Optical and thermal modeling of parabolic trough concentrator systems

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OPTICAL AND THERMAL MODELING OF PARABOLIC TROUGHS CONCENTRATOR SYSTEMS

A dissertation submitted to
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for the degree of
Doctor of Sciences

presented by
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Abstract

To study the behavior of a solar parabolic trough concentrator (PTC) and to optimize its overall performance, a detailed model of the system is required. For this purpose, a 3D heat transfer model for PTC systems is developed in this thesis, and several studies in which the model is implemented are presented. Monte Carlo ray tracing, coupled to a finite volume solver, is used to model radiation, convection, and conduction heat transfer between all relevant surfaces of the receiver of the PTC. The non-uniform distribution of the incident solar radiation, the radiative exchange between the receiver surfaces, and the heat gain/loss around the receiver’s circumference and along the system’s axis are determined for spectral radiative properties of the receiver and concentrator surfaces. Comparing simulation results with experimental data from both on-sun and off-sun test setups indicates that the model accurately predicts heat transfer in PTC systems.

To estimate the expected energy yield of a solar field, the system performance is ideally determined at a single design point. To find such design points, thermal efficiencies of two PTC systems are evaluated with the detailed heat transfer model for a variety of operating conditions and geographical locations. By comparing results with yearly average efficiencies, it is shown that the most common choices of operating conditions at which solar field performance is evaluated are inadequate for predicting the yearly average efficiency, either significantly over- or under-predicting it by as much as 11.5%. An alternative simple method is presented of determining representative operating conditions for solar fields through weighted averages of the incident solar radiation. With this procedure, it is possible to accurately predict year-round performance of PTC systems within 0.3%, reduce the computational effort, and provide suitable operating conditions for the optimization process.
In the framework of a research project, a novel PTC receiver is developed that increases the system efficiency and reduces costs. A full optical and thermal analysis of improved selective coatings for this receiver is performed with the heat transfer model. The analysis shows that the best coating leads to higher efficiencies than existing coatings and confirms the potential of the novel coating material to be used for the new receiver concept. An off-sun test setup is proposed to heat the receiver to determine the temperature-dependent heat losses and to assess the performance of an innovative active vacuum system that is used to control pressure levels in the receiver. The expected behavior of the test setup is evaluated with the heat transfer model, and the predicted performance values are used to assess the specifications of the setup. Off-sun tests with a preliminary selective coating proved the feasibility of the testing process, and simulation results using the heat transfer model and spectral data agree well with experimental results.

To further increase the efficiency of PTC systems, the detailed heat transfer model is used to analyze the improvement potential of a typical PTC system with step-wise idealizations of system components. Sigmoid functions are used to idealize and optimize the optical behavior of the selective coating. Reflectance, absorptance, and transmittance behavior is evaluated for the glass envelope, assessing the potential performance of a system with an ideal selective glass. Optical properties of the primary concentrator mirror, such as reflectivity as well as tracking and surface errors, are successively idealized to reveal the differences between ideal and real concentrators. The effect of the supporting structure is also analyzed by reducing the number of heat collection elements (HCE) per mirror module, lowering the shaded area between HCEs, and removing the structure altogether. In a second step, several secondary mirror designs are evaluated and optimized, including a partially reflective glass surface, insulation in the vacuum annulus with reflective surfaces, as well as aplanatic mirror and tailored secondary designs. Idealizing components leads to increases in thermal efficiency between 4.3% and 7.3%, while the secondary designs enable efficiency increases of up to 1.6%.
Zusammenfassung


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Nomenclature

Latin characters

\( a \) Grid point coefficient, \([\text{W/mK}]\)
\( a_0 \) Coefficient, \([-\text{]}\)
\( A \) Area, \([-\text{]}\)
\( b_0 \) Coefficient, \([-\text{]}\)
\( c \) Coefficient, \([-\text{]}\)
\( c_0 \) Coefficient, \([-\text{]}\)
\( c_p \) Specific heat capacity, \([\text{J/kgK}]\)
\( \bar{c}_p \) Mean specific heat capacity, \([\text{J/kgK}]\)
\( C \) Effective concentration at the absorber tube, \([-\text{]}\)
\( C_1 \) Constant in Planck’s distribution, \([\text{W}/(\text{μm}^4/\text{m}^2\text{sr})]\)
\( C_2 \) Constant in Planck’s distribution, \([\text{μmK}]\)
\( C_s \) Secondary concentration, \([-\text{]}\)
\( d \) Distance, \([\text{m}]\)
\( d_0 \) Coefficient, \([-\text{]}\)
\( D \) Diameter, \([\text{m}]\)
\( e \) Angular error, \([\text{rad}]\)
\( e_{\lambda b} \) Directional spectral emissive power distribution, \([\text{W}/(\text{μm} \cdot \text{m}^2\text{sr})]\)
\( E \) Elevation above sea level, \([\text{km}]\)
\( f \) Friction coefficient, \([-\text{]}\)
\( F \) Focal length, \([\text{m}]\)
\( g \) Gravitational acceleration, \([\text{m/s}^2]\)
\( h \) Heat transfer coefficient, \([\text{W/m}^2\text{K}]\)
\( I \) Irradiance, \([\text{W/m}^2]\)
\( I_0 \) Solar constant, \([\text{W/m}^2]\)
\( I_{el} \) Electrical current, \([\text{A}]\)
$I_{\text{ex}}$  Extraterrestrial solar radiation, [W/m$^2$]
$I_l$  ASTM standard AM1.5 direct solar spectral irradiance, [W/m$^2\mu$m]
$k$  Thermal conductivity, [W/mK]
$K$  Scaled distance between secondary vertex and the focal line, [-]
$K_\theta$  Incidence angle modifier, [-]
$l$  Length, [m]
$l_{\text{seg1}}$  Length of an axial discretization element, [m]
$\dot{m}_1$  Mass flow rate, [kg/s]
$n$  Day of the year, [-]
$n_{\text{conc}}$  Number of concentrator modules in the solar field loop, [-]
$n_{\text{HCE}}$  Number of HCEs per concentrator module, [-]
$n_{\text{seg1}}$  Number of axial segments, [-]
$n_{\text{seg2}}$  Number of circumferential segments, [-]
$n_x$  Number of data points, [-]
$NA$  Scaled half-aperture of the primary mirror, [-]
$Nu$  Nusselt number, [-]
$P$  Perimeter, [m]
$P_{\text{pump}}$  Required power to pump the heat transfer fluid, [W]
$Pr$  Prandtl number, [-]
$\dot{q'}$  Power per unit length, [W/m]
$\dot{q''}$  Power flux per unit area, [W/m$^2$]
$\dot{Q}$  Total power, [W]
$Ra$  Rayleigh number, [-]
$Re$  Reynolds number, [-]
$s$  Scaled distance between primary and secondary vertices, [-]
$S$  Grid point source term, [W/m]
$T$  Temperature, [K]
$T_{\text{base}}$  Base temperature of the supporting structure, [K]
$T_{\text{sky}}$  Effective sky temperature, [K]
$u$  Measurement uncertainty associated with bias or precision error, [-]
Total measurement uncertainty, [-]

Velocity, [m/s]

Voltage, [V]

Volumetric flow rate, [l/min]

Aperture width, [m]

Vertical displacement from the focal line, [m]

**Greek Characters**

\( \alpha \)  
Absorptivity / absorptance, [-]

\( \alpha_{\text{diff}} \)  
Thermal diffusivity, [m²/s]

\( \beta \)  
Angular size of secondary objects, [°]

\( \beta_c \)  
Interaction coefficient, [-]

\( \beta_{\text{exp}} \)  
Volumetric thermal expansion coefficient, [K⁻¹]

\( \gamma \)  
Receiver circumferential angle, [°]

\( \delta \)  
Solar declination, [°]

\( \Delta T_{13} \)  
Temperature difference between HTF and absorber, [K]

\( \Delta T_{3,\text{max}} \)  
Maximum and minimum absorber temperature difference, [K]

\( \Delta x_i \)  
Measurement bias or precision error of variable \( x \), [-]

\( \Delta \eta_{\text{th}} \)  
Difference in thermal efficiency, [-]

\( \Delta \theta \)  
Subtended angle of control volume, [rad]

\( \varepsilon \)  
Emissivity / emittance, [-]

\( \varepsilon_{\text{tot}} \)  
Effective optical efficiency terms, [-]

\( \eta_{\text{col}} \)  
Collector efficiency, [-]

\( \eta_{\text{el}} \)  
Efficiency of power block, generator, and HTF pump, [-]

\( \eta_{\text{opt}} \)  
Optical efficiency of the solar field, [-]

\( \eta_{\text{opt,abs}} \)  
Absorber tube optical efficiency, [-]

\( \eta_{\text{opt,env}} \)  
Glass envelope optical efficiency, [-]

\( \eta_{\text{th}} \)  
Net thermal efficiency of the solar field, [-]

\( \theta \)  
Solar incidence angle at the concentrator aperture, [°]

\( \theta_s \)  
Acceptance angle, [°]
\( \theta_z \)  
Solar zenith angle, [°]

\( \lambda \)  
Wavelength, [\( \mu \text{m} \)]

\( \lambda_c \)  
Cut-off wavelength, [\( \mu \text{m} \)]

\( \lambda_{\text{mfp}} \)  
Mean free path, [cm]

\( \nu \)  
Kinematic viscosity, [m\(^2\)/s]

\( \rho \)  
Reflectivity / reflectance, [-]

\( \sigma \)  
Stefan-Boltzmann constant, [W/m\(^2\)K\(^4\)]

\( \sigma_x \)  
Standard deviation of data points, [-]

\( \tau \)  
Transmissivity / transmittance, [-]

\( \phi \)  
Latitude, [°]

\( \Phi \)  
Function associated with measurement uncertainty, [-]

\( \psi \)  
Rim angle, [°]

\( \omega \)  
Hour angle, [°]

**Subscripts**

1  
Heat transfer fluid

12  
HTF to inner absorber surface

2  
Inner absorber tube surface

23  
Absorber tube

3  
Outer absorber tube surface

34  
Outer absorber tube to inner glass surface

4  
Inner glass envelope surface

45  
Glass envelope

5  
Outer glass envelope surface

56  
Outer glass surface to atmosphere

6  
Atmosphere

abs  
Single absorber tube segment (HCE)

b  
Supporting structure bracket

bias  
Systematic measurement bias

C  
Constant source term
conc  Concentrator mirror
cond  Conductive heat transfer
conv  Convective heat transfer
D2   Inner absorber tube diameter
D5   Outer glass envelope diameter
DNI  Direct normal irradiation
gain  Heat gain of the heat transfer fluid
i    Circumferential segment
ins  Vacuum annulus insulation material
j    Axial segment
loss Total heat loss from the absorber tube
n    Receiver surface
prec Random precision error
rad  Radiative heat transfer
sec  Secondary (concentrator) mirror
sh   Shaded end parts of the HCE
solar Incident solar power
std  Standard conditions
system Solar field loop
T    Temperature dependent source term
track Tracking error
w    Yearly weighted average
y    Yearly average

**Abbreviations**

AR   Anti-reflective
CFD  Computational fluid dynamics
CPC  Compound parabolic concentrator
CPV  Concentrated photovoltaics
CRS  Central receiver system
<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>CSP</td>
<td>Concentrating solar power</td>
</tr>
<tr>
<td>CSR</td>
<td>Circumsolar ratio</td>
</tr>
<tr>
<td>DAQ</td>
<td>Data acquisition</td>
</tr>
<tr>
<td>DNI</td>
<td>Direct normal irradiation</td>
</tr>
<tr>
<td>DSG</td>
<td>Direct steam generation</td>
</tr>
<tr>
<td>DTA</td>
<td>Daily time-averaged value</td>
</tr>
<tr>
<td>FL</td>
<td>Focal line</td>
</tr>
<tr>
<td>HCE</td>
<td>Heat collection element</td>
</tr>
<tr>
<td>HTF</td>
<td>Heat transfer fluid</td>
</tr>
<tr>
<td>IR</td>
<td>Infrared</td>
</tr>
<tr>
<td>MC</td>
<td>Monte Carlo</td>
</tr>
<tr>
<td>NAC</td>
<td>New absorber coating</td>
</tr>
<tr>
<td>PSC</td>
<td>Preliminary selective coating</td>
</tr>
<tr>
<td>PTC</td>
<td>Parabolic trough concentrator</td>
</tr>
<tr>
<td>RAI</td>
<td>Reflective annulus insulation</td>
</tr>
<tr>
<td>RGS</td>
<td>Reflective glass surface</td>
</tr>
<tr>
<td>RMS</td>
<td>Root mean square</td>
</tr>
<tr>
<td>SMS</td>
<td>Simultaneous multiple surface</td>
</tr>
<tr>
<td>TMY</td>
<td>Typical meteorological year</td>
</tr>
<tr>
<td>YTA</td>
<td>Yearly time-averaged value</td>
</tr>
<tr>
<td>YWA</td>
<td>Yearly weighted average</td>
</tr>
</tbody>
</table>
1 Introduction

1.1 Solar Parabolic Trough Concentrator Technology

Parabolic trough concentrator (PTC) systems are currently the most mature and cost-effective technology to generate electricity through concentrating solar power (CSP) [1]. Unlike photovoltaic cells that convert solar radiation directly into electricity, CSP systems concentrate the incident solar radiation to generate heat. The thermal energy is then converted to electricity in a heat engine, distributed as process heat, or used in chemical applications. In a PTC, an array of parabolic-shaped mirrors concentrates the incident solar radiation onto the focal line where a tubular receiver is placed. The receiver generally consists of an absorber tube and a protective glass envelope surrounding the absorber. The absorbed solar radiation is transferred to a heat transfer fluid (HTF) that flows through the absorber tube and provides the thermal energy for the power cycle. The HTF is also used to provide heat to the thermal storage system, either directly using the fluid, or indirectly by transferring heat to a storage medium. The storage of the thermal energy produced by the solar field to overcome intermittence of the solar radiation (e.g. at night or during overcast weather) is an

Figure 1.1: Parabolic trough concentrator (PTC); adapted from [1].
important aspect in lowering the cost of electricity production in PTC systems [2]. In Figure 1.1, a PTC is shown, including the primary concentrator mirror, the tubular receiver, and the supporting structure. Figure 1.2 shows a 4 m section of a current-generation PTC receiver, also known as a single heat collection unit (HCE).

The primary concentrator mirror usually consists of parabolic-shaped glass facets that are coated with a highly reflective metallic material. As a result, these mirrors display excellent optical behavior even after many years of operation. On the other hand, the glass is heavy, expensive, requires a rigid supporting structure, and is prone to breakage. Several alternatives have been proposed, including the use of thinner glass or metallic panels that are coated with a polymer reflector [1, 3]. The absorber tube, which absorbs the concentrated solar radiation and transfers the energy to the HTF, is generally made of stainless steel and coated with a selective coating. The coating has a high optical absorptance in the visible light part and a very low thermal emittance in the infrared (IR) part of the spectrum. As a result, a high fraction of the incident solar radiation is absorbed, while significantly reducing losses through emitted thermal radiation. More details on the composition and properties of these selective coatings are given in Section 5.2. To protect the selective coating from degradation and to reduce heat losses, a glass envelope is placed concentrically.

![Figure 1.2: Current-generation PTC receiver.](http://www.phovo.de/phovoblog/wp-content/uploads/2012/07/SCHOTT-Solar-gewinnt-CSP-Today-Award_hier_SCHOTT-PTR-70-Receiver_Foto_SCHOTT-Solar.jpg, Accessed 01/14.)
around the absorber tube and a vacuum is established during the production of the HCE in the annular space between the glass and the absorber, with pressures of around 0.01 Pa [1]. At these pressure levels, molecular conduction across the annulus is insignificant and radiation heat transfer only can be considered to occur between the hot absorber and the cooler glass. The envelope is usually made from borosilicate glass and coated with an anti-reflective (AR) coating on both sides to increase the transmittance and reduce reflection losses.

Synthetic oils are used as HTF in the majority of PTC systems in operation today [3]. The hot HTF that leaves the solar field is used to produce steam in heat exchangers, which in turn is fed to a conventional steam turbine to generate electricity. Commonly used synthetic oils consist of mixtures of biphenyl and diphenyl oxides, which are stable up to temperatures of around 400°C, depending on the mixture [1]. There are many drawbacks to synthetic oils, most notably high cost, flammability, toxicity, and issues associated with their use for thermal storage [1]. Nevertheless, these materials are widely used due to their favorable properties, such as thermal stability, low viscosity, and liquid state from room temperature up to around 400°C [1, 4, 5]. The first commercial oil-based PTC systems combined with traditional power plants were built more than 30 years ago [1]. After a period of low activity, advances in technology and new political incentives caused a resurgence of PTC systems, most notably in the United States and Spain. In Spain, the implemented feed-in tariffs have resulted in numerous PTC projects since the beginning of the new millennium. Currently, more than 45 PTC power plants with a total electric power capacity of more than 2.2 GW are either in operation or being constructed [3, 6]. In the United States, PTC technology accounts for more than 1.3 GW of electric power capacity currently under construction or in operation [3, 6]. PTC systems are also being built or already in operation all over the world at a total electric power capacity of around 800 MW (excluding Spain and the United States), with several new PTC power plants being developed in India and on the African continent [3, 6].
Using synthetic oils as HTF has long been the industry standard. However, to address the above-mentioned drawbacks of these materials, the use of molten salt as HTF is investigated, with pilot plants already in operation [7]. Molten salts used in PTC systems are generally mixtures of NaNO₃, NaNO₂, KNO₃, and CaNO₃. Molten salts have higher densities than synthetic oils and are stable up to temperatures of about 600°C, depending on the mixture [2]. While the higher maximum temperatures allow for higher Carnot efficiencies in the power cycle, these salt mixtures also have high melting points. This constitutes a major drawback of these materials, because the PTC system must avoid freezing of the molten salt [2]. The main advantage of molten salt is that it can be simultaneously used as the HTF and as the thermal storage medium, thus avoiding the need for heat exchangers. Apart from molten salts, other materials and concepts have been proposed to store a portion of the generated heat, including synthetic oils, concrete, rocks, water/steam, or ceramic materials [1, 8, 9].

Alternative materials besides synthetic oils and molten salts are considered as well for use as HTF. Water may be used as HTF inside the absorber tube to produce saturated or superheated steam. In this process, known as direct steam generation (DSG), the steam that leaves the solar field is directly fed to steam turbines in the power block. The advantages of DSG are mainly the increases in the overall efficiency of the power plant through the avoidance of heat exchange processes and the higher temperatures that can be reached. On the other hand, there are challenges with stratification and the two-phase flow in the evaporation zone [1, 3]. In recent years, several pilot plants have been designed and/or built where the feasibility of DSG has been demonstrated [10, 11].

To a lesser extent, air has been used as HTF in PTC systems [12]. Using air has many advantages, as it is free, poses no environmental risks, allows for low system pressures, and temperature constraints due to chemical instability or phase change are avoided. However, air is associated with major challenges, most notably its low heat capacity and heat transfer coefficient, which lead to substantial pressure losses [12].
The main application of PTC systems is to generate electricity in a power plant by combining a solar field with a conventional heat engine. In most cases, the generated heat is used to produce steam, which is then used to run a steam turbine, as described above. However, the steam can also be used for other applications, such as providing industrial or domestic process heat, feed salt water desalination plants, or supply heat for detoxification and purification [3]. Even though PTC systems are a mature industrial technology, research and development is ongoing to improve existing applications, components, and materials, as well as to develop new technologies related to PTCs. More recently, a strong focus is put on the development of systems implementing DSG [10, 11] and molten salt technologies [7].

1.2 Modeling PTC Systems

Building PTC power plants is capital intensive due to the high material cost, the large land use, and the complex installation. For these reasons, simulations and modeling of PTC plants is needed to evaluate the behavior of the system and to optimize components before construction begins. Models can be used for various tasks, such as assessing the mechanical loads on the supporting structure of the PTC, simulating wind flow around the mirrors and over the solar field, describing the behavior of the power block that is used to convert thermal energy to electricity, or determining the amount of soiling on the mirrors and the receiver. The vast majority of the models, however, deal only with the transfer of energy from concentrated solar power to the useful thermal energy stored in the HTF. In the literature, a large number of these heat transfer models have been presented with varying degrees of detail and modeling approaches.

Heat transfer models of PTC systems usually assume uniform temperature distribution around the receiver’s circumference, despite the non-uniform distribution of solar radiation incident on the receiver [13-24]. For example, a 1D heat transfer analysis is performed at a receiver cross section and stepwise integrated along the receiver axis to determine the overall heat transfer of the PTC.
field [17]. Some recent modeling studies take into account non-uniform distributions [25-29], by applying the Monte Carlo (MC) ray tracing method to determine the incident solar radiation distribution. The radiative exchange between the various receiver surfaces and the atmosphere in all mentioned models is determined analytically, assuming gray or semi-gray, diffuse and opaque surfaces. A recent model implements more accurate heat transfer correlations [21] and models the radiative exchange in more detail by dividing the receiver surfaces into different zones and using configuration factors. Nevertheless, the glass envelope is still modeled as an opaque gray-diffuse surface hence the transmission of radiation at shorter wavelengths is not taken into account. In general, none of the reported models use spectral optics in the computation of the radiative exchange.

1.3 Thesis Goals

A major drawback of most of the reported heat transfer models is that they use empirical data for the optical evaluation. Therefore, these models are only valid for existing PTC designs or small deviations from these designs. In addition, the simplified modeling of the radiation terms does not allow for a detailed analysis of the heat transfer, especially for systems operating at higher temperatures. As a result, a combined optical and thermal analysis of new PTC systems with a high degree of detail is generally not possible with existing heat transfer models. For these reasons, the following goals are pursued in this thesis:

- Develop a detailed 3D heat transfer model to accurately describe and predict the behavior of existing and newly developed PTC systems.
- Validate the numerical results extracted from the model using existing experimental data to ensure accurate prediction of PTC system behavior.
- Analyze the behavior of a novel PTC receiver in both an off-sun test setup as well as in a generic PTC solar field.
- Optimize various system components and implement secondary optics, to quantify the overall improvement potential of PTC systems.
2 Heat Transfer Model

2.1 Introduction

In this chapter, a new 3D steady-state optical and thermal model for PTC systems is presented. The model is implemented in a Fortran program and consists of two parts. First, the non-uniform distribution of the incident concentrated solar radiation is computed using an in-house MC ray tracing code [30]. Then, the temperature distribution of all relevant receiver surfaces and the various modes of heat transfer are determined by applying the finite volume (FV) method for conduction and convection heat transfer and MC ray tracing for thermally emitted radiation iteratively. Figure 2.1 shows a cross section of a typical receiver consisting of an absorber tube and a glass envelope. Six locations are identified for the heat transfer analysis: The HTF flow inside the absorber (1), the inner (2) and outer (3) absorber tube surfaces, the inner (4)
and outer (5) glass envelope surfaces, and the surrounding atmosphere (6). The HTF and atmosphere temperatures $T_1$ and $T_6$ are assumed to be constant around the circumference of the receiver, while the surface temperatures $T_2$, $T_3$, $T_4$, and $T_5$ are non-uniform. Figure 2.1 also shows the modes of heat transfer considered in the heat transfer analysis.

