solar receiver mounted in the solar dish system as well as minimizing the difference between the maximum stress of the receiver in both systems.

The scaling quality was evaluated using a set of main and secondary scaling indicators. As the main scaling indicators were defined the mean absorber stress surface error $\Delta\bar{\sigma}$, the maximum absorber stress surface error $\Delta\sigma_{max}$, and the stress point error $\Delta\hat{\sigma}_{pt}$.

To identify the most suitable receiver configuration a multi-objective optimization was performed utilizing distributing computing of a Linux-based network. The optimization allowed the selection of three potentially interesting receiver designs lying on the Pareto-optimal frontier. It was shown that the thermal performance (in terms of air outlet temperature and receiver efficiency) remains almost constant between the reference receiver and the cases 'A', 'B' and 'C' whereas the pressure drop across the receiver differs more. All cases presented showed material temperatures and stresses below the permissible limits.

When the primary scaling indicators of the scaled versions were compared with the minimum third principal stress of the reference receiver it was shown that the mean and point errors committed were in the region of 6.6% and between 7.6 and 15.7%, respectively, whereas the maximum stress surface errors were between 35.4 and 39.8%. Similar behavior was observed for the secondary scaling indicators.

In a final step, case 'B' was selected for further investigation and experimental evaluation as it presented a suitable trade-off between minimizing the stress point error to guarantee similar mechanical peak loads and minimizing the maximum stress surface error which shows the largest deviations to the reference receiver while an excellent overall scaling quality was achieved.

The selection of a solar receiver that yields representative results when tested in the KTH solar simulator compared to the full-scale OMSoP solar dish system is a crucial step in the development process of small-scale hybrid solar micro gas-turbines systems for off-grid generation.

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Nomenclature

Abbreviations

CSP	Concentrating Solar Power
DNIR	Direct Normal Irradiance
FP	Focal Point
HFSS	High Flux Solar Simulator
MGT	Micro Gas-Turbine
PDF	Probability Density Function

Symbols

A	Area	[m ²]
a	Absorption coefficient	[1/m]
D	Diameter	[m]
E	Irradiance (flux)	[W/m ²]
h	Specific enthalpy	[J/kg]
i	Light source spectral distribution	[W/m ²]
K_{ext}	Extinction coefficient	[1/m]
L	Length	[m]
$\dot{M}$	Mass flow	[kg/s]
n	Refractive index	[-]
P	Power	[W]
p	Pressure	[Pa]
$\dot{q}$	Surface heat source	[W/m ²]
$\dot{q}_v$	Volumetric heat source	[W/m ³]
r	Radial direction, radius	[m]
R	Reflection	[m]
s	Path length	[m]
u	Velocity	[m/s]
T	Temperature	[K]
z	Axial distance, axial direction	[m]

Greek Symbols

α	Absorptivity	[-]
δ	Flux error	[W/m ²]
ε	Emissivity	[-]
ε_p	Porosity	[-]
η	Efficiency	[-]
θ	Zenith angle	[rad]
Δ	Wavelength discretization	[m]
ρ	Reflectivity	[-]
ρ	Density	[kg/m ³]
σ	Stress	[MPa]
σ_s	Scattering coefficient	[1/m]
τ	Transmissivity	[-]
φ	Radiation absorption constant	[-]
φ	Azimuth angle	[rad]
ϕ	Cell diameter	[m]
χ	Random variable (0-1)	[-]
ψ	Exemplary variable	[-]

Subscripts

a	Absorber
ap	Aperture
b	Beam
c	Cavity
com	Comparison
ct	Center
el	Electric
fp	Focal plane
g	Glass
i	Incidence
in	Inlet
m	Measurement
max	Maximum
min	Minimum

<i>n</i>	Nozzle
λ	Spectral
<i>o</i>	Other
<i>out</i>	Outlet
<i>p</i>	Passing
<i>per</i>	Perpendicular
<i>par</i>	Parallel
<i>pt</i>	Point
<i>rad</i>	Radiation
<i>rec</i>	Receiver
<i>ref</i>	Reference
RT	ray-tracing
<i>t</i>	Transmitted
<i>tot</i>	Total
<i>v</i>	Volumetric
<i>w</i>	Wall
I, II, III	Principal direction indices

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