### 2.2 Discretization Method

To account for the non-uniform radiation and temperature distribution around the circumference of the receiver surfaces as well as the temperature changes along the receiver’s axis, the heat transfer is evaluated at a number of surface segments. The circumference of each receiver surface is divided into $n_{seg2} = 100$ uniform segments. The receiver is divided into $n_{seg1}$ axial segments, where $n_{seg1}$ depends on the length of the collector system under consideration. For a typical solar field loop around 12 axial segments are used. Axial conduction is neglected due to the small temperature gradients along the receiver. Thus, the 2D steady-state energy conservation for an axial segment $j$ yields,

\[
0 = \frac{1}{r} \frac{\partial}{\partial r} \left( r k \frac{\partial T}{\partial r} \right) + \frac{1}{r} \frac{\partial}{\partial \theta} \left( k \frac{\partial T}{\partial \theta} \right) + S \tag{2.1}
\]

Equation (2.1) is solved numerically in a control volume for each surface segment [31]. The source term $S$ is linearized for a circumferential segment $i$ on surface $n$ with temperature $T$ (see Figure 2.2) by combining a constant part for the radiative terms and a temperature-dependent part for the convective terms:

\[
S_{n,i,j} = S_{C,n,i,j} + S_{T,n,i,j} T_{n,i,j}.
\]

At a surface segment, Eq. (2.1) is then discretized,

\[
a_{n,i,j} T_{n,i,j} = a_{n,i-1,j} T_{n,i-1,j} + a_{n,i+1,j} T_{n,i+1,j} + a_{n\pm1,i,j} T_{n\pm1,i,j} + S_{n,i,j} \tag{2.2}
\]

using grid point coefficients $a$. Each node of the grid represents the center of a surface segment and exchanges heat by conduction with three neighbor nodes, two neighbors on the same surface ($i-1$, $i+1$) and, depending on the position, a neighbor on the surface below or above the current one ($n\pm1$). Convection from a node to a fluid is modeled as a boundary condition, included in the terms $S_{n,i,j}$ and $a_{n,i,j}$, which are described in more detail in Appendix A.
2.3 Incident Solar Radiation

For a receiver exposed to concentrated solar radiation, the non-uniform incident radiation absorbed at the absorber tube and in the glass envelope is determined by MC ray tracing. For this purpose, a geometric model of a PTC system is generated. Structural elements such as support rods and shielding of receiver joints are included in the geometric model to take into account shading effects. Figure 2.3 illustrates a typical configuration consisting of three concentrator modules, a receiver, and structure elements, which are also shown in more detail in Figure 6.10. The results of the ray tracing of solar radiation for this geometry are used to describe the radiation distribution of a system with any number of concentrator modules, with the data for the center part being used for all but the first and last module in the system.

The parabolic surface is modeled as a specular reflector with an angular reflection error $e_{\text{conc}}$ [32]. Two circular cylinders with Fresnel reflection/refraction behavior are used to model the outer and inner surface of the glass envelope, and an absorbing medium between the two cylinders models absorption in the glass. The absorber tube is modeled as a cylinder with diffuse reflection and emission, but directional surface properties can also be incorporated. For all surfaces, constant values or spectrally dependent optical
Figure 2.3: Visualization of a typical ray tracing setup with a detailed view of the receiver surfaces.
parameters such as reflectivity, emissivity or transmissivity can be used. The incident rays are directed at the parabolic concentrating mirrors at a specified incidence angle using a CSR 5% sunshape angular distribution [33] and a spectral distribution equal to the ASTM standard AM1.5 direct solar spectrum. To simulate tracking errors, the rays are displaced around the concentrator’s axis (y-axis in Figure 2.3) by $e_{\text{track}} = 1$ mrad. Depending on the desired accuracy, a total number of incident rays on the order of $10^7$ to $10^8$ are directed in the MC ray tracing at the collector geometry model, and the resulting absorbed radiation distributions at the absorber tube ($q'_{\text{solar,3},ij}$) and in the glass ($q'_{\text{solar,45},ij}$) are used in the subsequent heat transfer analysis as heat sources.

To highlight the benefits of using MC ray tracing to analyze the effects of the incidence angle, numerical results from simulations are presented here and compared to the empirically derived incidence angle modifier (IAM) [13]:

$$K_\theta = \cos \theta + 0.000884 \theta - 0.00005369 \theta^2$$

(2.3)

The IAM describes the decrease in optical efficiency with increasing angles of incidence and is defined as the ratio between the optical efficiency at a given angle and the efficiency at $\theta = 0^\circ$. Thus, the IAM includes cosine losses at the concentrator aperture as well as other directionally-dependent effects such as glass transmittance and absorber absorptivity [13]. The optical efficiency at the absorber is defined here as the radiation absorbed at the absorber tube $q'_{\text{solar,3}}$ divided by the direct normal irradiation (DNI) $q'_{\text{DNI}} = I_{\text{DNI}} \cdot w_{\text{conc}}$. Forristall [17] uses the same IAM to determine the optical efficiency at the absorber,

$$\eta_{\text{opt,abs}}(\theta) = \frac{q'_{\text{solar,3}}(\theta)}{q'_{\text{DNI}}} = \varepsilon'_{\text{tot}} \cdot K_\theta(\theta) \cdot \tau_{45} \cdot \alpha_3$$

(2.4)

where $\varepsilon'_{\text{tot}}$ is the product of several effective optical efficiency terms accounting for various losses such as shadowing, tracking errors, mirror geometry error, and mirror reflectivity [17]. Similarly, an optical efficiency for the glass envelope is defined,

$$\eta_{\text{opt,env}}(\theta) = \frac{q'_{\text{solar,45}}(\theta)}{q'_{\text{DNI}}} = \varepsilon'_{\text{tot}} \cdot K_\theta(\theta) \cdot \alpha_{45}$$

(2.5)
Table 2.1: Design parameters of the Forristall model [17].

<table>
<thead>
<tr>
<th>Design Parameter</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length of a solar field loop</td>
<td>$l_{\text{system}}$</td>
<td>779.52 m</td>
</tr>
<tr>
<td>Absorber tube; length</td>
<td>$l_{\text{abs}}$</td>
<td>4.06 m</td>
</tr>
<tr>
<td>Absorber tube; inner diameter</td>
<td>$D_2$</td>
<td>0.066 m</td>
</tr>
<tr>
<td>Absorber tube; outer diameter</td>
<td>$D_3$</td>
<td>0.07 m</td>
</tr>
<tr>
<td>Absorber tube; solar absorptivity</td>
<td>$\alpha_3$</td>
<td>0.955</td>
</tr>
<tr>
<td>Glass envelope; inner diameter</td>
<td>$D_4$</td>
<td>0.109 m</td>
</tr>
<tr>
<td>Glass envelope; outer diameter</td>
<td>$D_5$</td>
<td>0.115 m</td>
</tr>
<tr>
<td>Glass envelope; solar transmittance</td>
<td>$\tau_{45}$</td>
<td>0.965</td>
</tr>
<tr>
<td>Glass envelope; solar absorptance</td>
<td>$\alpha_{45}$</td>
<td>0.02</td>
</tr>
<tr>
<td>Concentrator; number of modules</td>
<td>$n_{\text{conc}}$</td>
<td>64</td>
</tr>
<tr>
<td>Concentrator; length</td>
<td>$l_{\text{conc}}$</td>
<td>12 m</td>
</tr>
<tr>
<td>Concentrator; aperture width</td>
<td>$w_{\text{conc}}$</td>
<td>4.8235 m</td>
</tr>
<tr>
<td>Concentrator; rim angle</td>
<td>$\psi_{\text{conc}}$</td>
<td>70°</td>
</tr>
<tr>
<td>Concentrator; reflectivity</td>
<td>$\rho_{\text{conc}}$</td>
<td>0.85</td>
</tr>
<tr>
<td>Concentrator; angular reflection error</td>
<td>$e_{\text{conc}}$</td>
<td>2 mrad</td>
</tr>
<tr>
<td>Concentrator; tracking error</td>
<td>$e_{\text{track}}$</td>
<td>1 mrad</td>
</tr>
</tbody>
</table>

In this section, MC ray tracing is performed to investigate the optical efficiencies of both the absorber tube and the glass envelope for incidence angles ranging from $0^\circ$ to $60^\circ$. The design parameters for a parabolic trough field with a total length of 779.52 m are taken from [17] and are summarized in Table 2.1. For the optical properties of the glass and the absorber’s selective coating, the same constant values as in [17] are used. The values for $\rho_{45}$, $\tau_{45}$, and $\epsilon_{45}$ are used to calculate values of refractive index and absorption coefficient that are then provided to the MC ray tracing program. To account for the decreasing absorptivity of the absorber’s selective coating at increasing incidence angles, a directional dependence for the absorber emissivity is applied [34]:

$$\epsilon_3(\theta) = \left[1 - a\left(\frac{1}{\cos \theta} - 1\right)^6\right] \cdot \epsilon_3(\theta = 0^\circ)$$  \hspace{1cm} (2.6)
with $a = 0.5$ and $b = 2$. Due to the lack of optical data for the concentrator mirrors, the mirror reflectivity is chosen such that the same optical efficiency at the absorber is obtained at normal incidence as Forristall [17]. All optical parameters that are used are also listed in Table 2.1. Figure 2.4 shows the absorbed radiative power at the absorber tube and in the glass envelope relative to the solar radiation incident on the concentrator aperture,

$$\dot{q}'_{\text{solar,conc}} = \dot{q}'_{\text{DNI}} \cos \theta$$

(2.7)

By using the same angle modifier $K_{\theta}$ and constant efficiency $\epsilon'_{\text{tot}}$, the Forristall model predicts the same decrease in absorption efficiencies for the glass and absorber at increased incidence angles. In contrast, MC ray tracing predicts that the fraction of incident radiative power absorbed in the glass increases for incidence angles $\theta > 10^\circ$. At $\theta = 60^\circ$, the two predictions differ by a factor of about 2. This is because higher incidence angles increase the path length inside the glass, resulting in lower transmittance. Additionally, the higher absorber reflectivity increases the amount of radiation passing through the glass.

![Figure 2.4: Power absorbed at the absorber and in the glass divided by the solar radiation incident on the concentrator aperture.](image-url)
2.4 Heat Gain

The heat gain of the receiver is simply the convective heat transfer from the inner absorber surface (2) to the HTF (1):

$$\dot{q}_{\text{gain},i,j} = \dot{q}_{\text{conv},12,i,j} = h_{12,i,j} \left( T_{1,i,j} - T_{2,i,j} \right) \cdot \frac{\pi D_2}{n_{\text{seg}2}}$$  \hspace{1cm} (2.8)

The heat transfer of an axial segment $j$ is determined by summing up the values for all circumferential surface segments $i$, and the overall heat transfer of the system is derived by calculating the mean of the values of all axial elements $j$:

$$\dot{q}'_{\text{gain}} = \frac{1}{n_{\text{seg}1}} \sum_{j} \dot{q}'_{\text{gain},j} = \frac{1}{n_{\text{seg}1}} \sum_{j} \sum_{i} \dot{q}'_{\text{gain},i,j}$$  \hspace{1cm} (2.9)

The heat transfer coefficient $h_{12} = \frac{\text{Nu}_{D2} k_1}{D_2}$ for the convective heat transfer to the HTF is determined using the Nusselt number correlation of Gnielinski [35]

$$\text{Nu}_{D2} = \frac{f_2}{8 \left( \text{Re}_{D2} - 1000 \right) \text{Pr}_1} \left( \frac{\text{Pr}_1}{\text{Pr}_2} \right)^{0.11}$$  \hspace{1cm} (2.10)

with

$$f_2 = \left( 1.82 \log_{10} \left( \text{Re}_{D2} \right) - 1.64 \right)^2$$  \hspace{1cm} (2.11)

For the thermal properties of the HTF, tabulated data from the manufacturers are used. The thermal conductivity of the absorber tube is given by $k_{25}(T) = 0.0153 \cdot T + 14.775$ W/mK [17].

2.5 Heat Loss from the Absorber Tube

The heat loss from the absorber tube is defined as the sum of radiative and convective heat transfer to the glass, and the conduction losses through the supporting structure:

$$\dot{q}'_{\text{loss},i,j} = \dot{q}'_{\text{conv},34,i,j} + \dot{q}'_{\text{rad},34,i,j} + \dot{q}'_{\text{rad},36,i,j} + \dot{q}'_{\text{cond},\text{loss},i,j}$$  \hspace{1cm} (2.12)

The heat loss is primarily dependent on the net radiative transfer to the glass envelope that is either absorbed by the glass ($\dot{q}'_{\text{rad},34}$) or transmitted through the glass ($\dot{q}'_{\text{rad},36}$). Together with the radiation emitted by the inner glass surface that
Figure 2.5: Visualization of the MC ray tracing for a single segment of the absorber tube (a) and inner glass surface (b) at the top of the receiver. The dashed lines represent radiation transmitted through the glass envelope.
is transmitted to the atmosphere \( (q_{\text{rad,46}}') \), these terms are determined using MC ray tracing. A total of around \( 10^7 \) rays are distributed to all surface segments based on the magnitude of their emitted radiative power. After the entire ray tracing procedure, the net radiative exchange for each segment is known and is included in the overall heat transfer analysis in the source term. Figure 2.5 shows a selected number of rays emitted by a single segment of the absorber tube (a) and the glass envelope (b). Most of the rays emitted in the IR spectrum by the absorber tube are absorbed by the glass, either immediately or after a number of reflections. At higher temperatures, radiation at shorter wavelengths can be transmitted through the glass, represented by the dashed lines.

The conductive heat transfer through the vacuum annulus between the nodes \( T_{3,i,j} \) and \( T_{4,i,j} \), and the conductive losses to the supporting structure are determined using heat transfer correlations presented in [17]. For most cases, high vacuum is assumed in the annulus between the absorber tube and the glass envelope, with a pressure of 0.0001 torr. A heat transfer correlation for a convective flow [36] is adopted,

\[
th_{34} = \frac{k_{\text{std}}}{\left( D_3/2 \ln \left( \frac{D_4}{D_3} \right) + \beta_c \lambda_{\text{mfp}} \left( \frac{D_3}{D_4} + 1 \right) \right)}
\]

where \( k_{\text{std}} \) is the thermal conductivity of the annulus gas (e.g. air) at standard conditions, \( \beta_c = 1.571 \) is the interaction coefficient, and \( \lambda_{\text{mfp}} \) is the mean free path [17]. The conduction losses through the supporting structure of a single HCE are modeled as an infinite fin identical to [17]:

\[
\dot{q}_{\text{cond,loss},i,j}' = \sqrt{h_b P_b k_b A_b \left( T_{\text{base},i,j} - T_6 \right) / l_{\text{abs}}}
\]

The base temperature \( T_{\text{base},i,j} \) is assumed to be 10 degrees Kelvin lower than the outer absorber tube temperature \( T_{3,i,j} \). The bracket temperature, which is estimated as \( (T_{\text{base},i,j} + T_6)/3 \), is used to determine the effective heat transfer coefficient \( h_b \) using the correlations described in Section 2.6. For the supporting structure displayed in Figure 2.3 and Figure 6.10 values for the bracket perimeter \( P_b = 0.2032 \) m, the thermal conductivity \( k_b = 48 \) W/mK, and the bracket cross-sectional area \( A_b = 1.613 \cdot 10^{-4} \) m\(^2\) are used.
2.6 Heat Loss from the Glass Envelope

The heat transferred from the absorber tube to the inner glass surface across the annulus is carried away by radiative emission that is transmitted through the glass and by conduction to the outer surface, using $k_{45} = 1.04$ W/mK [17]. At the outer glass envelope surface, the conductive term together with the absorbed solar radiation is released to the atmosphere by means of combined convection and radiation. Two Nusselt number correlations are used for the convective heat transfer coefficient $h_{56} = \text{Nu}_{D5}k_5/D_5$, depending on whether forced convection by wind or natural convection without wind occurs. For the natural convection case [37]:

$$\text{Nu}_{D5} = \left(0.6 + \frac{0.387 \text{Ra}_{D5}^{1/6}}{1 + (0.559 / \text{Pr}_{56})^{9/16}}\right)^{2/3}(2.15)$$

using the Rayleigh number

$$\text{Ra}_{D5} = \frac{g \beta_{\text{exp},56} \alpha_{\text{diff},56} \nu_{56}}{a_{56}} (T_5 - T_6) D_5^3(2.16)$$

The Prandtl number $\text{Pr}_{56}$, the thermal expansion coefficient for an ideal gas $\beta_{\text{exp},56} = 1/T_{56}$, the thermal diffusivity $\alpha_{\text{diff},56}$, and the kinematic viscosity $\nu_{56}$ are evaluated at the film temperature $T_{56} = (T_5 + T_6)/2$, see Appendix C. For the forced convection case [38]:

$$\text{Nu}_{D5} = C \cdot \text{Re}_{D5}^{n} \cdot \text{Pr}_{6}^{a} \left(\frac{\text{Pr}_6}{\text{Pr}_5}\right)^{1/4}(2.17)$$

using $n = 0.37$ and, depending on the local Reynolds and Prandtl numbers, the parameters listed in Table 2.2. Since the radiation emitted from the outside surface of the glass envelope (5) does not interact with any other surfaces, it can be determined analytically using the segment’s temperature and total emittance, and the effective sky temperature, assumed to be $T_{\text{sky}} = T_6 - 8$ K [17]:

$$q_{\text{rad},56,i,j} = \frac{n}{\text{seg}^2} \varepsilon_{45,i,j} \sigma \left(T_{56,i,j}^4 - T_{\text{sky}}^4\right)(2.18)$$
Table 2.2: Parameters for the forced convection Nusselt number [38].

<table>
<thead>
<tr>
<th>$Re_{D5}$</th>
<th>$C$</th>
<th>$m$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 – 40</td>
<td>0.75</td>
<td>0.4</td>
</tr>
<tr>
<td>40 – 1000</td>
<td>0.51</td>
<td>0.5</td>
</tr>
<tr>
<td>1000 – 2 · 10^5</td>
<td>0.26</td>
<td>0.6</td>
</tr>
<tr>
<td>2 · 10^5 – 10^6</td>
<td>0.076</td>
<td>0.7</td>
</tr>
</tbody>
</table>

2.7 Heat Transfer Analysis

At each axial segment $j$, the 2D heat transfer is determined based on the constant HTF temperature $T_{1,j}$. The temperatures of all surfaces are determined by iteratively computing thermally emitted radiation from surfaces (3) and (4) by MC ray tracing and the conduction and convection heat transfer as described in Section 2.2. This process is repeated until an RMS temperature difference of 0.05 K or less is reached between iterations at convergence for all surface segments. This results in percentage efficiencies that are converged to one decimal figure. Afterwards, the computed heat gain $\dot{q}_{\text{gain},j}^{'}$ is used to calculate the new HTF temperature $T_{1,j+1}$ at the next segment along the receiver’s axis:

$$ T_{1,j+1} = T_{1,j} + \frac{\dot{q}_{\text{conv},12,j}^{'} \cdot l_{\text{seg1}}}{\dot{m}_1 \cdot c_{p,1,j}} $$

(2.19)

where $c_{p,1,j}$ is the mean specific heat capacity of the HTF, $\dot{m}_1$ is the HTF mass flow rate, and $l_{\text{seg1}}$ is the length of an axial segment. Changes in potential and kinetic energy are neglected. With the new HTF temperature and the incident radiation distribution of that segment, the entire 2D heat transfer analysis is repeated. After the analysis of all $n_{\text{seg1}}$ segments, 3D temperature and heat transfer distributions are available for all receiver surfaces.
3 Model Validation

3.1 Introduction

To validate a heat transfer model, ensuring that it accurately describes the behavior of PTC systems, simulation results can be compared to experimental data available in the literature. Although several experimental studies of PTC systems have been reported in the literature, extensive data on the optical and thermal performance are rather scarce. In general, the experimental studies are divided into on-sun and off-sun test setups. In on-sun tests, a section of a PTC solar field, including receiver and primary concentrator mirror, is used in real-life conditions to heat a HTF while measuring various temperatures and the DNI. The setup allows assessing both the optical and thermal performance of the system, as direct solar radiation and a HTF is present. Such on-sun tests can be conducted on special test platforms, which are only a few meters long and may be equipped with a 2D tracking device [13, 39, 40]. Alternatively, a modified loop of a PTC solar field is used as a test loop for the experimental testing of the system [10, 17, 41, 42]. Sometimes IR cameras are used to derive the thermal behavior of the system from the temperature readings of the glass envelope [43].

In contrast to the test platform, where usually only small volumetric flow rates of the HTF are possible, the test loop allows for much more realistic heat transfer conditions in the absorber tube. However, realistic operating conditions can be a drawback, for instance when a specific solar incidence angle is desired over a longer period of time that does not occur in real life. A test platform using the 2D tracking system, on the other hand, can be oriented such that a

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constant solar incidence angle (for example $\theta = 0^\circ$) is realized over the entire time of testing. These controlled conditions make data sets of test platforms very valuable and useful for the validation of heat transfer models. As a result, the very comprehensive on-sun test report by Dudley et al. [13] of a tracking test platform is used for validation by the majority of the heat transfer models in the literature. The report presents on-sun test results on heat loss and collector efficiency for two receiver types and various receiver configurations. The report also provides data on the HTF, the optical efficiency of the system, and the optical properties of the glass envelope and the selective coatings.

A simpler way of learning about the thermal behavior of a PTC system is to test only the receiver in a controlled laboratory atmosphere. In these so-called off-sun tests, no solar radiation, concentrator mirror, or HTF is present, but a single HCE is electrically heated. Although no information can be gained about the optical and thermal efficiency of the system with such a setup, valuable information about the performance of the selective coating can be extracted from the thermal heat losses of the absorber tube. The majority of these off-sun test setups presented in the literature use electrical resistance cartridge heaters and heating coils that are placed in a copper tube to even out the temperature distribution [44-48]. The copper tube is inserted into the absorber tube to heat the receiver to a desired temperature. The total thermal loss of the HCE equals the electric power required to maintain that temperature. As an alternative, the HCE itself can be used as the heat generator, by passing an electric current directly through the absorber tube. Although this method simplifies the setup and allows for a much more uniform heat generation in the HCE, this procedure does not appear to be used frequently [49]. For validation purposes, detailed reports such as those presented by Burkholder et al. [44, 45] on off-sun heat loss measurements of the Solel UVAC3 and Schott PTR70 receivers are generally used. Despite the lack of incident solar radiation and HTF, these reports offer interesting data, including measurement of glass temperatures.
In this chapter, experimental data from the on-sun test report by Dudley et al. [13] and the off-sun test report by Burkholder et al. [45] are used to validate the detailed heat transfer model described in Chapter 2. Measurement data of thermal efficiency, heat loss, and glass temperatures are compared to results from simulations of the test setups using the heat transfer model.

3.2 SEGS LS-2 Field Tests

From the various results presented in the on-sun test report by Dudley et al. [13], two data sets are chosen for comparison with the new heat transfer model. One set of data is the result of efficiency measurements with varying solar irradiation, while the other data set is from heat loss measurements performed without direct solar irradiation. Both experimental campaigns were performed under various operating conditions, such as wind speed, ambient temperature or volumetric flow rate of the HTF. A Luz LS-2 collector with a cermet-coated receiver under vacuum and Syltherm 800 as the HTF were used in these experiments [4]. In Table 3.1, design parameters of the test setup are listed.

There are several ways to model the optical behavior of the absorber tube and the glass envelope. In this section, full spectral optical data are used when available; otherwise a two-band approximation is implemented. For the spectral simulations, the reflectivity curve “Old Cermet ave” found in [50] is used to calculate the absorber tube’s spectral emissivity for all radiation sources. Due to the lack of detailed spectral data for the glass envelope, a two-band approach with a cut-off wavelength of \( \lambda_{c,45} = 2.6 \, \mu m \) is taken. The glass transmittance is set to 0.935 for \( \lambda \leq \lambda_{c,45} \) and 0.0 for \( \lambda > \lambda_{c,45} \), and the reflectance is set to 0.045 for \( \lambda \leq \lambda_{c,45} \) and 0.14 for \( \lambda > \lambda_{c,45} \) [17]. In contrast to using spectral data, Forristall [17] and most other modeling studies implement a semi-gray approach, in which two separate sets of gray optical data are used, one for the incident solar radiation and the other for the mostly IR exchange radiation within the receiver. For instance, the constant optical properties of the glass
Table 3.1: Design parameters of the SEGS LS-2 field tests by Dudley et al. [13, 17].

<table>
<thead>
<tr>
<th>Design Parameter</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Direct solar irradiation (on-sun)</td>
<td>$I_{DNI}$</td>
<td>880.6-982.3 W/m²</td>
</tr>
<tr>
<td>Incidence angle (on-sun)</td>
<td>$\theta$</td>
<td>0°</td>
</tr>
<tr>
<td>Wind speed</td>
<td>$v_6$</td>
<td>0.1-4.2 m/s</td>
</tr>
<tr>
<td>Ambient temperature</td>
<td>$T_6$</td>
<td>19.9-31.1°C</td>
</tr>
<tr>
<td>HTF volumetric flow</td>
<td>$\dot{V}_1$</td>
<td>27.4-56.8 l/min</td>
</tr>
<tr>
<td>Absorber tube; length</td>
<td>$l_{abs}$</td>
<td>4 m</td>
</tr>
<tr>
<td>Absorber tube; inner diameter</td>
<td>$D_2$</td>
<td>0.066 m</td>
</tr>
<tr>
<td>Absorber tube; outer diameter</td>
<td>$D_3$</td>
<td>0.070 m</td>
</tr>
<tr>
<td>Absorber tube; solar absorptivity</td>
<td>$\alpha_3$</td>
<td>0.92</td>
</tr>
<tr>
<td>Absorber tube; thermal emissivity</td>
<td>$\varepsilon_3$</td>
<td>$f(T_3)$</td>
</tr>
<tr>
<td>Glass envelope; inner diameter</td>
<td>$D_4$</td>
<td>0.109 m</td>
</tr>
<tr>
<td>Glass envelope; outer diameter</td>
<td>$D_5$</td>
<td>0.115 m</td>
</tr>
<tr>
<td>Glass envelope; transmittance (Incidence/Exchange radiation)</td>
<td>$\tau_{45}$</td>
<td>0.935 / 0.0</td>
</tr>
<tr>
<td>Glass envelope; reflectance (Incident/Exchange radiation)</td>
<td>$\rho_{45}$</td>
<td>0.045 / 0.14</td>
</tr>
<tr>
<td>Concentrator; length</td>
<td>$l_{conc}$</td>
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</tr>
<tr>
<td>Concentrator; aperture width</td>
<td>$w_{conc}$</td>
<td>5 m</td>
</tr>
<tr>
<td>Concentrator; rim angle</td>
<td>$\psi_{conc}$</td>
<td>70°</td>
</tr>
<tr>
<td>Concentrator; reflectivity (Gray/Spectral case)</td>
<td>$\rho_{conc}$</td>
<td>0.903 / 0.892</td>
</tr>
<tr>
<td>Concentrator; angular reflection error</td>
<td>$e_{conc}$</td>
<td>2 mrad</td>
</tr>
<tr>
<td>Concentrator; tracking error</td>
<td>$e_{track}$</td>
<td>1 mrad</td>
</tr>
</tbody>
</table>

mentioned above are separately used as gray values for the incidence and exchange radiation. In this section, this approach is adopted for comparison reasons by using these constant optical values taken from [17]. The absorptivity of the absorber tube for the incident radiation is 0.92. For the exchange radiation, a regression fit of experimental data as a function of the surface temperature (in Kelvin) is used to determine the thermal emissivity [13, 17],

$$\varepsilon_3 = 0.000327 \cdot T_3 - 0.065971$$  \hspace{1cm} (3.1)
The reflectivity of the concentrator mirror is set to 0.903 in the gray case and to 0.892 in the spectral case to match the experimentally measured optical efficiency of the testing system [13].

One of the benefits of using MC ray tracing and 3D heat transfer analysis is the detailed information that can be extracted on the radiation and temperature distribution of the various surfaces. Figure 3.1 shows the distribution of the absorbed solar radiative power by the absorber tube ($q_{solar,3}$) and the glass envelope ($q_{solar,45}$) as a function of the circumferential angle $\gamma$. The asymmetry in the two peaks of absorbed radiation is a result of the simulated tracking error. Besides the average concentration ratio at the absorber tube of 16.6, peaks in the local concentration ratio occur at two points and equal 43.5 and 45.4. Obviously, the radiation distribution depends strongly on the optical characteristics of the concentrator mirror and the optical properties of the glass and absorber coating. The non-uniform radiation distribution also causes a non-uniform temperature distribution. The shape of that distribution, however, is not

![Figure 3.1: Circumferential distribution of the incident solar radiation absorbed at the absorber and in the glass for $I_{DNI} = 933.7$ W/m².](image-url)
the same as the radiation distribution, as it depends strongly on the material thermal properties and the operating conditions. Figure 3.2 shows the temperature profiles for the outer absorber surface and the outer glass surface as a function of $\gamma$ for two on-sun tests performed by Dudley et al. [13]. The temperature distributions are generally smoother than the incident radiation distributions. The difference between absorber tube and HTF temperatures is significant. For example, for $T_1 = 102.2^\circ$C, the difference between mean outer absorber tube temperature and HTF temperature is $\Delta T_{13} = 57.8$ K. This is primarily due to the low heat transfer coefficient caused by the low HTF mass flow rates obtained in the testing facility. For this reason, there is also a significant temperature difference between the portion of the absorber surface exposed to the concentrated solar radiation and the cool side facing the sky. The difference between maximum and minimum absorber temperatures is $\Delta T_{3,max} = 128.5$ K for $T_1 = 102.2^\circ$C. With increasing HTF temperature, the convective heat transfer coefficient increases and the temperature difference

Figure 3.2: Temperature distribution around the circumference of the outer absorber and glass surfaces.
between the absorber tube and HTF as well as the difference between cool and hot absorber sides decreases. For $T_1 = 379.5^\circ C$ these differences are $\Delta T_{13} = 19.3 \text{ K}$ and $\Delta T_{3,\text{max}} = 51.8 \text{ K}$, respectively.

The results of the simulations of the irradiated field test setup by Dudley et al. [13] are displayed in Figure 3.3. The collector efficiency $\eta_{\text{col}}$, defined as the heat gain of the HTF divided by the solar radiation incident on the concentrator aperture, is plotted for various average differences between the HTF temperature and the ambient air temperature $T_1 - T_6$. The solid line represents the results assuming gray surfaces and using a temperature correlation for the absorber’s emissivity, whereas the dashed line displays the results using spectral properties. As expected, the gray results show a good agreement with the Forristall model [17], since all the optical and thermal properties were adopted from it. The agreement with the actual experimental value varies for different temperature points. Nevertheless, the model results are within the uncertainty limits of the experimental data for all temperatures. It appears that the correlation for the selective coating’s emissivity used by Forristall may

![Figure 3.3: On-sun field test: Collector efficiency.](image-url)
under-predict the radiative emission. The correlation used by Forristall [17] matches data from field tests performed by Dudley et al. [13], however, these consist of only two data points. In laboratory tests, Dudley et al. also determined higher emittance values, so it is possible that the actual emissivity of this receiver’s selective coating is higher than predicted by the correlation used by Forristall. When spectral properties are used for the absorber’s emissivity, the overall higher emitted radiation matches the experimental data better. The fact that a significant portion of the emitted radiation from the absorber tube is transmitted through the glass and not absorbed in it can also have an impact on the overall heat loss from the absorber. The under-prediction of the absorber tube emittance by correlations is also indicated by the measured and simulated heat loss shown in Figure 3.4. The heat loss is again plotted for various differences between HTF temperature and ambient temperature, with the solid and dashed lines representing the gray and spectral models, respectively. Similar to the mentioned efficiency measurements, the low absorber emissivity under-predicts the heat loss. However, the new heat transfer model still lies within the figure.

Figure 3.4: On-sun field test: Heat loss of the absorber tube.
uncertainty limits for all temperatures. The agreement, however, becomes worse at higher temperatures. Better optical and thermal property data are expected to improve the model predictions, as seen when spectral properties are used. The emissivity of the selective coating “Old Cermet ave” is overall higher than the temperature correlation. The fact that radiation is also allowed to be transmitted through the glass contributes to the higher overall heat losses.

3.3 Schott PTR70 Lab Tests

In this section, experimental measurements of heat loss for the Schott 2008 PTR70 receiver in the lab [45] are compared to the current model. As explained in the introduction to this chapter, in this setup there is no incident solar radiation and no HTF present. Instead, the inner absorber surface is electrically heated to maintain a constant temperature. The available data include the measured heat loss for various temperatures of the inner absorber tube surface and the temperature on the outer glass surface. Identical to the previous section, the analysis is performed using both spectral data and assuming gray surfaces. Spectral data for Schott’s “New Absorber Coating” (NAC) [51], shown in Figure 5.2, is used to determine the absorber’s emissivity. A two-band approach with a cut-off wavelength of $\lambda_{c,45} = 2.6 \, \mu m$ is used for the glass properties. The transmittance is set to 0.965 for $\lambda \leq \lambda_{c,45} \, \mu m$ and 0.0 for $\lambda > \lambda_{c,45} \, \mu m$, and the reflectance is set to 0.015 for $\lambda \leq \lambda_{c,45} \, \mu m$ and 0.11 for $\lambda > \lambda_{c,45} \, \mu m$ [17, 45]. Using the measured heat losses and temperatures, together with a simple heat transfer model, Burkholder et al. determined the thermal emittance of the absorber tube as a function of the average absorber temperature (in °Celsius) [45],

$$\varepsilon_3 = 0.062 + 2 \cdot 10^{-7} \cdot T_3^2 \quad \text{(3.2)}$$

This correlation is also used here for the gray case. Table 3.2 lists the design parameters for the test setup as well as the constant values above the cut-off wavelength that are used for the optical properties of the glass in the gray case.
Figure 3.5 shows the absorber tube’s heat loss as a function of the average absorber tube temperature for gray and spectral surfaces. Values measured experimentally by Burkholder et al. [45] are also indicated. The agreement between the gray model and the experiments is good for all temperatures. This was expected as the correlation for the absorber tube emissivity used in the gray model was determined in the test report from the very same experimental data. The spectral data agrees less well with the experiments, indicating that the available spectral reflectivity data does not exactly correspond to the investigated coating. The heat loss is under-predicted at lower temperatures and over-predicted at higher temperatures.

In the test report, the emittance of the receiver was determined using the heat loss measurements. However, this approach does not guarantee the accurate calculation of the glass temperature since it depends on a correct heat transfer model and accurate optical properties. The average glass temperature measured in the experiments and determined in the simulations is plotted against the average absorber temperature in Figure 3.6. The computed temperatures for both the gray surface assumption and the spectrally dependent surfaces are shown as well. The agreement of the gray model is good only for lower temperatures. At higher temperatures, the glass temperatures are over-predicted. This is most likely due to inaccurate emittance values of the glass.

Table 3.2: Design parameters of the Schott PTR70 lab tests by Burkholder et al. [45].

<table>
<thead>
<tr>
<th>Design Parameter</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ambient temperature</td>
<td>$T_6$</td>
<td>23°C</td>
</tr>
<tr>
<td>Absorber tube; length</td>
<td>$l_{ab}$</td>
<td>4.06 m</td>
</tr>
<tr>
<td>Absorber tube; inner diameter</td>
<td>$D_2$</td>
<td>0.066 m</td>
</tr>
<tr>
<td>Absorber tube; outer diameter</td>
<td>$D_3$</td>
<td>0.070 m</td>
</tr>
<tr>
<td>Absorber tube; emissivity</td>
<td>$\varepsilon_3$</td>
<td>$f(T_3)$</td>
</tr>
<tr>
<td>Glass envelope; inner diameter</td>
<td>$D_4$</td>
<td>0.115 m</td>
</tr>
<tr>
<td>Glass envelope; outer diameter</td>
<td>$D_5$</td>
<td>0.120 m</td>
</tr>
<tr>
<td>Glass envelope; transmittance</td>
<td>$\tau_{45}$</td>
<td>0.0</td>
</tr>
<tr>
<td>Glass envelope; reflectance</td>
<td>$\rho_{45}$</td>
<td>0.11</td>
</tr>
</tbody>
</table>
Figure 3.5: Off-sun laboratory test: Heat loss of the absorber tube.

Figure 3.6: Off-sun laboratory test: Average glass temperature.
By using the heat transfer model and parameters presented in the report [45], the computed glass temperature yields similarly high values. However, significantly lower glass temperatures are predicted when using the spectral properties for both the absorber tube and the glass envelope. At lower absorber temperatures, the glass temperature is under-predicted, most likely because the emissivity of the selective coating is lower than estimated in the experimental report. At higher absorber temperatures, the emissivity is much higher than determined in the experiments, yet the glass temperature remains lower than measured. The reason for this is primarily the radiation transmitted through the glass, which causes the glass temperature to drop compared to an opaque glass model.

3.4 Conclusions

The 3D optical and thermal model presented in this thesis allows for the incorporation of non-uniform temperature and heat transfer distributions and the identification of critical peak temperatures and heat fluxes. In addition, the use of spectral optical data for the selective coating and for the glass envelope enables a more accurate radiative analysis. Simulation results produced by the heat transfer model agree well with experimental data on thermal efficiency, heat loss, and glass temperatures from both on-sun and off-sun tests. The comparison affirms that the model accurately predicts heat transfer in PTC systems. Besides the beneficial information on peak temperatures and heat fluxes, the 3D model also has the potential to predict glass temperatures more accurately than previous gray models and temperature correlations. In addition, the use of MC ray tracing enables a combined analysis of both the optical and thermal performance of more complicated receiver designs, for example when adding secondary mirror systems, as well as under various operating conditions, such as the solar incidence angle. This detailed and validated heat transfer model can now be used to describe the behavior of new PTC systems, for which no empirical correlations like an IAM exist.
4 Design Point for Performance Analysis

4.1 Introduction

The detailed heat transfer model described in Chapter 2 and validated in Chapter 3 can be used to design and/or optimize PTC systems. In the optimization process, certain parameters that influence the system performance are varied, while those that cannot be controlled are held constant. Among the parameters for which appropriate values need to be chosen in the optimization process are a nominal set of operating conditions – the so-called design point. The design point generally consists of a set of operating conditions and the resulting system output. For the collector field of a CSP system, the most relevant operating condition parameters are the DNI and the sun’s incidence angle at the system aperture. Each CSP plant is designed for such a design point. However, in practice the goal of every CSP system is to produce the maximum amount of electricity or process heat under varying conditions over an entire year of operation. The DNI and the sun’s incidence angle are determined by the geographical location and the meteorological properties of the location of the CSP plant. Since the DNI and the incidence angle are subject to regular (predictable) and irregular (unpredictable) variations, in principle the optimization has to be performed for an appropriately sampled set of values of the DNI and incidence angle. For practical reasons, this is not feasible and the question arises whether the yearly performance can be accurately predicted based on a judiciously chosen set of values for the DNI and the incidence angle at the concentrator aperture.

In the literature, numerous studies of the design, analysis and optimization of CSP systems are presented, but no clear consensus exists on the choice of design point for CSP applications. The majority of the studies use the conditions at the equinox (usually March 21) [52-55] or the summer solstice (June 21 in the northern hemisphere) [19, 56-58]. Less common is the use of the conditions at the winter solstice [8]. Most studies specifically set the design point at solar noon of the particular day [8, 19, 53, 55, 57], while another study uses operating conditions that represent the daily time-average (DTA) of conditions on the day in question [52]. To determine the DNI at the design point and throughout the entire year, both analytical correlations [52-54, 59] and measured data are used [7, 19, 57, 60]. Sources of measurement data include direct measurements of DNI at ground level, estimates from satellite data, and compilations of measured and modeled data for a typical meteorological year (TMY) at the location in question.

With suitable solar incidence angle and DNI values, the performance of the CSP system can be assessed at the design point and over the entire year using various modeling techniques. In cases where detailed models are used and the computational effort to calculate the model response at every operating condition would be excessive, the model response is determined at a limited number of typical days in the year [53-55, 57, 61]. A yearly average is then calculated from the data determined at these days. However, in studies that analyze both the design point and the yearly average, significant differences are found in the modeled behavior for the two cases. For instance, higher system efficiencies are reported for CSP plant models when analyzed at the design point (equinox) than when integrated over the entire year [54, 55].

Numerous optimization studies on CSP systems have also been performed. The majority of studies focus on maximizing the system’s efficiency by optimizing components such as the heliostat field for central receiver systems (CRS) [53, 55], the solar field of PTCs [19, 52, 56, 60, 62], or the power block [7, 55, 57, 62]. It is also observed in these studies that the choice of operating
conditions used as an input to the model has an influence on the result of the optimization process [60, 62]. In the past, relatively few studies have dealt with the analysis and optimization of the absorber tube of PTC systems because detailed information on the optical behavior of the concentrator mirror and the receiver is necessary when using analytical models. Studies that deal with optimizing the absorber diameter determine the optical efficiency by using probability distributions of the optical errors in the various components of the concentrators [56], or by using empirical correlations of the optical efficiency as a function of the incidence angle [19].

In this chapter, the detailed heat transfer model is used to determine the year-round efficiency of PTC systems at various geographic locations. In addition, the thermal efficiency is evaluated at specific operating conditions representing commonly used design points. It is shown that the choice of the design point has a crucial effect on predictions of a system’s nominal efficiency, which can differ significantly from the true yearly average performance. A simple method is presented for determining the design point at which the yearly average performance of a PTC field is accurately predicted. The analysis is repeated at multiple geographic locations and for two PTC designs to demonstrate the wide applicability of the proposed design point. A parameter study on the absorber tube diameter is performed to illustrate the effects of the choice of design point on the predicted optimum system design as well as the system performance.

4.2 Modeling

4.2.1 PTC System

The optical and thermal performance of the solar field of a typical high-temperature PTC system is evaluated using the detailed 3D heat transfer model described in Chapter 2. A single loop of the solar field consists of 48 concentrator modules in series that heat the HTF from 290°C at the inlet of the loop to a
maximum temperature of 550°C at the outlet, which represents the current target of research and development of modern high-temperature PTC systems. The HTF flow rate is adjusted to maintain the same outlet temperature under all operating conditions. Therefore, the receiver surface temperature, and consequently heat loss, remains relatively constant, whereas the heat gain strongly depends on the different operating conditions. The thermal efficiency of the PTC system is defined as the net useable thermal power delivered by the solar field (i.e., net heat gain $\dot{Q}_{\text{gain}} = -\dot{q}_{\text{conv,12}} \cdot I_{\text{system}}$ of the HTF minus the power required to pump the HTF) divided by the DNI over the total aperture area of the concentrator mirrors,

$$\eta_{\text{th}} = \frac{\dot{Q}_{\text{gain}} - P_{\text{pump}}/\eta_{\text{el}}}{I_{\text{DNI}} \cdot A_{\text{conc}}} = \frac{\dot{Q}_{\text{abs}} - \dot{Q}_{\text{loss}} - P_{\text{pump}}/\eta_{\text{el}}}{I_{\text{DNI}} \cdot A_{\text{ap}}}$$  \hspace{1cm} (4.1)

where $P_{\text{pump}}$ is the required pumping power for a fully developed flow [63] (determined using a friction-factor correlation [64]), and $\eta_{\text{el}} = 32.7\%$ is the product of typical efficiencies of the power block, the electric generator, and the HTF pump [20, 65]. The required pumping power is included in the calculation of the thermal efficiency to take into account larger pressure losses caused by smaller absorber tube diameters. These pressure losses become significant at absorber diameters below about 0.04 m. This ensures that the outcome of optimization processes (e.g., in Section 4.3.5) are not biased towards smaller absorber diameters. The HTF is assumed to be a commercial molten salt [66]. The spectral reflectivity curve of a selective coating developed by ENEA, shown in Figure 5.2, is used to calculate the spectral emissivity $\varepsilon_3$ of the absorber tube [67]. A current-generation receiver with an outer absorber tube diameter of 0.07 m is assumed [68]. For the glass envelope, a two-band approach is used with the optical properties described in Section 3.3. The optical parameters of the concentrator mirrors are set to values to produce realistic optical behavior. The selected parameters yield results that agree well with measured incidence-angle-dependent intercept factors [69, 70] and optical efficiencies [13, 71]. The aperture width of the concentrator is 6 m, which is a
Table 4.1: Design parameters of a typical high-temperature PTC solar field.

<table>
<thead>
<tr>
<th>Design Parameter</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ambient temperature</td>
<td>$T_6$</td>
<td>25°C</td>
</tr>
<tr>
<td>HTF temperature</td>
<td>$T_1$</td>
<td>290-550°C</td>
</tr>
<tr>
<td>Length of a solar field loop</td>
<td>$l_{\text{system}}$</td>
<td>584.64 m</td>
</tr>
<tr>
<td>Absorber tube; length</td>
<td>$l_{\text{abs}}$</td>
<td>4.06 m</td>
</tr>
<tr>
<td>Absorber tube; inner diameter</td>
<td>$D_2$</td>
<td>0.064 m</td>
</tr>
<tr>
<td>Absorber tube; outer diameter</td>
<td>$D_3$</td>
<td>0.07 m</td>
</tr>
<tr>
<td>Glass envelope; inner diameter</td>
<td>$D_4$</td>
<td>0.119 m</td>
</tr>
<tr>
<td>Glass envelope; outer diameter</td>
<td>$D_5$</td>
<td>0.125 m</td>
</tr>
<tr>
<td>Concentrator; number of modules</td>
<td>$n_{\text{conc}}$</td>
<td>48</td>
</tr>
<tr>
<td>Concentrator; length</td>
<td>$l_{\text{conc}}$</td>
<td>12 m</td>
</tr>
<tr>
<td>Concentrator; aperture width</td>
<td>$w_{\text{conc}}$</td>
<td>6 m</td>
</tr>
<tr>
<td>Concentrator; rim angle</td>
<td>$\psi_{\text{conc}}$</td>
<td>80°</td>
</tr>
<tr>
<td>Concentrator; reflectivity</td>
<td>$\rho_{\text{conc}}$</td>
<td>0.92</td>
</tr>
<tr>
<td>Concentrator; angular reflection error</td>
<td>$e_{\text{conc}}$</td>
<td>4 mrad</td>
</tr>
<tr>
<td>Concentrator; tracking error</td>
<td>$e_{\text{track}}$</td>
<td>1 mrad</td>
</tr>
</tbody>
</table>

A good approximation of currently used collectors [3]. All relevant thermal, geometric, and optical parameters are summarized in Table 4.1. Although the ambient temperature can also be included in the operating conditions of the design point, it is not considered in this analysis, as its effect is much smaller than that of the DNI and the incidence angle. As a result, a constant ambient temperature of $T_6 = 25^\circ\text{C}$ is used in all simulations.

### 4.2.2 Operating Conditions

The solar incidence angle $\theta$ on the concentrator aperture of north-south oriented PTC systems on level ground at each time during a year is calculated with [72],

$$\cos \theta = \sqrt{\cos^2 \theta_z + \cos^2 \delta \sin^2 \omega}$$

(4.2)

using the local hour angle $\omega$, the solar declination on the $n^{\text{th}}$ day of the year,
\[ \delta = 23.45^\circ \cdot \sin \left( 360^\circ \frac{284 + n}{365} \right) \]  
(4.3)

and the solar zenith angle

\[ \cos \theta_z = \cos \phi \cos \delta \cos \omega + \sin \phi \sin \delta \]  
(4.4)

where \( \phi \) is the location’s latitude. The DNI is determined using an empirical correlation for the atmospheric transmittance as a function of the solar zenith angle and the location’s elevation, assuming clear sky conditions [72, 73],

\[ I_{\text{DNI}} = \left[ a_0 + b_0 \exp \left( -c_0 / \cos \theta_z \right) \right] I_{\text{ex}} \]  
(4.5)

using coefficients for the 23 km visibility model at mid-latitude summer [73],

\[ a_0 = 0.97 \left[ 0.4237 - 0.00821 (6 - E)^2 \right] \]  
(4.6)

\[ b_0 = 0.99 \left[ 0.5055 + 0.00595 (6.5 - E)^2 \right] \]  
(4.7)

\[ c_0 = 1.02 \left[ 0.2711 + 0.01858 (2.5 - E)^2 \right] \]  
(4.8)

and the extraterrestrial radiation on the \( n^{\text{th}} \) day of the year,

\[ I_{\text{ex}} = \left[ 1 + 0.033 \cos \left( 360n / 365 \right) \right] I_0 \]  
(4.9)

where \( E \) is the location’s elevation in km above sea level and \( I_0 = 1367 \text{ W/m}^2 \) is the solar constant [72].

4.2.3 Geographic Location

Four locations are analyzed that represent different combinations of high or low latitude and high or low elevation. These locations therefore experience different distributions of incidence angle and DNI over the course of a year: In general, higher elevations lead to higher DNI due to the thinner atmosphere, while at lower latitudes smaller incidence angles occur. In addition, three of the four locations contain instrument stations from which DNI measurements are readily available. The locations at high latitude are Seville, Spain and Golden, Colorado, USA. For the analysis of low latitude, the location of a solar power plant at Keahole, Hawaii, USA and a potential site in the Chilean Atacama desert [59] are chosen. The geographical position and elevation of all four locations are listed in Table 4.2.
Table 4.2: Position and elevation of all locations.

<table>
<thead>
<tr>
<th>Location</th>
<th>Latitude</th>
<th>Longitude</th>
<th>Elevation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Seville, Spain</td>
<td>37.42° N</td>
<td>5.90° W</td>
<td>30 m</td>
</tr>
<tr>
<td>Golden, USA</td>
<td>39.74° N</td>
<td>105.18° W</td>
<td>1830 m</td>
</tr>
<tr>
<td>Keahole, USA</td>
<td>19.71° N</td>
<td>156.04° W</td>
<td>20 m</td>
</tr>
<tr>
<td>Atacama, Chile</td>
<td>22.32° S</td>
<td>69.33° W</td>
<td>1600 m</td>
</tr>
</tbody>
</table>

4.2.4 Year-Round Performance

Using a location’s position and elevation, the incidence angle at the concentrator aperture is calculated for each point in time of the entire year, and the corresponding DNI is determined using the empirical correlation described above. Since a minimum solar power input is necessary to run a CSP plant, points with a DNI below 200 W/m² are not considered. Figure 4.1 shows an example of data points of $I_{DNI}$ and $\theta$ over the course of one full year for Seville using both the empirical correlation (Figure 4.1a) and TMY data [74] (Figure 4.1b) to determine the DNI. Note that negative incidence angles, corresponding to solar azimuth angles to the north for locations in the northern hemisphere, are reflected to positive. In Figure 4.1a, each lemniscate-shaped loop represents a specific time of day over the course of the entire year, moving from the day of the summer solstice (on the left side of Figure 4.1a), via the autumnal equinox to the day of the winter solstice (on the right side of the plot), and finally via the vernal equinox back to the summer solstice. In this particular plot, each loop is separated by a 30 minute interval from the next, starting from dawn and dusk at the bottom of the plot, to solar noon at the top. In contrast, the scatter plot of $I_{DNI}$ versus $\theta$ for the TMY data, shown in Figure 4.1b, only partially resembles the highly-ordered structure of the empirical DNI correlation. Because the TMY data accounts for real weather conditions, the DNI at any particular time can be higher or lower than predicted by the empirical clear-sky DNI model. The upper limits of DNI agree well for lower incidence angles, but the number of outliers increases significantly for higher $\theta$, even though the overall trend of the DNI at solar noon can still be seen.
Figure 4.1: DNI versus incidence angle over a full year for Seville, Spain, using (a) the empirical correlation and (b) TMY data to determine the local DNI.
The complexity of the 3D heat transfer model applied to determine the solar field performance requires prohibitive computational resources when evaluated at every time during the year. Instead, to determine the year-round performance of the solar field, the model is applied at 28 operating conditions (circles in Figure 4.1a) that are evenly spaced over the range of operating conditions experienced during the year, and the thermal efficiency $\eta_{th}$ is computed at each of these points. A 5th-order polynomial is then fitted to the results, which is subsequently used to predict the efficiency for all points in time over a full year. Using the model fit, a contour plot for each location can be obtained by calculating $\eta_{th}$ over the entire domain of DNI and incidence angle, as shown for Seville in Figure 4.2. The thermal efficiency increases with higher DNI and lower incidence angles, reaching its maximum for $\theta = 0$ and the maximum available DNI. The thermal efficiency is much more strongly dependent on $\theta$ than the DNI, due to decreasing optical efficiencies and increasing cosine losses at higher $\theta$, which reduce the absorbed solar power at the receiver more significantly than a direct reduction of the incident solar radiation.

![Figure 4.2: Thermal efficiency as a function of DNI and incidence angle for Seville.](image-url)
In addition, the yearly average thermal efficiency $\eta_{th,y}$ can be calculated from the local thermal efficiency values assessed for all points in time over the course of an entire year. This yearly average efficiency, defined as the annual thermal output of the solar field divided by the annual DNI input, corresponds to the DNI-weighted average of the local thermal efficiency,

$$\eta_{th,y} = \frac{\int \eta_{th} I_{DNI} \, dt}{\int I_{DNI} \, dt}$$ \hspace{1cm} (4.10)

For instance, for the location of Seville, the conditions of the scatter plots of Figure 4.1 are used to determine $\eta_{th}$ with the 5th-order polynomial model fit at each available data point during the year. Together with the corresponding DNI, determined using either the empirical correlation or TMY data, the yearly average efficiency is calculated according to Eq. (4.10). Besides $\eta_{th,y}$, the polynomial model fit is also used to determine the efficiency for specific DNI and incidence angle values corresponding to the summer solstice and vernal equinox at solar noon (square and triangle in Figure 4.1a). The vernal equinox is assumed to occur at noon on March 21 in the northern hemisphere and on September 23 in the southern hemisphere, while the summer solstice is assumed to occur at noon on June 21 and December 21, respectively.

When available, results from the presented analytical approach are compared to meteorological data. In addition to the empirical correlation for DNI, measured TMY data are used for the locations of Golden and Keahole [75] and Seville [74] (Figure 4.1b). The same 5th-order polynomial fit of solar field efficiency derived from the 3D heat transfer model results is applied to the distribution of $\theta$ and DNI provided by the TMY data, for the purpose of deriving a prediction of $\eta_{th,y}$ based on measured climatic conditions. The yearly average thermal efficiency is again determined by calculating the efficiency at each available data point of the year with a DNI over 200 W/m².
4.3 Results

4.3.1 Year-round Performance

In Table 4.3, the yearly average thermal efficiencies $\eta_{th,y}$ determined using the empirical DNI correlation as well as using TMY data are shown for all four locations. When using the DNI correlation, $\eta_{th,y}$ is higher at lower latitude and higher elevation. The thinner atmosphere at higher elevation leads to higher DNI and hence to a higher solar input. At lower latitudes, the incidence angles are on average smaller, which leads to higher optical efficiencies as well as an increase of the irradiance at the aperture. The overall higher solar input in these cases reduces the importance of thermal losses from the receiver, leading to higher $\eta_{th,y}$. Of the four locations, the highest average efficiency over the course of a full year is reached for Atacama with $\eta_{th,y} = 66.2\%$. The lowest $\eta_{th,y}$ is reached in high-latitude and low-elevation Seville. The same clear trend cannot be seen when using TMY data, mostly because climatic characteristics of each location have an effect of the overall solar input. For instance, when using the 5 km visibility model [73] to calculate the year-round DNI for a location with a more humid climate, such as Keahole, $\eta_{th,y}$ drops from 65.2%, yielded by the 23 km visibility model, to 60.8%, while the actual yearly average thermal efficiency of the TMY data lies in the middle of the two correlation results. In general, when more detailed information on the meteorological characteristics of a specific location is available, more accurate models for yearly irradiation are possible.

<table>
<thead>
<tr>
<th>Location</th>
<th>Empirical DNI</th>
<th>TMY data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Seville, Spain</td>
<td>58.5%</td>
<td>57.5%</td>
</tr>
<tr>
<td>Golden, USA</td>
<td>59.2%</td>
<td>57.4%</td>
</tr>
<tr>
<td>Keahole, USA</td>
<td>65.2%</td>
<td>63.0%</td>
</tr>
<tr>
<td>Atacama, Chile</td>
<td>66.2%</td>
<td>N/A</td>
</tr>
</tbody>
</table>
4.3.2 Analysis Using Conventional Design Points

In addition to the yearly average $\eta_{th,y}$, efficiencies are determined for specific combinations of $I_{DNI}$ and $\theta$ using the 5th-order polynomial model fit, including values corresponding to particular dates and times, as well as the time-averaged DNI and incidence angle over the entire year (YTA). These efficiency values and other results for all four locations are shown in Table 4.4 both when using the empirical correlation and, if available, TMY data to determine the DNI. Note that calculating the yearly average $\eta_{th,y}$ (Table 4.3) and the efficiency at the YTA operating conditions (Table 4.4) does not necessarily yield the same results. In fact, the efficiencies calculated from average DNI and $\theta$ values are significantly higher than the actual yearly average efficiencies. These differences between each design point and the corresponding yearly average efficiency ($\eta_{th} - \eta_{th,y}$) are also listed in Table 4.4. Other sets of operating conditions include the DNI and incidence angle on the summer solstice and vernal equinox at solar noon, as well as DTA operating conditions on the day of the equinox. The DNI values for these design points are determined using the empirical correlation. It can be seen that the calculated efficiencies for each location vary significantly, depending on the chosen set of operating conditions. For Golden, the difference between the efficiency calculated for the solstice and for the equinox is more than 17%, mostly due to the difference in incidence angles. At higher latitudes the efficiency at the equinox is significantly lower than the actual yearly average $\eta_{th,y}$ listed in Table 4.3, while the efficiencies of all other operating conditions are significantly higher than the average. In fact, none of the specific operating conditions of any location yield results that lie within 1% of the calculated annual efficiencies, and differ by as much as 11.5%. Therefore, the conventional choices of operating conditions for evaluating solar field performance are inadequate for predicting the yearly average efficiency of the solar field.
Table 4.4: Thermal efficiencies calculated based on different $I_{DNI}$ and $\theta$ values, together with corresponding optimum absorber diameters.

<table>
<thead>
<tr>
<th>Design point</th>
<th>$I_{DNI}$ [W/m²]</th>
<th>$\theta$ [°]</th>
<th>$\eta_{th}$ [%]</th>
<th>$\eta_{th} - \eta_{th,y}$ [%]</th>
<th>$D_3$ [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Seville, Spain</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Empirical DNI, YTA</td>
<td>620</td>
<td>25.3</td>
<td>62.1</td>
<td>3.6</td>
<td>0.062</td>
</tr>
<tr>
<td>TMY data, YTA</td>
<td>594</td>
<td>25.5</td>
<td>61.6</td>
<td>4.1</td>
<td>0.061</td>
</tr>
<tr>
<td>Solstice, noon</td>
<td>828</td>
<td>14.0</td>
<td>70.0</td>
<td>11.5</td>
<td>0.063</td>
</tr>
<tr>
<td>Equinox, noon</td>
<td>801</td>
<td>37.8</td>
<td>53.6</td>
<td>-4.9</td>
<td>0.070</td>
</tr>
<tr>
<td>Equinox, DTA</td>
<td>645</td>
<td>26.0</td>
<td>61.9</td>
<td>3.4</td>
<td>0.062</td>
</tr>
<tr>
<td><strong>Golden, USA</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Empirical DNI, YTA</td>
<td>782</td>
<td>25.9</td>
<td>63.4</td>
<td>4.2</td>
<td>0.065</td>
</tr>
<tr>
<td>TMY data, YTA</td>
<td>687</td>
<td>26.5</td>
<td>62.0</td>
<td>4.6</td>
<td>0.064</td>
</tr>
<tr>
<td>Solstice, noon</td>
<td>979</td>
<td>16.3</td>
<td>69.7</td>
<td>10.5</td>
<td>0.066</td>
</tr>
<tr>
<td>Equinox, noon</td>
<td>971</td>
<td>40.1</td>
<td>52.2</td>
<td>-7.0</td>
<td>0.075</td>
</tr>
<tr>
<td>Equinox, DTA</td>
<td>801</td>
<td>24.9</td>
<td>64.1</td>
<td>4.9</td>
<td>0.065</td>
</tr>
<tr>
<td><strong>Keahole, USA</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Empirical DNI, YTA</td>
<td>669</td>
<td>17.4</td>
<td>67.1</td>
<td>1.9</td>
<td>0.061</td>
</tr>
<tr>
<td>TMY data, YTA</td>
<td>554</td>
<td>18.2</td>
<td>65.2</td>
<td>2.2</td>
<td>0.059</td>
</tr>
<tr>
<td>Solstice, noon</td>
<td>834</td>
<td>3.7</td>
<td>73.1</td>
<td>7.9</td>
<td>0.063</td>
</tr>
<tr>
<td>Equinox, noon</td>
<td>851</td>
<td>20.1</td>
<td>67.2</td>
<td>2.0</td>
<td>0.065</td>
</tr>
<tr>
<td>Equinox, DTA</td>
<td>679</td>
<td>13.6</td>
<td>68.9</td>
<td>3.7</td>
<td>0.061</td>
</tr>
<tr>
<td><strong>Atacama, Chile</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Empirical DNI, YTA</td>
<td>817</td>
<td>18.0</td>
<td>68.1</td>
<td>1.9</td>
<td>0.064</td>
</tr>
<tr>
<td>Solstice, noon</td>
<td>1039</td>
<td>1.1</td>
<td>75.4</td>
<td>9.2</td>
<td>0.066</td>
</tr>
<tr>
<td>Equinox, noon</td>
<td>988</td>
<td>21.3</td>
<td>67.3</td>
<td>1.1</td>
<td>0.068</td>
</tr>
<tr>
<td>Equinox, DTA</td>
<td>822</td>
<td>13.4</td>
<td>70.1</td>
<td>3.9</td>
<td>0.064</td>
</tr>
</tbody>
</table>
4.3.3 Analysis Using Weighted Averages

Taking the YTA of the DNI and incidence angle over the course of a year consistently over-estimates the efficiency, due to too high values of $I_{DNI}$ and/or too low values of $\theta$. Since these two parameters are not independent variables, taking the average of each separately does not necessarily produce representative operating conditions. The approach proposed here is to use the average value of a parameter that links both variables: the incident direct irradiance at the aperture of the concentrator $I_{solar,conc}$, which is the DNI multiplied with the cosine of the incidence angle (see section 2.3). By using the yearly average $I_{solar,conc}$, expressions for the DNI and the cosine of the incidence angle can be derived, which are essentially the weighted mean values. The average DNI is weighted with $\cos \theta$, and the average incidence angle is weighted with the DNI,

\[
I_{DNI,w} = \frac{\int I_{solar,conc} \cos \theta \, dt}{\int I_{DNI} \cos \theta \, dt} = \frac{\int I_{DNI} \cos \theta \, dt}{\int \cos \theta \, dt}
\]

(4.11)

\[
\cos \theta_w = \frac{\int I_{solar,conc} \, dt}{\int I_{DNI} \, dt} = \frac{\int I_{DNI} \cos \theta \, dt}{\int I_{DNI} \, dt}
\]

(4.12)

In the following, $I_{DNI,w}$ and $\theta_w$ are called the yearly weighted averages (YWA) of the DNI and the incidence angle. In Table 4.5, thermal efficiencies are shown that are determined with the 5th-order polynomial fit at the YWA values of the DNI and the incidence angle according to Eq. (4.11) and (4.12). The values for $I_{DNI}$ are lower and the values for $\theta$ are higher than those obtained for independent time-averages (i.e., YTA) of each quantity (Table 4.4), with the exception of the DNI in Atacama. The efficiencies calculated at these conditions now agree very well with the yearly average efficiencies $\eta_{th,y}$ from Table 4.3, differing at most by 0.3%. By applying this procedure, it is possible to predict the system’s overall yearly performance by evaluating only a single operating condition. The resulting prediction is considerably more accurate than those produced from the commonly used design points listed in Table 4.4.
Table 4.5: Thermal efficiencies calculated with $I_{DNI,w}$ and $\theta_w$ values determined by the yearly weighted average (YWA). The optimum absorber diameter determined at each design point is also shown.

<table>
<thead>
<tr>
<th>Design point</th>
<th>$I_{DNI,w}$ [W/m²]</th>
<th>$\theta_w$ [°]</th>
<th>$\eta_{th}$ [%]</th>
<th>$\eta_{th} - \eta_{th,y}$ [%]</th>
<th>$D_3$ [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Seville, Spain</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Empirical DNI, YWA</td>
<td>619</td>
<td>30.1</td>
<td>58.6</td>
<td>0.1</td>
<td>0.063</td>
</tr>
<tr>
<td>TMY data, YWA</td>
<td>590</td>
<td>30.9</td>
<td>57.6</td>
<td>0.1</td>
<td>0.063</td>
</tr>
<tr>
<td>Golden, USA</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Empirical DNI, YWA</td>
<td>777</td>
<td>31.1</td>
<td>59.5</td>
<td>0.3</td>
<td>0.067</td>
</tr>
<tr>
<td>TMY data, YWA</td>
<td>678</td>
<td>32.2</td>
<td>57.6</td>
<td>0.2</td>
<td>0.065</td>
</tr>
<tr>
<td>Keahole, USA</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Empirical DNI, YWA</td>
<td>667</td>
<td>21.6</td>
<td>64.9</td>
<td>-0.3</td>
<td>0.062</td>
</tr>
<tr>
<td>TMY data, YWA</td>
<td>552</td>
<td>22.8</td>
<td>62.7</td>
<td>-0.3</td>
<td>0.059</td>
</tr>
<tr>
<td>Atacama, Chile</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Empirical DNI, YWA</td>
<td>817</td>
<td>22.1</td>
<td>66.0</td>
<td>-0.2</td>
<td>0.065</td>
</tr>
</tbody>
</table>

4.3.4 Analysis of a Current Generation PTC System

To evaluate whether the results obtained in the previous sections are valid for other PTC systems in addition to that outlined in Section 4.2.1, an identical analysis is performed for a generic PTC system that approximates a current commercial generation. The HTF is changed to synthetic oil [5] and the HTF outlet temperature is reduced to 390°C to match the oil’s operating temperature limit. In addition, the absorber tube’s selective coating is changed. The spectral reflectivity curve of Schott’s NAC coating is used to determine the absorber’s spectral emissivity (see Section 3.3). All other parameters of the PTC system remain unchanged. In the following, results are discussed for the location of Seville. The yearly average thermal efficiency $\eta_{th,y}$ determined with the
Table 4.6: Analysis of a current generation PTC system for Seville, Spain.

<table>
<thead>
<tr>
<th>Design point</th>
<th>$I_{\text{DNI}}$ [W/m²]</th>
<th>$\theta$ [°]</th>
<th>$\eta_{\text{th}}$ [%]</th>
<th>$\eta_{\text{th}} - \eta_{\text{th,y}}$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\eta_{\text{th,y}}$ (Empirical DNI)</td>
<td>-</td>
<td>-</td>
<td>63.1</td>
<td>-</td>
</tr>
<tr>
<td>$\eta_{\text{th,y}}$ (TMY Data)</td>
<td>-</td>
<td>-</td>
<td>62.4</td>
<td>-</td>
</tr>
<tr>
<td>Empirical DNI, YTA</td>
<td>620</td>
<td>25.3</td>
<td>66.9</td>
<td>3.8</td>
</tr>
<tr>
<td>TMY data, YTA</td>
<td>594</td>
<td>25.5</td>
<td>66.6</td>
<td>4.2</td>
</tr>
<tr>
<td>Solstice, noon</td>
<td>828</td>
<td>14.0</td>
<td>73.3</td>
<td>10.2</td>
</tr>
<tr>
<td>Equinox, noon</td>
<td>801</td>
<td>37.8</td>
<td>57.2</td>
<td>-5.9</td>
</tr>
<tr>
<td>Equinox, DTA</td>
<td>645</td>
<td>26.0</td>
<td>66.5</td>
<td>3.4</td>
</tr>
<tr>
<td>Empirical DNI, YWA</td>
<td>619</td>
<td>30.1</td>
<td>63.3</td>
<td>0.2</td>
</tr>
<tr>
<td>TMY data, YWA</td>
<td>590</td>
<td>30.9</td>
<td>62.6</td>
<td>0.2</td>
</tr>
</tbody>
</table>

empirical correlation for DNI and with TMY data, as well as efficiencies calculated at specific operating conditions are shown in Table 4.6. The same trends can be observed as with the first PTC system: except for noon on the equinox, all commonly used operating conditions yield higher efficiencies than the actual yearly average. When using DNI and $\theta$ values determined through the YWA, the calculated efficiencies are within 0.2% of the values determined with the yearly average values. For the three other locations the same trends are observed; detailed results for those locations are listed in Appendix B.

4.3.5 Optimization of Absorber Diameter

Performing an optimization of the PTC system is only practical for a single design point. Thus, it is important to choose the right operating conditions that will optimize the system output over the entire year of operation. For this purpose, the results of optimizing the absorber diameter $D_3$ at various operating conditions are discussed in this section. For all four locations, a parameter study is conducted with varying absorber diameters using the operating conditions listed in Table 4.4 and Table 4.5. Figure 4.3 shows the results of this parameter study for Golden. The thermal efficiency drastically drops for
absorber diameters below 0.06 m due to a lower optical efficiency as well as increased pumping losses. At larger diameters, the efficiency decreases approximately linearly due to higher heat losses caused by the larger absorber surface, while the optical efficiency increases only marginally. It can be seen that the different design points yield different optimum absorber diameters that maximize the thermal efficiency. The optimum absorber diameters for all four locations are also listed for each operating condition in Table 4.4 and Table 4.5. For the case of Golden, it can be seen that the difference in optimum absorber size between using the operating conditions on the equinox at noon and the YTA of the TMY data is 11 mm. The smallest variability in optimum absorber diameter is observed for Keahole. This location also displays the smallest optimum absorber diameters for each design point. In general, higher latitudes and elevations result in larger optimum absorber diameters, as incidence angles are larger and the solar input is higher. Larger incidence angles require larger absorbers to achieve the same optical efficiency due to greater spread in the concentrated solar radiation at the focal line of the concentrator, while a higher solar input reduces the impact of thermal losses from the receiver, also allowing for larger absorber diameters.

Even though operating conditions affect the optimization results, the various absorber sizes do not cause significant differences in the yearly average thermal efficiency. For the case of Golden, the absorber diameters $D_3 = 0.064$ m and $0.075$ m are selected and a 5$^{th}$-order polynomial model fit of the efficiency is derived for each absorber size using the procedure described in Section 4.2.4. The diameters correspond to the optimum absorber size determined for the parameters study using the operating conditions of the YTA of the TMY data and at solar noon on the equinox, respectively. The calculated yearly average thermal efficiency for the two absorber sizes are virtually identical at $\eta_{th,y} = 59.2\%$ and $\eta_{th,y} = 59.1\%$, respectively. These results confirm the previously reported broad optimum in absorber tube diameter [52]. In the case of the absorber diameter, the choice of design point only has a small effect on the
Figure 4.3: Thermal efficiency as a function of the absorber diameter for different $I_{DNI}$ and $\theta$ values for Golden, USA.

overall performance of the system and is hence less critical. Nevertheless, this parameter study illustrates the effect of the design point on the outcome of an optimization process and the importance of choosing the right operating conditions, especially if the analyzed system parameter has a stronger effect on the system response.

### 4.4 Conclusions

Various operating conditions that are commonly selected to design, analyze, and optimize CSP systems were used to compute the thermal efficiency of two different PTC solar fields at four distinct locations. None of them consistently reflect the yearly average efficiency computed analytically and using meteorological data. For a specific system and location, the different design point efficiencies vary significantly and differ by as much as 11.5% from the actual yearly average values. As an alternative, operating conditions determined...
through the weighted averages of DNI and incidence angle are shown to yield results that agree very well with the yearly average efficiency values. For all tested PTC systems and locations, the efficiency of the solar field at the representative operating condition lies within 0.3% of the yearly average efficiency. Using these weighted average values as the design point, it is possible to accurately predict the year-round performance of PTC systems with a single operating condition. In addition, this procedure saves a substantial amount of computational effort. Besides enabling an accurate prediction of performance, choosing the right design point can be important when optimizing system parameters, as different choices of operating conditions can produce different optimum parameters. This procedure may be extended to PTC systems with thermal storage by incorporating a dynamic model of the thermal storage subsystem.
5 Development of a Novel PTC Receiver\textsuperscript{1}

5.1 Introduction

The European Union FP7 project “HITECO” revolves around the development of a new receiver concept for PTC systems. The main goal of the project is to reduce the cost of electricity production achieved by increasing the PTC system’s overall efficiency and lowering the cost of manufacturing, installation, maintenance, and operation. The increase in efficiency is accomplished through three measures. First, the maximum HTF temperature is raised from 400°C, the industry standard for many years, to 600°C. The higher outlet temperatures of the solar field lead to increased efficiencies in the power cycle. To reach these temperatures, a molten salt HTF is being developed. Second, heat losses from the receiver are reduced partly through the implementation of a novel active vacuum system. This system allows controlling the pressure level inside the vacuum annulus of the receiver and purging unwanted gases, such as leaked hydrogen or air, with a heavy noble gas to reduce the molecular conduction across the annulus. In conventional receivers the vacuum in the annulus is intended to last for the entire lifecycle, so a very low pressure is established during manufacture. The active pressure control of the new vacuum system allows for higher pressures, which are less energetically expensive to maintain, reduce strain on components, and do not significantly increase heat losses. Other design improvements include the development of new receiver seals that replace the failure-prone conventional glass-to-metal seals, improved glass properties for the protective envelope, and an increase in the receiver length (see Section 6.3.4).

The third measure used to improve the system efficiency is to ensure good optical and thermal performance of the receiver at higher temperatures, for which a stable high-temperature selective coating for the absorber tube is required. Selective coatings for PTC receivers generally consist of ceramic-metallic multilayers called cermets. Due to the changes in refractive index between the different material layers, radiation is absorbed efficiently inside the multilayer [76, 77]. The selective coating generally consists of an IR reflective, metallic base on the substrate (in a PTC this is simply the stainless steel absorber tube), several cermet multilayers with various metal concentrations, and an AR coating on the top of the cermet [76, 77]. In Figure 5.1, a schematic of such a multilayer coating is shown. The IR reflective layer reduces the emittance of thermal radiation of the absorber tube and is usually made out of molybdenum or silver [67, 77]. The AR coating enhances the absorption of the incident solar radiation, using metal oxides with small refractive indices, such as SiO₂, MgF₂, or SiN₄. For the cermet multilayer generally transition metals (Mo, Ag, Cu, Ni, Fe, Cr, W, Pt, Ti, Al) are used in combination with different types of oxides, oxynitrides, or nitrides (Al₂O₃, SiO₂, CeO₂, ZnS, AlN, NbAlON, HfON, Si₃N₄) [67, 76, 77]. The layers of the selective coating are applied to the substrate with various physical vapor deposition techniques, such as vacuum evaporation, radiofrequency sputtering, or reactive direct current sputtering [76-78].
5.2 Comparison of Novel Solar Selective Coatings

In the framework of the HITECO project, a novel selective coating based on Mo-SiN$_4$ is developed with the goal of improved solar absorption and thermal emission performance at temperatures of up to 600$^\circ$C. Several candidate coatings have been produced with different layer thicknesses and filling factors [77]. In Figure 5.2, the spectral reflectivity of six Mo-SiN$_4$ candidate coatings with varying Mo content is plotted. For comparison, the spectral reflectivity curves of previously reported selective coatings developed by ENEA (see section 4.2.1) and Schott (see section 3.3) are also shown.

The new receiver and mirror geometries of the HITECO design require a heat transfer model that does not rely on historical correlations, as no data is available for such a design. As a result, the optical and thermal performance of the developed HITECO coatings is evaluated using the detailed 3D heat transfer model described in Chapter 2, to determine the best coating to be

![Figure 5.2: Spectral reflectivity of candidate selective coatings, together with existing coatings from ENEA and Schott (NAC). The preliminary selective coating (PSC) used in the first off-sun tests is also shown.](image-url)
implemented in the HITECO receiver. In a first step, solar absorptivity and thermal emissivity of the candidate coatings is calculated at 400°C and 600°C and compared to the coatings of ENEA and Schott. In a second step, the overall performance of all six HITECO candidate coatings is analyzed by simulating their use in a receiver in a generic solar field loop. The model determines the optical and thermal performance of each selective coating using fixed system parameters and the spectral data of the coatings.

5.2.1 Solar Absorptivity and Thermal Emissivity

Using the spectral reflectivity curves of the candidate selective coatings and the spectral data of the existing coatings, the solar-weighted absorptivity and the thermal emissivity is calculated at 400°C and 600°C. The solar absorptivity $\alpha_3$ is determined by weighting the spectral reflectivity data $\rho_3(\lambda)$ shown in Figure 5.2 with the ASTM solar spectrum $I_\lambda(\lambda)$,

$$\alpha_3 = \frac{\int_{\lambda_1}^{\lambda_2} (1 - \rho_3(\lambda)) \cdot I_\lambda(\lambda) d\lambda}{\int_{\lambda_1}^{\lambda_2} I_\lambda(\lambda) d\lambda}$$  \hspace{1cm} (5.1)

with $\lambda_1 = 0.25 \mu m$ and $\lambda_2 = 4 \mu m$ to account for the limits considered in the ASTM solar spectrum. The thermal emissivity at a given temperature $\varepsilon_3(T)$ is determined by weighting the spectral reflectivity data with the blackbody spectral emissive power $e_{3b}(\lambda, T)$,

$$\varepsilon_3(T) = \frac{\int_0^\infty (1 - \rho_3(\lambda)) \cdot e_{3b}(\lambda, T) d\lambda}{\int_0^\infty e_{3b}(\lambda, T) d\lambda}$$  \hspace{1cm} (5.2)

where, using Planck’s law,

$$e_{3b}(\lambda, T) = \frac{2\pi C_1}{\lambda^5 \left(e^{C_2/\lambda T} - 1\right)}$$  \hspace{1cm} (5.3)

with $C_1 = 0.59552 \cdot 10^8$ W·µm⁴/(m²·sr) and $C_2 = 14'387.75$ µm·K [79]. For the calculation of absorptivity and emissivity, the spectral reflectivity data $\rho_3(\lambda)$ is assumed to be constant below and above the spectral data limits, at values equal
Table 5.1: Solar-weighted absorptivity and thermal emissivity at 400°C and 600°C for all candidate selective coatings, as well for spectral data from literature.

<table>
<thead>
<tr>
<th>Coating</th>
<th>$\alpha_3$ [–]</th>
<th>$\varepsilon_3(400^\circ\text{C})$ [–]</th>
<th>$\varepsilon_3(600^\circ\text{C})$ [–]</th>
</tr>
</thead>
<tbody>
<tr>
<td>S2</td>
<td>0.687</td>
<td>0.022</td>
<td>0.037</td>
</tr>
<tr>
<td>S5</td>
<td>0.903</td>
<td>0.089</td>
<td>0.170</td>
</tr>
<tr>
<td>S10</td>
<td>0.941</td>
<td>0.140</td>
<td>0.250</td>
</tr>
<tr>
<td>S13</td>
<td>0.866</td>
<td>0.039</td>
<td>0.075</td>
</tr>
<tr>
<td>S14</td>
<td>0.926</td>
<td>0.055</td>
<td>0.109</td>
</tr>
<tr>
<td>S19</td>
<td>0.905</td>
<td>0.049</td>
<td>0.108</td>
</tr>
<tr>
<td>ENEA</td>
<td>0.939</td>
<td>0.071</td>
<td>0.122</td>
</tr>
<tr>
<td>NAC</td>
<td>0.957</td>
<td>0.088</td>
<td>0.169</td>
</tr>
</tbody>
</table>

to the first and last data point. In Table 5.1 the calculated values for $\alpha_3$ and $\varepsilon_3$ are listed for all coatings. The NAC is seen to have the highest solar absorptivity of $\alpha_3 = 0.957$. Candidate coating S14 has an absorptivity that is slightly below that of the ENEA coating. The high-temperature coatings ENEA and S14 (temperature limit: 600°C) also display more favorable emissivity values at higher temperature levels compared to the NAC (temperature limit: 400°C), with S14 having a lower emissivity than the ENEA coating. With this data, a first assessment of the quality of the various new coatings can be made, but a more thorough analysis is necessary, as only isolated properties were determined and not the actual performance in a PTC system.

5.2.2 Optical and Thermal Analysis

In addition to calculating solar-weighted absorptivity and thermal emissivity, a full optical and thermal analysis is performed, using the detailed 3D heat transfer model described in Chapter 2. This enables a more comprehensive evaluation of the potential of the selective coatings in a solar field of a PTC system as the competing effects of solar absorptivity and thermal emissivity are included. Various performance parameters, such as heat loss and heat gain, as
well as optical and thermal efficiencies are assessed for a solar field located in Seville, Spain. A solar incidence angle of \( \theta = 30.1^\circ \) and a DNI of \( I_{\text{DNI}} = 619 \text{ W/m}^2 \) is used. These operating conditions, determined through weighted yearly average values, represent a design point that produces thermal efficiencies that are lower than peak values and are more similar to the expected yearly average performance (see Table 4.5). Two solar field configurations are analyzed: a high-temperature system using a commercial molten salt HTF [66] and an outlet temperature of 550°C, and a low-temperature system using synthetic oil [5] with an outlet temperature of 390°C. All other system parameters of the solar field are identical to those described in Section 4.2.1. The thermal efficiency is determined from Eq. (4.1). The optical efficiency of the PTC system is defined as the solar power absorbed by the absorber tube divided by the DNI over the total aperture area of the concentrator mirrors (see Chapter 2),

\[
\eta_{\text{opt}} = \frac{\dot{Q}_{\text{abs}}}{I_{\text{DNI}} \cdot A_{\text{conc}}}
\]  

(5.4)

In Table 5.2, results are shown for the thermal analysis of the high-temperature PTC system, using molten salt as the HTF and an outlet temperature of 550°C. The heat gain and consequently the thermal efficiency of the coatings vary significantly, since \( \dot{Q}_{\text{loss}} \) and \( \eta_{\text{opt}} \) depend strongly on the spectral

<table>
<thead>
<tr>
<th>Coating</th>
<th>( \dot{Q}_{\text{gain}} ) [MW]</th>
<th>( \dot{Q}_{\text{loss}} ) [MW]</th>
<th>( \eta_{\text{opt}} ) [%]</th>
<th>( \eta_{\text{th}} ) [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>S2</td>
<td>8.92</td>
<td>0.45</td>
<td>48.7</td>
<td>46.3</td>
</tr>
<tr>
<td>S5</td>
<td>10.36</td>
<td>1.87</td>
<td>63.5</td>
<td>53.8</td>
</tr>
<tr>
<td>S10</td>
<td>9.79</td>
<td>2.92</td>
<td>66.0</td>
<td>50.8</td>
</tr>
<tr>
<td>S13</td>
<td>10.92</td>
<td>0.82</td>
<td>61.0</td>
<td>56.7</td>
</tr>
<tr>
<td>S14</td>
<td>11.38</td>
<td>1.16</td>
<td>65.1</td>
<td>59.1</td>
</tr>
<tr>
<td>S19</td>
<td>11.17</td>
<td>1.09</td>
<td>63.7</td>
<td>58.0</td>
</tr>
<tr>
<td>ENEA</td>
<td>11.29</td>
<td>1.41</td>
<td>66.0</td>
<td>58.6</td>
</tr>
<tr>
<td>NAC</td>
<td>11.08</td>
<td>1.85</td>
<td>67.2</td>
<td>57.5</td>
</tr>
</tbody>
</table>
behavior. Of all the candidate coatings, S14 displays the best performance with \( \eta_{th} = 59.1\% \), which is 0.5\% higher than the efficiency achieved with the ENEA coating. The lower solar absorptivity of S14 is compensated by the reduced heat losses due to the overall lower thermal emissivity compared to the ENEA coating. The worst performance is observed with coating S2, for which the low optical efficiency leads to a very low heat gain that cannot be compensated by the lowest thermal emissivities, and consequently the lowest heat losses, of all the coatings.

In Table 5.3, results are shown for the analysis performed assuming a low-temperature PTC system, using synthetic oil as the HTF and an outlet temperature of 390°C. For this configuration, the NAC displays the best thermal efficiency with \( \eta_{th} = 63.3\% \). The lower temperatures reduce the heat losses from the absorber and hence increase the influence of the solar absorptivity on the thermal efficiency. The high absorptivity of the NAC thus enables a high thermal performance. This effect also equalizes the performance of the ENEA and S14 coatings, both reaching virtually identical thermal efficiencies.

<table>
<thead>
<tr>
<th>Coating</th>
<th>( \dot{Q}_{\text{gain}} ) [MW]</th>
<th>( \dot{Q}_{\text{loss}} ) [MW]</th>
<th>( \eta_{\text{opt}} ) [%]</th>
<th>( \eta_{\text{th}} ) [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>S2</td>
<td>9.16</td>
<td>0.22</td>
<td>48.7</td>
<td>47.4</td>
</tr>
<tr>
<td>S5</td>
<td>11.52</td>
<td>0.71</td>
<td>63.5</td>
<td>59.6</td>
</tr>
<tr>
<td>S10</td>
<td>11.62</td>
<td>1.10</td>
<td>66.0</td>
<td>60.1</td>
</tr>
<tr>
<td>S13</td>
<td>11.40</td>
<td>0.34</td>
<td>61.0</td>
<td>59.0</td>
</tr>
<tr>
<td>S14</td>
<td>12.08</td>
<td>0.46</td>
<td>65.1</td>
<td>62.4</td>
</tr>
<tr>
<td>S19</td>
<td>11.86</td>
<td>0.40</td>
<td>63.7</td>
<td>61.3</td>
</tr>
<tr>
<td>ENEA</td>
<td>12.11</td>
<td>0.60</td>
<td>66.0</td>
<td>62.6</td>
</tr>
<tr>
<td>NAC</td>
<td>12.24</td>
<td>0.70</td>
<td>67.2</td>
<td>63.3</td>
</tr>
</tbody>
</table>

Table 5.3: Results of the heat transfer analysis for a low-temperature PTC solar field, using synthetic oil with a maximum HTF temperature of 390°C.
5.3 Off-Sun Heat Loss Tests

Based on the results presented in Section 5.2, the selective coating S14 is selected for the subsequent implementation in the HITECO receiver. A receiver prototype is evaluated on an off-sun test bench in terms of its heat loss and the performance of systems components such as the vacuum system and bridle seals. For this purpose, two receiver sections are heated to temperatures of up to 600°C, while the required heating power, temperatures, and pressures are measured at several locations. Figure 5.3 shows a rendering of the receiver prototype with the two receiver sections, as well as sealing, connection, and expansion components. To determine the required specifications of the test equipment, in particular the heating system, and to assess at what locations on the receiver temperature sensors need to be placed, the off-sun test setup is analyzed using the detailed 3D heat transfer model and the spectral data of the selective coating S14.

![Figure 5.3: Rendering of the receiver prototype for off-sun testing consisting of two receiver sections, as well as sealing, connection, and expansion components (not to scale). In addition, locations of temperature and voltage measurements are shown.](http://www.hitecoproject.com/wp-content/uploads/2010/12/Tubo.jpg, Accessed 01/14)
5.3.1 Modeling

In this analysis, the heat loss from the absorber tube at temperatures $T_2$ (see Figure 2.1) is determined for an off-sun test setup without incident solar radiation and HTF. The calculated heat loss is equal to the heating power required by the heating system to reach steady-state for a given set of design parameters. Apart from the absorber tube temperature, these design parameters include the pressure and composition of the gas in the vacuum annulus, as well as operating conditions such as the ambient temperature. Relevant design parameters are listed in Table 5.4. The proposed HITECO prototype has a larger geometry than current-generation receivers, with an outer absorber diameter of 8 cm and a HCE length of around 6 m. The spectral reflectivity data of the selective coating S14 is used to calculate the spectral emissivity of the absorber. At 100°C the absorber’s thermal emissivity is around 2%, and at 600°C the emissivity is approximately 11%, see Section 5.2.1. The heat loss from the absorber is determined for inner absorber tube temperatures $T_2$ ranging from 100°C to 600°C. The pressure in the vacuum annulus is varied between 0.01 Pa and 100 Pa for atmospheric air, pure krypton, and pure hydrogen. Air represents the benchmark condition for heat loss. Krypton is used as an option to reduce the heat losses. By purging the vacuum chamber with the heavy noble gas, the conductive heat transfer from the absorber tube to the glass envelope can be lowered.

<table>
<thead>
<tr>
<th>Design Parameter</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ambient temperature</td>
<td>$T_6$</td>
<td>22°C</td>
</tr>
<tr>
<td>Vacuum annulus pressure</td>
<td>$p_{34}$</td>
<td>0.1-100 Pa</td>
</tr>
<tr>
<td>Absorber tube; inner temperature</td>
<td>$T_2$</td>
<td>100-600°C</td>
</tr>
<tr>
<td>Absorber tube; length</td>
<td>$l_{abs}$</td>
<td>6.03 m</td>
</tr>
<tr>
<td>Absorber tube; inner diameter</td>
<td>$D_2$</td>
<td>0.074 m</td>
</tr>
<tr>
<td>Absorber tube; outer diameter</td>
<td>$D_3$</td>
<td>0.08 m</td>
</tr>
<tr>
<td>Glass envelope; inner diameter</td>
<td>$D_4$</td>
<td>0.1535 m</td>
</tr>
<tr>
<td>Glass envelope; outer diameter</td>
<td>$D_5$</td>
<td>0.16 m</td>
</tr>
</tbody>
</table>
Hydrogen represents the worst-case scenario, as the light molecules significantly increase the heat transfer across the vacuum annulus. In conventional PTC systems that use synthetic oils as HTF, hydrogen is produced during the partial degradation of the oil at higher temperatures. By diffusing into the vacuum annulus, hydrogen negatively affects the performance of the receiver.

In Figure 5.4, simulation results for the case of air in the vacuum annulus are shown. The total heat loss is plotted as a function of the inner absorber tube temperature for different annulus gas pressures. It can be seen that at lower pressures, the heat losses are not significantly affected by the gas pressure. Below 0.1 Pa, the molecular conduction through the vacuum annulus becomes negligible, and lower pressures do not reduce the heat losses significantly. At higher pressures, a similar trend can be observed. At pressures above 10 Pa, the heat losses only increase marginally with higher gas pressures, as the fluid approaches the transition from the slip regime to the continuum regime. At very low pressures of 0.01 Pa, the total heat losses at $T_2 = 500^\circ\text{C}$ are around

![Figure 5.4: Total heat loss as a function of the inner absorber tube temperature for different air pressures.](image-url)
4.98 kW, which corresponds to 413 W/m of heat loss per unit length of HCE. Compared to the Schott PTR70 receiver (see Section 3.3), which has a heat loss of around 481 W/m \[45\] at these temperatures, the HITECO prototype is expected to have a lower heat loss even though the absorber tube area is 14\% larger.

In the following, results for the case of the vacuum annulus filled with air at 100 Pa are discussed as a worst-case scenario. Among all analyzed conditions, heat losses are at their maximum in this particular case mostly due to the high conductive heat transfer across the annular gap. Results for this case are summarized in Table 5.5, which lists surface temperatures as well as heat transfer rates as a function of inner absorber tube temperatures, see Figure 2.1. It can be seen that the maximum heat loss at \(T_2 = 600^\circ\text{C}\) is 13.47 kW (or 1116 W/m). Even with these high pressures, the outer glass temperature at \(T_2 = 500^\circ\text{C}\) is only 84.5°C, which is significantly lower than the measured glass temperature of around 95°C of the PTR70 receiver \[45\] at the same absorber tube temperatures. This is partly due to the larger diameter of the HITECO receiver, which increases the heat transfer to the environment.

<table>
<thead>
<tr>
<th>(T_2) [°C]</th>
<th>100.0</th>
<th>200.0</th>
<th>300.0</th>
<th>400.0</th>
<th>500.0</th>
<th>600.0</th>
</tr>
</thead>
<tbody>
<tr>
<td>(T_3) [°C]</td>
<td>99.9</td>
<td>199.9</td>
<td>299.9</td>
<td>399.8</td>
<td>499.7</td>
<td>599.5</td>
</tr>
<tr>
<td>(T_4) [°C]</td>
<td>27.7</td>
<td>36.5</td>
<td>48.4</td>
<td>64.8</td>
<td>86.7</td>
<td>113.8</td>
</tr>
<tr>
<td>(T_5) [°C]</td>
<td>27.6</td>
<td>36.2</td>
<td>47.7</td>
<td>63.5</td>
<td>84.5</td>
<td>110.4</td>
</tr>
<tr>
<td>(\dot{Q}_{\text{cond},34}) [kW]</td>
<td>0.24</td>
<td>0.61</td>
<td>1.03</td>
<td>1.5</td>
<td>2.01</td>
<td>2.55</td>
</tr>
<tr>
<td>(\dot{Q}_{\text{rad},34}) [kW]</td>
<td>0.05</td>
<td>0.21</td>
<td>0.60</td>
<td>1.42</td>
<td>2.86</td>
<td>5.09</td>
</tr>
<tr>
<td>(\dot{Q}_{\text{rad},36}) [kW]</td>
<td>0.00</td>
<td>0.01</td>
<td>0.09</td>
<td>0.50</td>
<td>1.92</td>
<td>5.61</td>
</tr>
<tr>
<td>(\dot{Q}_{\text{cond},\text{loss}}) [kW]</td>
<td>0.02</td>
<td>0.05</td>
<td>0.09</td>
<td>0.13</td>
<td>0.17</td>
<td>0.21</td>
</tr>
<tr>
<td>(\dot{Q}_{\text{rad},56}) [kW]</td>
<td>0.18</td>
<td>0.48</td>
<td>0.92</td>
<td>1.61</td>
<td>2.68</td>
<td>4.30</td>
</tr>
<tr>
<td>(\dot{Q}_{\text{conv},56}) [kW]</td>
<td>0.10</td>
<td>0.33</td>
<td>0.71</td>
<td>1.30</td>
<td>2.17</td>
<td>3.33</td>
</tr>
<tr>
<td>(\dot{Q}_{\text{loss}}) [kW]</td>
<td>0.31</td>
<td>0.88</td>
<td>1.82</td>
<td>3.55</td>
<td>6.96</td>
<td>13.47</td>
</tr>
</tbody>
</table>
The heat loss for the case of krypton in the vacuum chamber is plotted in Figure 5.5. It can be seen that the heat loss does not significantly increase at higher pressures. At very low pressures, the conductive heat transfer is almost nonexistent, identical to the air case. As the pressure increases, the heavy noble gas keeps the conduction losses lower compared to the case of air. In Figure 5.6, heat loss as a function of the absorber temperature is plotted for different pressures of hydrogen in the vacuum chamber. The heat loss at low pressures is similar to the first two cases, mostly due to the insignificance of conduction at these pressures, even when using hydrogen. At higher pressure, however, the heat losses increase significantly, reaching values about twice as large as in the case of air. The reasons for the large losses when using hydrogen can be seen in Table 5.6, where surface temperatures and heat transfer rates are summarized for the three gases at pressures of 100 Pa and $T_2 = 600{^\circ C}$. The larger heat losses are a result of the large conductive heat transfer from the absorber tube to the glass envelope, due to the small molecular weight of hydrogen. This increased

![Figure 5.5: Total heat loss as a function of the inner absorber tube temperature for different krypton pressures.](image-url)
heat transfer also causes much higher glass temperatures, reaching values of over 200°C with hydrogen and around 100°C for air and krypton. From the table, it becomes apparent that a maximum heating power of around 25 kW is required to reach temperatures of 600°C with hydrogen present in the vacuum annulus.

To heat the prototype receiver in an off-sun test setup, several possibilities exist to supply the required power to the absorber tube to overcome the heat losses. A convenient way is to uniformly heat the absorber tube through impedance heating, by passing a high electric current through the absorber tube. Using tabulated data for the electrical resistivity [80] of the stainless steel absorber tube, the required currents and the associated voltage drop across the entire prototype can be determined. For a heating system with a maximum electric power output of around 25 kW, a current of around 1000 A is required. This electric current results in a voltage drop of around 25 V across the 12 m prototype receiver.

![Figure 5.6: Total heat loss as a function of the inner absorber tube temperature for different hydrogen pressures. Note the different scale of the ordinate axis.](image)

Figure 5.6: Total heat loss as a function of the inner absorber tube temperature for different hydrogen pressures. Note the different scale of the ordinate axis.
Table 5.6: Surface temperatures and modes of heat transfer for different annulus gases at $p_{34} = 100 \text{ Pa}$ and $T_2 = 600^\circ\text{C}$.

<table>
<thead>
<tr>
<th>Annulus gas</th>
<th>Air</th>
<th>Kr</th>
<th>H$_2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_3$ [°C]</td>
<td>599.5</td>
<td>599.6</td>
<td>599.0</td>
</tr>
<tr>
<td>$T_4$ [°C]</td>
<td>113.8</td>
<td>98.7</td>
<td>208.9</td>
</tr>
<tr>
<td>$T_5$ [°C]</td>
<td>110.4</td>
<td>95.9</td>
<td>199.4</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{cond},34}$ [kW]</td>
<td>2.55</td>
<td>0.94</td>
<td>15.76</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{rad},34}$ [kW]</td>
<td>5.09</td>
<td>5.11</td>
<td>4.91</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{rad},36}$ [kW]</td>
<td>5.61</td>
<td>5.61</td>
<td>5.58</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{cond,loss}}$ [kW]</td>
<td>0.21</td>
<td>0.21</td>
<td>0.21</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{rad},56}$ [kW]</td>
<td>4.30</td>
<td>3.35</td>
<td>12.91</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{conv},56}$ [kW]</td>
<td>3.33</td>
<td>2.67</td>
<td>7.73</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{loss}}$ [kW]</td>
<td>13.47</td>
<td>11.87</td>
<td>26.48</td>
</tr>
</tbody>
</table>

5.3.2 Test Bench

With the specifications for the heating system determined by modeling the receiver prototype, the off-sun test bench is designed. The test bench essentially consists of the receiver prototype, the supporting structure, the vacuum system, the heating system, and the data acquisition (DAQ) system needed to record all measurement data. As mentioned above, the receiver consists of one long absorber tube that is supported and sealed by bridle seals and two 6 m long glass envelopes (see Figure 5.3). At one end of the receiver, the expansion bellows are attached to the absorber tube and the bridle support to accommodate the thermal expansion of the hot absorber tube. At the other end, the absorber is sealed by a cylindrical component that enables the connection to the vacuum system. In the center of the receiver, a third bridle seal is used to seal the receiver, connect the two glass sections, and support the absorber tube. A supporting frame similar to those used in current-generation solar fields is used to support the receiver at the three bridle seals.

The vacuum system consists of a roughing pump and a turbo-molecular pump connected in series, pressure sensors positioned at several locations on the receiver and the piping network, and pressurized gas bottles connected to
system. This system allows for the control of the pressure inside the vacuum annulus of the receiver by pumping unwanted gases out and refilling the receiver volume with a particular gas. Nitrogen can be used to simulate the behavior at different air pressures without risking oxidation of the selective coating. Using krypton allows assessing the potential of heavy noble gases to reduce heat losses, as shown in Section 5.3.1. For safety reasons, hydrogen is not used to assess the worst-case scenario of heat conduction across the annular gap, but alternative gases like helium may be considered. To heat the absorber tube to a specific temperature, an impedance heating system is selected based on the specifications determined in the previous section. The heating system is basically a large transformer that converts a normal-voltage alternate current from a regular 63 A outlet to a high direct current. Thick copper cables are used to connect the heating system directly to the absorber tube at each end of the receiver.

Finally, the DAQ system is a collection of devices used to measure and record various physical quantities. Thermocouples measure temperatures at various locations on the receiver, most prominently on the absorber tube surface. The absorber temperature readings serve as an input to the DAQ system that controls the current output of the heating system, and are used for the calculation of the temperature-dependent heat loss of the receiver prototype. In addition to the temperature, the voltage is measured at several positions on the absorber tube to calculate the local heat generation using the output current of the heating system. At the steady-state, the dissipated heat in the two main sections of the receiver is equal to the total heat losses from the absorber tube. The DAQ system is also used to record measurement data of the vacuum system, such as pressure readings, and to control components like the vacuum pumps, valves, or the gas refilling feature. Lastly, a mass spectrometer is connected to the receiver to analyze the composition of the gas inside the vacuum annulus of the receiver. With this method, information on leak rates and the outgassing behavior of the system can be gained, for instance what kind of gases permeate through, or are adsorbed on, the receiver walls.
5.3.3 Preliminary Selective Coating

The industrial process with which the selective coating is continuously applied to long stainless-steel tubes is still under development. As part of this development, a first set of absorber tubes with a stable coating was produced and analyzed. This preliminary selective coating (PSC) has favorable optical properties only in the visible part of the spectrum, as can be seen in Figure 5.2 showing the spectral reflectivity of the PSC. The reflectivity in the IR spectrum is rather low, causing significantly higher thermal emittance than for optimized coatings. The coatings currently produced at the industrial scale clearly do not yet have the optical properties that were determined for the optimum coating candidate S14 that had been produced at the lab scale. Nevertheless, the PSC represents preliminary proof that solar selective coatings can be manufactured industrially with the innovative, continuously running process. This first set of PSC-coated absorber tubes also provide an opportunity to test the experimental setup and measurement equipment and ensure their readiness once the optimum selective coating is available at the industrial scale. For this purpose, preliminary off-sun tests are conducted using the PSC-coated absorber tubes. In addition to testing the experimental setup, these preliminary experimental data can be compared to simulation results generated by the heat transfer model. The unfavorable optical properties of the PSC lead to results that are much different from the modeling studies of Sections 3.3 and 5.3.1 where optimized coatings are used. This additional set of experimental data is therefore especially valuable, as it allows model validation at higher levels of heat transfer.

In this preliminary test campaign, the absorber tubes coated with the PSC are used to assemble a receiver prototype, which is then installed in the off-sun test setup described above. Heat loss and surface temperatures are measured for different absorber tube temperatures and vacuum annulus pressures. The high emissivity of the PSC allows for evaluating maximum temperatures that receiver components may experience and thus testing them near their critical temperature limits. Due to the high emissivity, the maximum absorber
temperature in the tests is 400°C at which point the power capacity of the impedance heating system is reached. To protect the selective coating from possible oxidation, the pressure in the vacuum annulus is kept at low values on the order of 1e-3 and 1e-4 mbar.

5.3.4 Preliminary Test Results

Among the surface temperatures measured in the off-sun tests, the average temperature on the outside of the absorber tube \( T_3 \) is the most important, as it serves as an input to the control of the heating system and is the main driver of the heat losses of the system. The average absorber temperature is determined using two temperature measurements at the center of each receiver section (see Figure 5.3),

\[
T_3 = \frac{T_{3,B} + T_{3,C}}{2} \tag{5.5}
\]

Analogously, the average temperature at the outer glass envelope surface \( T_5 \) is calculated using the readings at \( T_{5,B} \) and \( T_{5,C} \). The glass temperature serves as a measure of the temperatures reached by the receiver components. The glass temperature can also be compared to values obtained in other off-sun test setups and to results from simulations. The main parameter to be determined in the off-sun test is, however, the total heat loss from the absorber tube \( \dot{Q}_{\text{loss}} \).

The heat loss is determined using the electrical power across the receiver \( I_{\text{el}}(V_{\text{el},A} - V_{\text{el},D}) = I_{\text{el}}V_{\text{el}} \) and the contribution of axial conduction from the end parts of the absorber tube,

\[
\dot{Q}_{\text{loss}} = I_{\text{el}}V_{\text{el}} + \frac{k_3 \cdot A_{23}}{l_{\text{cond}}} (T_{3,A} - T_{3,B}) + \frac{k_3 \cdot A_{23}}{l_{\text{cond}}} (T_{3,D} - T_{3,C}) \tag{5.6}
\]

where \( k_3 \) is the thermal conductivity at the mean temperature (see Section 2.4), \( A_{23} = \pi(D_3^2 - D_2^2)/4 \) is the cross-sectional area of the absorber tube, and \( l_{\text{cond}} \) is the estimated effective conduction length from the ends of the receiver (Positions \( A \) and \( D \) in Figure 5.3) to the point where the absorber tube reaches the constant temperatures recorded at the center of the receiver (Positions \( B \) and
Measurements with a thermographic camera and an IR thermometer on the outside of the glass envelope showed that this length is approximately 0.1 m. Due to the use of an impedance heating system, the ends of the absorber tube are also heated and reach temperatures that are very similar to the values along the main sections of the receiver. This leads to axial conduction that is much smaller than in test setups using cartridge heaters. Therefore, the choice of $l_{\text{cond}}$ does not have a significant effect on the overall heat loss of the absorber tube. Nevertheless, to account for the inaccuracy in the estimation of the conduction length, a measurement bias of $\Delta l_{\text{cond,bias}} = 0.1$ m is used in the calculation of the uncertainty (see next section).

Table 5.7: Results of the preliminary off-sun tests, including annulus pressure, ambient temperature, average absorber and glass temperature, and heat loss from the absorber tube. The measurement uncertainty as described in Section 5.3.5 is also shown.

<table>
<thead>
<tr>
<th>$p_34$ [mbar]</th>
<th>$T_6$ [°C]</th>
<th>$T_3$ [°C]</th>
<th>$Q_{\text{loss}}$ [kW]</th>
<th>$T_5$ [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0e-4</td>
<td>20.9</td>
<td>103.3 ±1.1</td>
<td>0.48 ±0.01</td>
<td>32.1 ±1.1</td>
</tr>
<tr>
<td>1.0e-4</td>
<td>25.0</td>
<td>150.1 ±1.2</td>
<td>1.26 ±0.02</td>
<td>47.7 ±1.1</td>
</tr>
<tr>
<td>1.0e-4</td>
<td>25.1</td>
<td>200.1 ±1.2</td>
<td>2.55 ±0.03</td>
<td>63.9 ±1.1</td>
</tr>
<tr>
<td>1.0e-4</td>
<td>20.3</td>
<td>200.9 ±1.2</td>
<td>2.39 ±0.03</td>
<td>57.6 ±1.1</td>
</tr>
<tr>
<td>1.0e-4</td>
<td>20.5</td>
<td>201.0 ±1.2</td>
<td>2.39 ±0.03</td>
<td>58.3 ±1.1</td>
</tr>
<tr>
<td>3.0e-4</td>
<td>20.3</td>
<td>245.7 ±1.2</td>
<td>4.21 ±0.05</td>
<td>69.0 ±1.1</td>
</tr>
<tr>
<td>1.0e-4</td>
<td>24.3</td>
<td>250.1 ±1.2</td>
<td>4.66 ±0.06</td>
<td>84.6 ±1.1</td>
</tr>
<tr>
<td>1.0e-4</td>
<td>23.1</td>
<td>299.9 ±1.2</td>
<td>7.81 ±0.09</td>
<td>107.0 ±1.1</td>
</tr>
<tr>
<td>1.0e-4</td>
<td>23.5</td>
<td>300.1 ±1.2</td>
<td>7.69 ±0.09</td>
<td>107.9 ±1.1</td>
</tr>
<tr>
<td>3.0e-3</td>
<td>22.6</td>
<td>300.2 ±1.2</td>
<td>7.34 ±0.08</td>
<td>103.6 ±1.1</td>
</tr>
<tr>
<td>1.0e-4</td>
<td>21.6</td>
<td>346.6 ±1.3</td>
<td>11.99 ±0.13</td>
<td>136.1 ±1.2</td>
</tr>
<tr>
<td>6.8e-3</td>
<td>21.5</td>
<td>349.8 ±1.3</td>
<td>11.98 ±0.13</td>
<td>132.7 ±1.2</td>
</tr>
<tr>
<td>1.0e-4</td>
<td>22.7</td>
<td>350.1 ±1.3</td>
<td>12.10 ±0.13</td>
<td>136.9 ±1.2</td>
</tr>
<tr>
<td>5.4e-3</td>
<td>23.2</td>
<td>398.3 ±1.5</td>
<td>17.59 ±0.19</td>
<td>164.8 ±1.2</td>
</tr>
<tr>
<td>1.4e-3</td>
<td>20.6</td>
<td>398.7 ±1.5</td>
<td>17.57 ±0.19</td>
<td>164.8 ±1.2</td>
</tr>
</tbody>
</table>
In Table 5.7 the results of the preliminary off-sun tests are summarized, showing all relevant pressure, temperature, and heat loss values, together with the measurement uncertainty, described in the next section. The results are also visualized in Figure 5.7 and Figure 5.8 showing heat loss and glass temperature as a function of the average absorber tube temperature, respectively. The error bars are within the size of the circular markers. The simulation results in Section 5.3.1 have shown that low annulus pressures do not have a significant effect on the heat transfer in the receiver. Therefore, all measurement results shown in Table 5.7 are plotted as one data set in Figure 5.7 and Figure 5.8. For visual reasons a curve is fitted to the data illustrating the general trend of the results (solid line). In addition, simulated values of heat loss and glass temperature are shown from the modeling study of Section 5.3.1 for the case of air at 0.1 Pa and the optimum candidate coating S14 (dashed line). As expected, $\dot{Q}_{\text{loss}}$ and with it $T_5$ are significantly higher in the off-sun tests than what the

![Figure 5.7: Heat loss as a function of the absorber tube temperature from the preliminary off-sun tests using the PSC (circle). A trend line is fitted through the data points (solid line). The dashed line shows a simulation using the candidate coating S14.](image)
Figure 5.8: Glass temperature as a function of the absorber tube temperature from the preliminary off-sun tests using the PSC (circle). A trend line is fitted through the data points (solid line). The dashed line shows a simulation using the candidate coating S14.

modeling results suggest for an optimum selective coating, due to the higher overall emittance of the PSC. While $\dot{Q}_{\text{loss}}$ follows a smooth trend when $T_3$ is increased, as seen by the fitted curve, the data points for $T_5$ are scattered more. Although no measurements at temperatures above 400°C and pressures above 1e-3 mbar were possible in these tests, the results confirm that the off-sun test setup is well suited to performing detailed and extensive heat loss measurements with high accuracy, as suggested by the following uncertainty analysis.

5.3.5 Uncertainty Analysis

The uncertainty analysis of the experimental measurement data is conducted using the root-sum-square uncertainty model [45, 81]. The total uncertainty $U_\Phi$ for a function $\Phi$ is calculated from the uncertainties associated with the
systematic measurement bias (induced by the measurement devices) and the random precision error (caused by the signal noise),

\[
U_\Phi = \sqrt{u_{\Phi,\text{bias}}^2 + 4 \cdot u_{\Phi,\text{prec}}^2}
\]  

(5.7)

The uncertainties \( u_\Phi \) are determined from the root-sum-squares of the partial derivatives of \( \Phi \) with respect to each variable \( (x_i) \) and the error associated with the variable \( (\Delta x_i) \),

\[
u_{\Phi}^2 = \sum_i \left( \frac{\partial \Phi}{\partial x_i} \right)^2 \Delta x_i^2
\]  

(5.8)

Equation (5.8) is evaluated for both bias and precision errors using different error terms \( \Delta x_i \). The precision errors are determined with the standard deviation of the data points used for the calculation of the steady-state temperatures and voltages, and the number of measurement points,

\[
\Delta x_{i,\text{prec}} = \frac{\sigma_x}{\sqrt{n_x}}
\]  

(5.9)

Estimates of the measurement error are used to calculate the bias associated with the temperature and voltage measurements. The K-type thermocouples have an accuracy of ±1.1°C or ±0.4% of the reading (in °C), whichever is higher\(^3\). The module of the DAQ system\(^4\) that is used to record the temperature readings has an accuracy between ±1.2°C and ±1.3°C for temperatures between 20°C and 400°C. Assuming that all errors are independent and normally distributed, the measurement bias of a single temperature reading \( \Delta T_{\text{bias}} \) is between ±1.6°C (at 20°C) and ±1.8°C (at 400°C). The module of the DAQ system\(^5\) that measures the voltage drop \( V_{el} \) across the receiver has an accuracy of ±0.13% of the reading plus ±0.031 V. The measurement bias \( \Delta V_{el,\text{bias}} \) is hence between ±0.032 V at \( V_{el} = 1 \) V and ±0.064 V at \( V_{el} = 25 \) V. According to the manufacturer, the Hall-effect sensor in the impedance heating system that

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measures the electrical current through the absorber tube ($I_{\text{el}}$) has a response of 4 mV/A and an accuracy of around 1%. The resulting voltage is recorded by a module in the DAQ system\textsuperscript{5} that has an accuracy of $\pm 0.13\%$ of the reading plus $\pm 0.0063$ V. Assuming that all errors are independent and normally distributed the measurement bias for an electric current of 1000 A is $\Delta I_{\text{el,bias}} = \pm 10.41$ A.

In the analysis, the uncertainty associated with the average absorber tube temperature, the average glass envelope temperature, and the heat loss from the absorber tube is determined. Using Eq. (5.5) and (5.8), the uncertainty associated with the average absorber temperature is,

$$u_{T_3}^2 = \left(\frac{\partial T_3}{\partial T_{3,B}}\right)^2 \Delta T_{3,B}^2 + \left(\frac{\partial T_3}{\partial T_{3,C}}\right)^2 \Delta T_{3,C}^2 = \frac{\Delta T_{3,B}^2 + \Delta T_{3,C}^2}{4} \quad (5.10)$$

The uncertainty for the average glass temperature is calculated identically. The uncertainty associated with the heat loss from the absorber tube,

$$u_{\dot{Q}_{\text{loss}}}^2 = \left(\frac{\partial \dot{Q}_{\text{loss}}}{\partial I_{\text{el}}}\right)^2 \Delta I_{\text{el}}^2 + \left(\frac{\partial \dot{Q}_{\text{loss}}}{\partial V_{\text{el}}}\right)^2 \Delta V_{\text{el}}^2 + \sum_i \left(\frac{\partial \dot{Q}_{\text{loss}}}{\partial T_{3,i}}\right)^2 \Delta T_{3,i}^2 + \left(\frac{\partial \dot{Q}_{\text{loss}}}{\partial l_{\text{cond}}}\right)^2 \Delta l_{\text{cond}}^2 \quad (5.11)$$

is determined using Eq (5.6),

$$u_{\dot{Q}_{\text{loss}}}^2 = V_{\text{el}}^2 \Delta I_{\text{el}}^2 + I_{\text{el}}^2 \Delta V_{\text{el}}^2 + \frac{k_3^2 A_{23}^2}{l_{\text{cond}}^2} \left(\Delta T_{3,A}^2 + \Delta T_{3,B}^2 + \Delta T_{3,C}^2 + \Delta T_{3,D}^2\right) \quad (5.12)$$

The total uncertainty calculated according to Eq. (5.7) is shown in Table 5.7 for $T_3$, $T_5$, and $\dot{Q}_{\text{loss}}$. It can be seen that the uncertainty associated with the temperature measurements is on the order of 1°C, which corresponds to a relative uncertainty between 0.4% and 3.4%, depending on the actual value of the reading. For the heat loss, the uncertainty remains more or less constant with relative values between 1% and 2% for heat losses between 0.5 kW and 17.5 kW. Overall, the uncertainty is very low for all measurement parameters, in part due to the use of the impedance heating system that allows for an exact assessment of the heating power with minimal end effects.
5.3.6 Model Validation

The spectral reflectivity data of the PSC is used to simulate the preliminary heat loss tests using the detailed heat transfer model. This represents an opportunity to compare experimental data with simulation results using spectral optical data for the coating used in the tests, unlike the previous off-sun validation study (see Section 3.3), where the spectral data of the actual receiver was not available. In addition, the high overall emittance of the PSC allows for comparing heat transfer values that are in a range that is not typically encountered in optimized PTC systems. Although no experimental data is available for temperatures above 400°C and for annulus pressures above 1e-3 mbar, the following comparison of this additional set of data still increases the confidence in the accuracy of the heat transfer model.

The experimental data of the preliminary off-sun tests and the results from simulations using the spectral data of the PSC and the design parameters outlined in Section 5.3.1 are shown in Figure 5.9 and Figure 5.10. The error

![Figure 5.9: Parity plot of heat loss values predicted by the heat transfer model versus heat loss measured in the off-sun tests, both using the PSC.](image-url)
bars for the experimental values again lie within the marker size. Figure 5.9 shows the agreement between the simulated and measured heat loss from the absorber tube when assuming a spectrally-dependent, semi-transparent envelope and a gray, opaque glass. As with the previous off-sun validation study, the optical properties of the glass described in Section 3.3 are used for the spectral and the gray case. From Figure 5.9, it can be seen that the agreement between model and experiments is very good for lower absorber temperatures. At higher temperatures, the heat transfer model under-estimates the heat losses. This may be due to a possible deterioration of the selective coating at higher temperatures, which further increases the emissivity, or higher conduction losses through the supporting structure that are under-estimated in the heat transfer model. At higher temperatures, the assumption of an opaque glass leads to even lower predicted heat losses, as the lack of radiation transmitted through the glass reduces the overall heat losses.
Similar to the previous off-sun validation study, this insulation effect of the opaque glass can also be seen in the glass temperature, plotted in Figure 5.10. The agreement between model and experiment is generally very good for all temperatures and both modeling approaches. However, the lack of radiation transmitted through the opaque glass in the gray case increases the glass temperatures by about 4°C. In general, the spectral approach to the glass envelope produces results that agree better with the experimental data than the gray, opaque assumption. The RMS error of the heat loss values is 0.75 W for the spectral case and 0.80 W for the gray case, while the RMS error of the glass temperatures is 3.2°C and 3.8°C, respectively.

5.4 Conclusions

In the framework of the HITECO project, several candidates of a new selective coating were developed. The detailed 3D heat transfer model was used to assess the influence of each coating on the overall performance of a generic PTC solar field in terms of the coating’s measured optical properties. The analysis of the optical and thermal performance of the new coatings yielded thermal efficiencies that are 0.5% higher than the results achieved with comparable existing selective coatings. Based on these simulations, the best-performing coating was selected for the HITECO off-sun receiver prototype. The model was further used to describe the behavior of the off-sun test bench by analyzing the performance of the receiver prototype including the new selective coating. The results assisted in determining the specifications of the test equipment, such as the power requirements of the heating system or the placement of thermocouples. Heat losses and glass temperatures in the HITECO prototype are expected to be lower than for existing PTC receivers, with maximum heat losses of around 25 kW reached at 600°C with hydrogen at 100 Pa present in the annulus. Once assembled and calibrated, the test bench will allow measuring the heat loss at temperatures of up to 600°C and at pressures ranging from 0.01 Pa up to 100 Pa. Using the refilling feature, the receiver performance can be
analyzed with nitrogen or krypton present in the annulus at various pressures. Furthermore, the data collected may be compared to the predictions made by the heat transfer model presented in Section 5.3.1. In addition to the heat loss, the temperature distribution on various receiver components, such as the glass envelope or the bridle seals calculated by the heat transfer model can be compared to the values measured by the thermocouples in the off-sun tests. A first set of off-sun tests was conducted using a preliminary version of the new selective coating with non-optimized optical properties. The tests show that the manufacturing and testing processes are feasible to industrially produce and evaluate solar selective coatings. The heat transfer model is used to simulate these preliminary tests by utilizing spectral data of the PSC. The calculated heat losses and surface temperatures agree well with experimental data. The assumption of a semi-transparent glass envelope allows for more accurate results than when an opaque glass is assumed. Once the optimized selective coating is available, the actual off-sun tests can commence and higher temperatures and annulus pressures will be possible. By comparing the simulated and measured results, the heat transfer model can be further validated at these conditions.
6 Optimization of PTC Systems

6.1 Introduction

Even though PTC systems are currently the most mature and cost-effective CSP technology, there is still room for improvement. To further increase the efficiency of these systems and reduce the cost of electricity production, all system components need to be continuously optimized. As shown in the previous chapters, the main components of a PTC system include the absorber tube, the protective glass envelope, the reflective primary concentrator mirror, and the supporting structure that connects and holds the receiver. Determining the effect of the different components on the system performance is required to assess the most fruitful direction of development and optimization.

Previous studies have focused on the effect of the receiver geometry on the thermal efficiency of the PTC system by varying the absorber tube or glass envelope diameters [17, 19, 56, 82]. Other studies analyze the effect of various parameters of the primary concentrator mirror, such as the rim angle, aperture width, mirror reflectivity, slope error, tracking error, or mirror misalignment [17, 52, 56, 83-85]. To a lesser extent, analyses have investigated the effect of operating conditions, such as the DNI, the solar incidence angle, or the mass flow of the HTF [17, 82]. In most studies, several parameters are analyzed individually, but rarely are parameters varied simultaneously or are idealizations implemented to quantify the overall improvement potential of a certain component. In the reviewed literature, no studies have addressed the effect of varying the optical properties of the selective coating or the glass envelope or investigated the impact of the supporting structure.

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A different method to boost the overall performance of CSP systems is to add a secondary mirror to increase the concentration at the receiver. For PTC systems a number of solutions have been proposed. Simple designs include reflective surfaces on insulation material in the vacuum annulus [82] or shaped reflective surfaces behind the absorber tube [86-88]. A secondary concentrator that is commonly implemented in point and line focus systems is the compound parabolic concentrator (CPC) [89, 90]. Besides being widely used for point focus systems, such as CRS, different designs have been proposed for line focus systems, such as PTCs and Fresnel concentrators [85, 91, 92]. Another class of secondaries is the hyperbolic concentrator, also known as the “trumpet” [90]. Several designs have been presented, including for line focus systems [83, 92, 93]. CPCs and trumpets generally require lower primary mirror rim angles to achieve the best optical performance. Alternative designs have been proposed for larger rim angles, such as multiple truncated CPCs [94, 95]. In addition, several aplanatic systems have been presented [96-100], for which the primary and secondary mirrors can be described by a parameterization [101]. Different configurations have been proposed, mainly for point focus systems in concentrated photovoltaic (CPV) applications, where flat absorbers are used.

Besides analytical descriptions of the geometries of the previously mentioned secondary designs, tailoring methods using the edge-ray principle or “string” method are implemented to construct secondary mirrors [89, 90]. A wide variety of designs for secondary optics has been proposed, which are tailored using the edge-ray principle. One such design is a secondary shape that approximates a cone or V-shape concentrator [102, 103]. Another tailored secondary concentrator that can be used with a conventional PTC primary is the so-called “seagull”, named after the reflector’s resemblance to a flying seabird [104, 105]. This design consists of an analytically derived involute and a tailored wing. While the edge-ray principle presumes a given primary concentrator shape, both the primary and secondary geometries can also be tailored at the same time, using the simultaneous multiple surface (SMS)
method [89, 90]. Designs that are tailored using SMS are the proposed “snail” and “helmet” designs, which also allow a sizeable gap between the secondary mirror and the absorber [106]. Recently, a SMS-tailored secondary for conventional PTC tubular receivers was proposed, using a primary shape that is approximately parabolic [107]. Other tailored two-mirror compounds include a concentrator system using a dielectric material [108], a tailored secondary with a circular primary [84], or a tailored parabolic-hyperbolic concentrator [109]. Over the years, a number of unconventional concentrator designs have also been proposed. For the sake of completeness, three designs are mentioned here, including a design with an array of CPCs [110], a design with a stationary primary and a tracking secondary concentrator [111], and a tracking secondary system in an inflatable PTC for CPV applications [112].

Several problems are commonly encountered when the secondary mirror systems described above are applied to PTCs. For example, when smaller rim angles of the primary mirror are required, problems with tracking and mechanical stability occur [85, 94, 106]. In addition, some secondary-mirror designs are rather complex [92, 94, 110], which may result in high manufacturing and assembly costs. Several secondary mirrors lead to increased heat losses by touching the hot absorber tube, while others leave an intentional gap between the mirror and the absorber and suffer from higher optical losses [94, 96, 106, 107]. From a modeling point of view, the proposed secondary designs are analyzed mostly in terms of their optical performance alone. A combined optical and thermal analysis of a CSP system that incorporates secondary mirrors is missing from the literature.

The objectives of this chapter are to assess the potential improvements of the efficiency of PTC systems by component idealizations and secondary optics, and to identify the most promising avenues for further research and development. Accordingly, the focus of this study is not on the absolute performance of PTC systems, but on the differences in and the sensitivity of the optical and thermal behavior relative to a generic PTC system due to changes in
its components. In contrast to previous investigations, this study is based on a combined 3D optical and thermal analysis. The combined analysis is necessary because the optical performance of a PTC system has a strong impact on its thermal performance. The approach adopted in this study consists of two steps. In the first step, the optimization potentials of major PTC system components, namely the selective coating, the glass envelope, the primary concentrator mirror, and the supporting structure, are assessed relative to a selected benchmark PTC system design. The components are idealized step-by-step to evaluate the associated potential benefit. Starting from realistic values for the components, the idealizations allow the sensitivity and therefore the improvement potential of the optical and thermal behavior to be quantified. In the second step, selected secondary-mirror designs are subjected to the same detailed 3D optical and thermal analysis. The designs are selected to be compatible with conventional PTC systems, which require a primary concentrator rim angle of at least 80° and a tubular receiver, as well as overall simplicity to minimize manufacturing and assembly costs. Among the selected designs are a simple reflective glass surface, annulus insulation with reflective surfaces, an aplanatic two-mirror configuration, and the tailored “seagull” secondary. The secondary-mirror designs are optimized for maximum thermal efficiency and compared to the benchmark design. This detailed study enables an evaluation of secondary mirrors from a thermal point of view, extending previously published assessments on the economic and optical value of these concepts.

### 6.2 Benchmark Design

The assessment of potential improvements starts with the typical high-temperature PTC system described in Section 4.2.1 and Table 4.1. Identical to Section 5.2.2, for all simulations in this chapter a solar incidence angle of $\theta = 30.1^\circ$ and a DNI of $I_{\text{DNI}} = 619 \text{ W/m}^2$ is used, operating conditions that produce representative efficiencies for a PTC system located in Seville, Spain. At this
stage, the optimization does not yet consider any changes in material properties or the addition of secondary optics. Instead, the optimization considers only the absorber tube diameter $D_3$ and the concentrator rim angle $\psi_{\text{conc}}$. The concentrator aperture width is not changed to keep the theoretical solar input at the system aperture constant. Figure 6.1 shows a fitted contour plot of the thermal efficiency as a function of the two parameters, obtained from the simulation results. The contour plot is obtained from a 4th-order polynomial least-squares fit to results from 54 simulations with the detailed 3D heat transfer model. (The root mean square difference between the fit and the results from the heat transfer model is around 0.02 percentage points.) In the figure, the global maximum of $\eta_{\text{th}} = 60.0\%$ is indicated by the circle. Values of the thermal efficiency, together with the optical efficiency and the heat loss from the absorber tube per unit length of the receiver, are listed in Table 6.1.

Figure 6.1: Contour plot of thermal efficiency as a function of absorber tube diameter and concentrator rim angle fitted to a total of 54 simulation points (scatter dots). The optimum configuration with $D_3 = 0.058$ m and $\psi_{\text{conc}} = 112^\circ$ is indicated by the circle.
Table 6.1: Results of a typical high-temperature PTC system, the optimum configuration, and the selected benchmark design.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>$D_3$ [m]</th>
<th>$\psi_{\text{conc}}$ [°]</th>
<th>$\eta_{\text{opt}}$ [%]</th>
<th>$\dot{q'}_{\text{loss}}$ [W/m]</th>
<th>$\eta_{\text{th}}$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Typical PTC system</td>
<td>0.070</td>
<td>80</td>
<td>75.1</td>
<td>268</td>
<td>58.6</td>
</tr>
<tr>
<td>Optimum design</td>
<td>0.058</td>
<td>112</td>
<td>66.0</td>
<td>219</td>
<td>60.0</td>
</tr>
<tr>
<td>Benchmark design</td>
<td>0.060</td>
<td>80</td>
<td>65.0</td>
<td>229</td>
<td>58.8</td>
</tr>
</tbody>
</table>

Figure 6.2: Main factors impacting thermal efficiency, including optical efficiency, heat losses, and the required pumping power, plotted as a function of $D_3$ for $\psi_{\text{conc}} = 110^\circ$. Heat losses and pumping power are shown as a fraction of the solar input identical to how the optical and thermal efficiencies are defined.

The relative contribution of the factors that affect the thermal efficiency are plotted in Figure 6.2 as a function of $D_3$ for $\psi_{\text{conc}} = 110^\circ$. For commonly used absorber tube diameters (e.g., $D_3 = 0.07$ m), the pumping losses are negligible. For smaller diameters, however, the pumping losses increase significantly, reaching values of 0.3% of the solar input $I_{\text{DNI}} A_{\text{conc}}$ at $D_3 = 0.04$ m. With larger
absorber diameters the thermal losses increase mostly due to the larger surface area of the absorber tube, accounting for 4.2-8.4% of the solar input. The optical losses increase with smaller absorber sizes and represent 32.7% to 37.5% of the solar input. Similar trends and relative contributions are observed for these parameters at concentrator rim angles other than \( \psi_{\text{conc}} = 110^\circ \). The thermal efficiency is observed to exhibit a local maximum because of the opposite trends of the optical and heat losses.

From Figure 6.1, it can be seen that the maximum efficiency is achieved at rim angles around 110°, depending on the chosen absorber diameter. For larger absorber sizes, the optimum rim angle increases, reaching a global maximum at \( \psi_{\text{conc}} = 112^\circ \) and \( D_3 = 0.058 \) m. This optimal receiver is shown in Figure 6.3, together with selected rays from the MC ray tracing. As the absorber diameter is not selected based on a specific sun angle, the mean distance of the receiver from the primary mirror has a stronger effect on the final concentration. In turn, this has a significant effect on the optical efficiency of the receiver. Therefore, the optimum for the rim angle moves from 90°, where the highest theoretical concentration at 100% interception occurs, towards 120°, where the smallest mean distance between receiver and primary concentrator is attained.

Figure 6.3: Visualization of the optimum PTC configuration at \( D_3 = 0.058 \) m and \( \psi_{\text{conc}} = 112^\circ \), together with selected rays generated by MC ray tracing. FL denotes the focal line.
However, rim angles above 90° have several mechanical and economic disadvantages. A mirror with a rim angle of 110° has a glass surface that is over 15% larger compared to a typical concentrator mirror with a rim angle of 80°, increasing material cost and requiring a heavier supporting structure. The larger mirror surface also causes greater wind loads and the associated deformation of the mirror surface may negate the improved optical performance. In the following analyses, a rim angle of 80° is used as baseline since it combines good optical performance and a compact mirror-receiver configuration. Since it is also a configuration widely used today, a receiver optimized for this rim angle can be combined with a commercially available mirror. The optimum absorber diameter for this baseline rim angle of 80° is around 0.06 m. This absorber size produces near-optimum efficiencies for virtually all investigated rim angles, as the optimum range is very broad (see Figure 6.1). Therefore, these geometric characteristics define what is called the benchmark design in this chapter. The benchmark design is identical to the typical PTC system described in Section 4.2.1 and Table 4.1, with the exception of $D_3 = 0.06$ m and $D_2 = 0.054$ m. The thermal efficiency of the benchmark design is $\eta_{th} = 58.8\%$. All subsequent analyses in this study will be compared to this benchmark design. In the following, the optimization potential of the benchmark design is assessed by idealizing selected components. In Section 6.3, the absorber tube diameter is kept constant at $D_3 = 0.06$ m, whereas in Section 6.4, the absorber size is varied (while keeping the wall thickness constant) as part of the optimization analysis.

6.3 Optimization Potential of Idealized PTC Components

6.3.1 Absorber Tube Selective Coating

The absorber tube selective coating ensures high solar absorption and low re-radiation losses. To assess the optimization potential of the selective coating, the spectral behavior can be idealized. For this purpose, a sigmoid function is fitted to the spectral reflectivity data of the ENEA coating. For the function
the coefficients are determined to be $a_0 = 0.992$, $b_0 = 0.0105$, $c_0 = 9.302 \, \mu m$, and $d_0 = -4.559$. Figure 6.4 shows the spectral reflectivity curve of the ENEA coating (solid line) and the fitted sigmoid curve (dashed line). Simulations with the benchmark design, using the fitted sigmoid curve to calculate the spectral emissivity of the absorber tube, yielded results very close to those obtained using the actual spectral data, as shown in Table 6.2. The sigmoid curve can now be used to easily implement idealizations. Perfect solar absorption is attained by setting $a_0 = 1.003$ and $b_0 = 0$, yielding $\rho_3 = 0$ for the wavelength limit below the inflection point, here referred to as the cut-off. Setting $a_0 = 1.006$ and $b_0 = 0.0105$ produces a sigmoid curve with ideal reflectivity at higher wavelengths, resulting in $\rho_3 = 1$ for wavelengths above the cut-off. Finally, using $a_0 = 1.017$ and $b_0 = 0$ combines both effects, yielding a sigmoid curve with $\rho_3 = 0$ below and $\rho_3 = 1$ above the cut-off. The improvements in heat loss and thermal

$$\rho_3(\lambda) = \frac{a_0}{1 + \exp\left(c_0/\lambda + d_0\right)} + b_0$$

(6.1)
Table 6.2: Simulation results for different spectral data of the selective coating.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>( \eta_{\text{opt}} )</th>
<th>( q'_{\text{loss}} )</th>
<th>( \eta_{\text{th}} )</th>
<th>( \Delta \eta_{\text{th}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Benchmark design (“ENEA”)</td>
<td>65.0</td>
<td>229</td>
<td>58.8</td>
<td>-</td>
</tr>
<tr>
<td>Sigmoid fit</td>
<td>65.0</td>
<td>228</td>
<td>58.8</td>
<td>0.0</td>
</tr>
<tr>
<td>Sigmoid fit; no solar reflection</td>
<td>65.7</td>
<td>230</td>
<td>59.4</td>
<td>0.6</td>
</tr>
<tr>
<td>Sigmoid fit; no IR emission</td>
<td>65.0</td>
<td>195</td>
<td>59.6</td>
<td>0.8</td>
</tr>
<tr>
<td>Sigmoid fit; no solar refl./IR emiss.</td>
<td>65.6</td>
<td>197</td>
<td>60.2</td>
<td>1.4</td>
</tr>
<tr>
<td>Sigmoid A; ( c = 10 , \mu m, \lambda_{c,3} = 1.7 , \mu m )</td>
<td>63.8</td>
<td>99</td>
<td>61.1</td>
<td>2.3</td>
</tr>
<tr>
<td>Sigmoid B; ( c = 1000 , \mu m, \lambda_{c,3} = 2.3 , \mu m )</td>
<td>68.1</td>
<td>82</td>
<td>65.8</td>
<td>7.0</td>
</tr>
</tbody>
</table>

Efficiency for each of these idealization steps are listed in Table 6.2. Starting from a realistic spectral behavior, such as the sigmoid fit of the ENEA coating, assuming perfect absorption characteristics in the solar spectrum allows for improvements in thermal efficiency of 0.6% due to improved optical efficiency. An increase of around 0.8%, achieved through lower heat losses, is possible with a coating that has zero thermal emission at higher wavelengths. Consequently, a coating displaying both ideal behaviors allows for an increase in the thermal efficiency of 1.4%.

In addition to idealizing the levels of reflectivity at both ends of the wavelength spectrum, the actual shape of the spectral reflectivity curve can also be optimized. For this purpose, a simpler sigmoid function is used, assuming ideal behavior in the solar \( (\rho_3 = 0) \) and IR \( (\rho_3 = 1) \) parts of the spectrum,

\[
\rho_3(\lambda) = \frac{1}{1 + \exp\left(\frac{c}{\lambda} - \frac{c}{\lambda_{c,3}}\right)}
\]  

(6.2)

where \( c \) is a coefficient determining the shape of the curve and \( \lambda_{c,3} \) is the cut-off wavelength. The higher \( c \), the more the reflectivity curve approaches a step function. Figure 6.4 shows the spectral reflectivity of two examples of such simplified sigmoid curves. The curve with coefficients \( c = 10 \, \mu m, \lambda_{c,3} = 1.7 \, \mu m \) (“Sigmoid A”) is a curve similar to the sigmoid fit of the ENEA coating, while the example with \( c = 1000 \, \mu m, \lambda_{c,3} = 2.3 \, \mu m \) (“Sigmoid B”) approximates a
step function with a cut-off at $\lambda_{c,3}$. Results of simulations performed with various simplified sigmoid curves are presented in the following figures. Selected results of configurations are summarized in the last two rows of Table 6.2. In Figure 6.5, heat loss per unit length of the solar field loop is shown as a function of the cut-off wavelength $\lambda_{c,3}$ for several values of the coefficient $c$. In subsequent plots, a spline is fit through the data points. As the cut-off wavelength increases, the heat loss increases because the total emissivity becomes larger. When $c$ is increased, the sigmoid curve approaches a step function, resulting in lower total emissivities for a specific cut-off wavelength. The cut-off wavelength also has a significant effect on the optical efficiency, as can be seen from Figure 6.6. The optical efficiency increases for larger $\lambda_{c,3}$ because of greater total solar-weighted absorptance. With increased $c$, however, a break in the continuous rise of the optical efficiency is observed between cut-off wavelengths of about 1.8 to 2.0 $\mu$m. This break stems from the drop in solar irradiance $I$, for this range of wavelengths, as can be seen from the ASTM solar spectrum that is also shown in Figure 6.6. Between wavelengths of around 2.0
Figure 6.6: Optical efficiency as a function of the cut-off wavelength for different sigmoid curve coefficients, together with the ASTM solar spectrum (right axis).

Figure 6.7: Thermal efficiency as a function of the cut-off wavelength for different sigmoid curve coefficients.
and 2.5 μm, the optical efficiency increases again. Above 2.5 μm, the solar intensity becomes extremely small, and the optical efficiency reaches a maximum that stays virtually constant for all higher $\lambda_{c,3}$.

Because of the shape of the solar spectrum, the optical efficiency is relatively insensitive to a cut-off wavelength in the range of 1.8 to 1.9 μm, provided that the reflectivity curve approaches a step function. However, the thermal efficiency is affected since the heat losses increase steadily with increasing cut-off wavelength, see Figure 6.5. This may be seen from Figure 6.7, where the thermal efficiency is plotted as a function of the cut-off wavelength. For higher $c$, the efficiency drops in the spectral band between 1.8 and 2.0 μm due to the reduced total solar input. Because of the increased heat losses in this part of the spectrum, the maximum thermal efficiency is reached at cut-off wavelengths that are below those determined for the best optical efficiency. To illustrate this point, the optimum cut-off for different sigmoid coefficients $c$ is plotted in Figure 6.8. For $c = 1000$ μm, the optimum cut-off is around $\lambda_{c,3} = 2.3$ μm, at which a thermal efficiency of $\eta_{th} = 65.8\%$ is reached.

![Figure 6.8: Optimum cut-off wavelength as a function of the coefficient $c$.](image-url)
(the highest of any of the configurations evaluated here). For smaller $c$, the optimum $\lambda_{c,3}$ decreases significantly. The optimum cut-off for a more realistic approximation of the reflectivity, such as a sigmoid function with $c = 10 \, \mu m$, is located at a wavelength of around $1.7 \, \mu m$. As a result, the optimum cut-off wavelength of a selective coating varies significantly, depending on the shape of the spectral reflectivity curve. In conclusion, an ideal selective coating, displaying a spectral step function behavior with a cut-off wavelength of $\lambda_{c,3} = 2.3 \, \mu m$ and reflectivity values of $\rho_3 = 0$ for $\lambda < \lambda_{c,3}$ and $\rho_3 = 1$ for $\lambda > \lambda_{c,3}$, has an improvement potential of around 7% compared to the benchmark design that uses an existing selective coating.

6.3.2 Glass Envelope

The optimization potential of the glass envelope is determined by solely changing the values below and above the cut-off wavelength $\lambda_{c,45} = 2.6 \, \mu m$, see Section 3.3. Three idealizations are assumed that are analyzed both individually and in combination. Ideal behavior in the solar spectrum is achieved by setting $\rho_{45} = 0$ and/or $\alpha_{45} = 0$ for $\lambda \leq \lambda_{c,45}$. This measure primarily increases the optical efficiency. In the IR spectrum, i.e., above the cut-off, ideal behavior is achieved by assuming zero emission from and zero absorption by the glass, reducing the radiative heat losses from the absorber tube. Table 6.3 shows the results of the corresponding simulations. It can be seen that compared to the benchmark design, the largest improvements for a single measure is achieved by setting the IR absorption to zero, significantly reducing the heat losses from the absorber tube. Therefore, an ideal selectively reflective coating in the IR spectrum could improve the thermal efficiency by around 4.2%. Comparable improvements in the solar spectrum can only be achieved by combining the effects of a perfect AR coating and perfect transmittance behavior, for example by using a very thin envelope. By removing both reflection and absorption losses from the glass in the solar spectrum, an increase in optical efficiency, and consequently also in the thermal efficiency, of 3% with respect to the benchmark design can be
Table 6.3: Simulation results for various configurations based on idealizing the optical properties of the glass envelope. The notation “x / y” indicates that the optical property is equal to x below the cut-off and y above it.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>$\rho_{45}$ [%]</th>
<th>$\tau_{45}$ [%]</th>
<th>$\alpha_{45}$ [%]</th>
<th>$\eta_{\text{opt}}$ [%]</th>
<th>$q'_{\text{loss}}$ [W/m]</th>
<th>$\eta_{\text{th}}$ [%]</th>
<th>$\Delta\eta_{\text{th}}$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Benchmark design</td>
<td>1.5 / 11</td>
<td>96.5 / 0</td>
<td>2 / 89</td>
<td>65.0</td>
<td>229</td>
<td>58.8</td>
<td>-</td>
</tr>
<tr>
<td>No solar reflection</td>
<td>0 / 11</td>
<td>98 / 0</td>
<td>2 / 89</td>
<td>66.5</td>
<td>229</td>
<td>60.2</td>
<td>1.4</td>
</tr>
<tr>
<td>No solar absorption</td>
<td>1.5 / 11</td>
<td>98.5 / 0</td>
<td>0 / 89</td>
<td>66.5</td>
<td>229</td>
<td>60.2</td>
<td>1.4</td>
</tr>
<tr>
<td>No IR absorption</td>
<td>1.5 / 100</td>
<td>96.5 / 0</td>
<td>2 / 0</td>
<td>65.0</td>
<td>74</td>
<td>63.0</td>
<td>4.2</td>
</tr>
<tr>
<td>No solar refl./abs.</td>
<td>0 / 11</td>
<td>100 / 0</td>
<td>0 / 89</td>
<td>68.1</td>
<td>230</td>
<td>61.8</td>
<td>3.0</td>
</tr>
<tr>
<td>No solar refl./IR abs.</td>
<td>0 / 100</td>
<td>98 / 0</td>
<td>2 / 0</td>
<td>66.5</td>
<td>75</td>
<td>64.4</td>
<td>5.6</td>
</tr>
<tr>
<td>No solar/IR abs.</td>
<td>1.5 / 100</td>
<td>98.5 / 0</td>
<td>0 / 0</td>
<td>66.5</td>
<td>74</td>
<td>64.5</td>
<td>5.7</td>
</tr>
<tr>
<td>No solar/IR refl./abs.</td>
<td>0 / 100</td>
<td>100 / 0</td>
<td>0 / 0</td>
<td>68.2</td>
<td>75</td>
<td>66.1</td>
<td>7.3</td>
</tr>
</tbody>
</table>

achieved. A glass envelope with completely ideal optical properties and a cut-off wavelength of 2.6 $\mu$m has a total improvement potential of around 7.3% compared to the benchmark design.

6.3.3 Primary Concentrator Mirror

The optimization potential of the primary concentrator is determined by idealizing the optical and mechanical properties of the mirror. The parameters include the reflectivity, the angular reflection error, and the tracking error, see Section 2.3. The results of the corresponding simulations are summarized in Table 6.4. Increasing the reflectivity from the benchmark value of $\rho_{\text{conc}} = 0.92$ to 1.0 assumes a mirror with an ideal reflection material without any absorption losses. This measure linearly increases the solar input to the system, resulting in a linear increase in $\eta_{\text{th}}$. From Table 6.4, it can be seen that with this increase in reflectivity, a gain in thermal efficiency of around 5.5% is achieved, which is roughly equal to the increase in reflected solar power times the optical efficiency. The effects of the angular reflection error $e_{\text{conc}}$, simulating dispersion caused by a non-ideal mirror surface, are visualized in Figure 6.9 for the cases of $e_{\text{track}} = 0$ and 1 mrad and $\rho_{\text{conc}} = 0.92$. For the case of $e_{\text{track}} = 1$ mrad,
Table 6.4: Simulation results for different configurations assuming various idealizations of the primary concentrator parameters.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>$e_{\text{track}}$ [mrad]</th>
<th>$e_{\text{conc}}$ [mrad]</th>
<th>$\rho_{\text{conc}}$ [-]</th>
<th>$\eta_{\text{opt}}$ [%]</th>
<th>$\dot{q}^\prime_{\text{loss}}$ [W/m]</th>
<th>$\eta_{\text{th}}$ [%]</th>
<th>$\Delta\eta_{\text{th}}$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Benchmark design</td>
<td>1.0</td>
<td>4.0</td>
<td>0.92</td>
<td>65.0</td>
<td>229</td>
<td>58.8</td>
<td>-</td>
</tr>
<tr>
<td>No tracking error</td>
<td>0.0</td>
<td>4.0</td>
<td>0.92</td>
<td>65.2</td>
<td>229</td>
<td>58.9</td>
<td>0.1</td>
</tr>
<tr>
<td>No surface error</td>
<td>1.0</td>
<td>0.0</td>
<td>0.92</td>
<td>66.2</td>
<td>229</td>
<td>60.0</td>
<td>1.2</td>
</tr>
<tr>
<td>No mirror absorption</td>
<td>1.0</td>
<td>4.0</td>
<td>1.00</td>
<td>70.6</td>
<td>228</td>
<td>64.3</td>
<td>5.5</td>
</tr>
<tr>
<td>No track./surf.</td>
<td>0.0</td>
<td>0.0</td>
<td>0.92</td>
<td>66.2</td>
<td>229</td>
<td>60.0</td>
<td>1.2</td>
</tr>
<tr>
<td>No track./abs.</td>
<td>0.0</td>
<td>4.0</td>
<td>1.00</td>
<td>70.8</td>
<td>228</td>
<td>64.5</td>
<td>5.7</td>
</tr>
<tr>
<td>No surf./abs.</td>
<td>1.0</td>
<td>0.0</td>
<td>1.00</td>
<td>71.9</td>
<td>228</td>
<td>65.6</td>
<td>6.8</td>
</tr>
<tr>
<td>No track./surf./abs.</td>
<td>0.0</td>
<td>0.0</td>
<td>1.00</td>
<td>71.9</td>
<td>228</td>
<td>65.6</td>
<td>6.8</td>
</tr>
</tbody>
</table>

Figure 6.9: Thermal efficiency for $\rho_{\text{conc}} = 0.92$ as a function of the angular reflection error for two different tracking errors.
as the dispersion increases, the optical efficiency decreases, leading to a considerable drop in thermal efficiency for $e_{\text{conc}} > 2$ mrad. Assuming a perfect mirror surface without any surface error, an improvement in thermal efficiency of around 1.2% compared to the benchmark design can be achieved. When therefection error approaches zero, the optical efficiency reaches its maximum value of $\eta_{\text{opt}} = 65.2\%$, being unaffected by the tracking error, as virtually all the radiation is intercepted by the absorber (see Figure 6.9). The tracking error has an increasingly significant effect only for larger $e_{\text{conc}}$, causing a reduction of 0.1% in thermal efficiency for the benchmark design with $e_{\text{conc}} = 4$ mrad and $\rho_{\text{conc}} = 0.92$. While the mirror reflectivity has by far the largest individual impact on the thermal efficiency, combining all three idealizations leads to an improvement of around 6.8%.

6.3.4 Supporting Structure

Another source of losses is the shading of the absorber tube by the structural components of the system, such as the support arms or the end bellows of the receivers. Figure 6.10 shows a side view of the supporting structure as it is used in the MC ray tracing simulations (see Figure 2.3) and the heat transfer analysis, as well as the relevant geometrical parameters. Three measures are analyzed to determine the impact of reducing shading losses. In a first step, the number of HCE per concentrator module ($n_{\text{HCE}}$) is reduced. In the benchmark design, this value is $n_{\text{HCE}} = 3$, with $l_{\text{abs}} = 4.06$ m. In the optimization, $n_{\text{HCE}}$, and

![Figure 6.10: Schematic of the supporting structure.](image)
with it the number of support arms, is reduced to two and one, respectively. The length of the concentrator module \((l_{\text{conc}} = 12 \text{ m})\) and the distance between two modules \((d_{\text{conc}} = 0.18 \text{ m})\) are kept constant, resulting in an increase of \(l_{\text{abs}}\) to 6.09 m and 12.18 m, respectively. The length of the shaded end parts of the HCE \((d_{\text{sh}})\) is coupled to \(d_{\text{conc}}\) for practical reasons to fulfill \(d_{\text{conc}} = d_{\text{sh}}\) for all configurations. As a second measure, \(d_{\text{sh}}\) and \(d_{\text{conc}}\) are reduced from 0.18 m in the benchmark design to zero. The configurations with \(d_{\text{conc}} = d_{\text{sh}} = 0\) simulate an ideal HCE without any bellows or connection parts at the end, which shade the absorber tube and decrease the useful length of the tube. With this idealization, the receiver becomes one single absorber tube and glass envelope running along the entire length of the solar field, with one single concentrator mirror of the same length. Finally, simulations are performed with and without support arms, to determine the associated shading losses.

In Table 6.5, the results of selected simulations are shown that implemented these idealizations. The largest effect in terms of improved thermal efficiency is achieved when reducing the number of HCE per concentrator module, which requires longer receivers and fewer support arms. Reducing the number of HCE per module to two and one leads to increases in the thermal efficiency of 1.4% and 2.8%, respectively. The increases in thermal efficiency are achieved both through higher optical efficiency but also through lower heat losses. Assuming an ideal receiver whose entire surface is irradiated and that requires no end parts to connect the single HCE \((d_{\text{sh}} = 0)\), an additional improvement of 0.8% is

<table>
<thead>
<tr>
<th>Configuration</th>
<th>(n_{\text{HCE}})</th>
<th>(d_{\text{sh}})</th>
<th>(\eta_{\text{opt}})</th>
<th>(q'_{\text{loss}})</th>
<th>(\eta_{\text{th}})</th>
<th>(\Delta\eta_{\text{th}})</th>
</tr>
</thead>
<tbody>
<tr>
<td>Benchmark design</td>
<td>3</td>
<td>0.18</td>
<td>65.0</td>
<td>229</td>
<td>58.8</td>
<td>-</td>
</tr>
<tr>
<td>2 HCE per module</td>
<td>2</td>
<td>0.18</td>
<td>66.4</td>
<td>225</td>
<td>60.2</td>
<td>1.4</td>
</tr>
<tr>
<td>1 HCE per module</td>
<td>1</td>
<td>0.18</td>
<td>67.7</td>
<td>221</td>
<td>61.6</td>
<td>2.8</td>
</tr>
<tr>
<td>1 HCE, no spacing</td>
<td>1</td>
<td>0</td>
<td>68.4</td>
<td>221</td>
<td>62.4</td>
<td>3.6</td>
</tr>
<tr>
<td>1 HCE, no spacing/support</td>
<td>1</td>
<td>0</td>
<td>69.0</td>
<td>218</td>
<td>63.1</td>
<td>4.3</td>
</tr>
</tbody>
</table>
possible with respect to the case of using 1 HCE per module. The higher efficiency is achieved over a shorter loop length, as the distance between the concentrator modules is also reduced to \( d_{\text{conc}} = 0 \). A completely idealized structure system, where support arms are not required, raises the thermal efficiency by another 0.6\%, resulting in a total improvement of 4.3\% compared to the benchmark design.

6.4 Secondary Optics

Most improvements discussed in the previous section depend on material properties that may be difficult to attain in practice. Nevertheless, the results are useful in guiding and prioritizing research and development efforts. The use of secondary optics at the receiver is another option to increase the optical efficiency and reduce heat losses by adapting the geometry of the absorber tube. In the following, four types of secondary-mirror designs are analyzed both optically and thermally and optimized for maximum thermal efficiency. The designs include a reflective glass surface, annulus insulation material with reflective surfaces, an aplanatic two-mirror configuration, and a tailored “seagull” secondary concentrator. The secondary mirror surfaces described here are modeled as perfect specular reflectors with a reflectivity of \( \rho_{\text{sec}} = 0.95 \). Emission of radiation and convection from the secondary surface, as well as thermal conduction through the mirrors are neglected apart from the insulation material discussed in Section 6.4.2. The benchmark design is again used as a starting point for the implementation of secondary optics, but in contrast to the previous section, the absorber tube diameter is now considered to be variable in the optimization process.

6.4.1 Reflective Glass Surface

The first type of secondary optics to be investigated is a simple reflective glass surface (RGS) [88, 106]. This entails coating a portion of the inner surface of the glass envelope with a highly reflective material (\( \rho_{\text{sec}} = 0.95 \)). In
addition, both the absorber tube and the glass envelope are vertically displaced from the focal line (FL) of the primary mirror. The absorber is moved downwards to intercept radiation that would pass beneath the absorber. The envelope is moved upwards so that the radiation missing the top of the absorber is reflected back towards it by the RGS. The reflective surface also reduces heat losses by reflecting thermally emitted radiation from the hot absorber tube. Figure 6.11 shows a schematic of this RGS system, together with selected rays from the MC ray tracing simulation of the incident concentrated solar radiation. The main parameters affecting the performance of the RGS configuration are the absorber diameter $D_3$, the displacement of the absorber $z_{23}$, the displacement of the envelope $z_{45}$, and the length of the reflective surface specified by the angle $\beta_{45}$. An initial analysis showed that the effect of the envelope diameter is not that significant (with or without RGS), which is consistent with previous observations [17]. Therefore, the benchmark value $D_5 = 0.125$ m was used for all the simulations discussed below. Parameter studies yielded maximum thermal efficiency with $D_3 = 0.050$ m, $z_{23} = -10$ mm, $z_{45} = 14$ mm, and $\beta_{45} = 83^\circ$. The optimum configuration using an RGS with the above parameters is shown in Figure 6.11. Detailed results of the optimization of all secondary

![Figure 6.11: Visualization of a configuration using a reflective glass surface (RGS) on the inside of the envelope, together with selected rays and design parameters.](image-url)
optics concepts discussed in this chapter are summarized in Table 6.6. It can be seen that with an RGS a small decrease in optical efficiency occurs due to the smaller absorber. Also, the heat losses are reduced by about 24%, leading to an improvement in the thermal efficiency of 1% compared to the benchmark design. The reduction in heat loss is larger than that caused by the reduction in the absorber surface, so a portion of the thermally emitted radiation is reflected back to the absorber tube. In addition, it should be noted that the optical efficiency of a conventional system with $D_3 = 0.050$ m and without an RGS would be around 63.1%. Therefore, although the secondary mirror shades most of the receiver, the RGS reflects enough incident solar radiation to achieve an increase in $\eta_{\text{opt}}$. As a consequence, the effective concentration at the absorber tube, $C = \frac{\eta_{\text{opt}} w_{\text{conc}}}{\pi D_3}$, for the optimum RGS system is 24.7 compared to $C = 20.7$ for the benchmark design.

### 6.4.2 Reflective Annulus Insulation

The next configuration studied is the reflective annulus insulation (RAI), in which the upper half of the vacuum annulus is filled with an insulation material ($k_{\text{ins}} = 0.04$ W/mK). This eliminates radiation losses from the absorber tube in that area, but also increases the conductive heat transfer from the absorber to the glass envelope. In addition, the exposed surfaces of the insulation material in the vacuum annulus are assumed to be coated with a highly reflective mate-
rial ($\rho_{\text{sec}} = 0.95$) to allow a portion of the solar radiation that hits the insulation to be reflected back to the absorber tube. The configuration resembles previously presented designs [82, 113], although the configuration is modified in this study to allow for more parameters to be optimized. Similar to the RGS, the parameters varied in the optimization study of the RAI are $D_3$, $z_{23}$ and $z_{45}$, as well as the dimensions of the insulation, determined by the angles $\beta_{23}$ and $\beta_{45}$, see Figure 6.12.

The parameter studies revealed that the best performance is achieved when the RAI covers less than the entire upper half of the absorber tube, as is the case in previously proposed designs [82, 113]. The lower optical efficiency due to the blocked incident solar radiation of such a system could not be compensated by the lower heat losses that would occur as more of the hot absorber tube surface is covered by the insulation material. Instead, the insulation is best limited to the uppermost part of the absorber tube where only little concentrated radiation is incident. The optimum configuration is shown in Figure 6.12 and is achieved with $D_3 = 0.052\, \text{m}$, $z_{23} = -9\, \text{mm}$, $z_{45} = 11\, \text{mm}$, $\beta_{23} = 43^\circ$, and $\beta_{45} = 83^\circ$. With this configuration, the heat losses are reduced by 51 W/m compared

Figure 6.12: Visualization of a configuration using reflective annulus insulation between the absorber and the envelope, together with selected rays and design parameters.
to the benchmark design, see Table 6.6. Despite the lower optical efficiency, an increase in the thermal efficiency of around 0.9% is achieved. The improvements obtained with the RAI are hence slightly lower than with the RGS.

### 6.4.3 Aplanatic Mirrors

In addition to the two simple designs assessed in the previous sections, more sophisticated secondary designs can be considered. Aplanatic mirrors are a promising option because they allow designs close to the thermodynamic limit [96]. However, most of the proposed designs are better suited for flat absorbers, and only a few configurations come into consideration for the use of tubular absorbers in PTC systems. The geometries of both the primary and secondary mirrors are constructed using an analytic solution of the mirror contours with parameters $s$, $K$, and $NA$ [99]. The parameter $s$ is the distance between the vertices of the primary and the secondary mirror, divided by the system’s focal length $F$. The distance between the vertex of the secondary and the FL, divided by $F$, is defined as $K$. The numerical aperture $NA$ is a measure of the maximum attainable concentration and the maximum angle at which a ray reaches the focus. The value of $NA$ is also equal to half of the primary’s aperture width, divided by $F$. In this study, the aperture width $w_{conc} = 6$ m is set as a constraint for any configuration, resulting in a focal length of $F = w_{conc}/2NA$ [99]. For the primary mirror, the same optical properties (reflectivity, angular errors) apply as for the parabolic primary of the benchmark design, while for the secondary mirror $\rho_{sec} = 0.95$ is used. Of all the analyzed configurations, only the category of dual-mirror aplanats with $s > 0$ and $K > 0$ yielded results that are comparable to the performance of the benchmark design. Some configurations of other categories (e.g., $s < 0$, $K < 0$) could not match the benchmark design as the larger secondary shapes resulted in low optical efficiencies. Other designs were dismissed because they either lack acceptable compactness or are unsuitable for circular receivers. In the optimization study, $D_3$, $s$, $K$, and $NA$ are simultaneously varied to find the configuration with the
maximum thermal efficiency. For any given configuration, the vertical position of the absorber tube is such that the top of the tube touches the vertex of the secondary mirror, see Figure 6.13.

The optimization study indicates that the maximum thermal efficiency of 60.4% is achieved with $D_3 = 0.048$ m, $s = 0.59$, $K = 0.0015$, and $NA = 0.998$. The corresponding receiver and secondary mirror are shown in Figure 6.13. Results for the optimum configuration, such as the optical and thermal efficiency, the effective concentration, and the total heat loss, are listed in Table 6.6. It can be seen that the thermal efficiency of the optimized aplanatic configuration is 1.6% higher than that of the benchmark design. This is partly because of a slightly higher optical efficiency despite the rather large shaded area above and beside the absorber tube. It is also due to lower heat losses, due to the smaller absorber size and the thermally emitted radiation that is reflected back to the absorber. The optimum configuration also exhibits a similar compactness than the benchmark design, corresponding to a parabolic trough with a rim angle of approximately 81°. A drawback of this design is that it would be difficult to implement in practice because the secondary mirror intersects the glass

Figure 6.13: Visualization of the optimum configuration using aplanatic mirrors, together with selected rays.
envelope. One solution would be to increase the diameter of the glass envelope to allow the entire secondary mirror to fit inside. That way, the secondary would lie completely within the protective vacuum atmosphere, allowing for the use of better suited materials [94, 106]. Furthermore, higher heat losses are incurred since the secondary mirror touches the absorber tube. (These losses were not considered in this study). A small gap between the secondary and the absorber may reduce heat loss with a slight reduction in the optical efficiency.

6.4.4 Tailored Seagull Design

The final secondary design is the tailored seagull concentrator. The mirror shape is constructed according to [104] using two main parameters. The first, denoted by $\theta_s$, can be interpreted as the variable acceptance angle rather than the assumed solar half-angle, since the angular spread of the incident solar radiation is normally distributed [32]. The second parameter, denoted by $C_s$, is the desired secondary concentration. The final size of the absorber tube is then determined from [104]

$$D_3 = \frac{w_{\text{conc}}}{\sin \psi_{\text{conc}}} \frac{\sin \theta_s}{C_s}$$  \hspace{1cm} (6.3)

In this study, the wing of the secondary reflector is designed as a straight line, which is a good approximation of the tailored shape [105]. Figure 6.14 shows a visualization of such a tailored seagull design. In the optimization study, $\theta_s$ and $C_s$ are simultaneously varied to find the configuration with the maximum thermal efficiency.

The results of the optimization study are presented as a contour plot in Figure 6.15, where the thermal efficiency is shown as a function of $\theta_s$ and $C_s$. The contour plot is obtained from a 4th-order polynomial least-squares fit to results from 90 simulations with the detailed 3D heat transfer model. (The root mean square difference between the fit and the results from the heat transfer model is around 0.07 percentage points.) Lowering $\theta_s$ decreases the absorber size, leading to worse interception and reducing the optical efficiency.
Figure 6.14: Visualization of the optimum configuration using the tailored seagull secondary design, together with selected rays.

Figure 6.15: Contour plot of thermal efficiency as a function of the acceptance angle $\theta_s$ and secondary concentration $C_s$ fitted to a total of 90 simulation points (scatter dots).
Increasing $C_s$ further decreases the diameter of the absorber tube and also increases the width of the tailored wing of the secondary, causing additional shading. Even with reduced heat losses from the smaller absorber, the lower optical performance leads to lower thermal efficiency. On the other hand, an increase in the acceptance angle causes better optical performance due to the increased absorber size, especially for small $C_s$, but due to the higher heat losses, these configurations again decrease the thermal efficiency. The maximum performance with $\eta_{th} = 59.6\%$ is achieved with the configuration displayed in Figure 6.14, using $\theta_s = 13.1$ mrad and $C_s = 1.72$, resulting in an absorber diameter of $D_3 = 0.046$ m. Values for the optical and thermal efficiencies, effective concentration, and total thermal losses are listed in Table 6.6. The optical efficiency of this optimum configuration is lower than that of the benchmark design due to the small absorber size. However, this is outweighed by the lower heat losses, leading to an overall increase in thermal efficiency of around 0.8%. Similar to the optimized aplanatic configuration, this secondary mirror design is difficult to implement in practice because the secondary touches the absorber. Again, a small gap between the absorber tube and the secondary mirror might improve the practicality and the overall performance of this design at the expense of reduced optical efficiency.

6.4.5 Temperature Distribution

Besides raising the thermal efficiency, some of the secondary designs presented here have the additional benefit of a more uniform distribution of the incident concentrated solar radiation. This leads to smaller temperature gradients along the circumference of the absorber tube, thereby reducing thermal stresses. The non-uniform distribution of both the incident solar radiation and surface temperatures can be accurately determined with the 3D heat transfer model. In Figure 6.16, the temperature distribution on the absorber tube is plotted as a function of the circumferential angle $\gamma$ for the benchmark design and the different optimum secondary configurations, assuming that $T_1 =$
290°C. The angle $\gamma = 180^\circ$ corresponds to the location at the bottom of the absorber tube, facing towards the primary concentrator mirror. It can be seen that the benchmark design exhibits the largest temperature difference of $\Delta T_{3,\text{max}} = 22.7$ K between the coldest and hottest part of the absorber. By contrast, the seagull design presents a much more balanced distribution with $\Delta T_{3,\text{max}} = 8.4$ K. The incident solar radiation is absorbed directly or via the involute or wing part of the secondary, causing several peaks of higher solar flux and temperature. The maximum temperature difference on the absorber tube of the aplanatic mirror configuration is also among the lowest with $\Delta T_{3,\text{max}} = 9.2$ K, but the secondary mirror causes absorption peaks near the top of the absorber, where little radiation is absorbed. As a result, the locations of maximum and minimum temperature are very close to each other, leading to higher temperature gradients than in the benchmark design.

![Figure 6.16: Absorber tube temperature as a function of the receiver circumferential angle at $T_1 = 290^\circ$C for different PTC configurations.](image-url)
6.5 Discussion

In this study, potential improvements in the performance of PTC systems were systematically evaluated using a detailed 3D optical and thermal analysis. The potential improvements in thermal efficiency due to ideal selective coating, glass envelope, and primary mirror are all close to 7%. The potential improvement due to an idealized supporting structure is only about 4%. Table 6.7 summarizes the results of each individual idealization, together with the results of a configuration in which all idealizations are combined. The ideal design has a thermal efficiency of 81.8%, an improvement of 23% compared to the benchmark design. This thermal efficiency comes close to the theoretical limit at zero heat losses (86.5%), where essentially only cosine losses caused by the solar incidence angle remain.

To demonstrate the robustness of the results obtained in the idealization and optimization study, simulations identical to those described in Section 6.3 were performed for an alternative PTC design. This alternative design is based on a concentrator mirror with an aperture width of $w_{\text{conc}} = 5$ m and a rim angle of $\psi_{\text{conc}} = 70^\circ$, approximating the Luz LS-2 collector [13]. Instead of molten salt, synthetic oil is used as the HTF [4]. The number of concentrator modules per loop is reduced to 24 to limit the maximum temperature at the outlet of the solar field to 390°C. The reflectance curve “Old Cermet ave” is used to

| Table 6.7: Simulation results for configurations with a single idealization and an ideal design with all idealization combined, together with the increases in thermal efficiency compared to the benchmark design. |
|---------------------------------|------------|----------------|----------------|-----------------------------|
| Configuration                  | $\eta_{\text{opt}}$ [%] | $q'_\text{loss}$ [W/m] | $\eta_{\text{th}}$ [%] | $\Delta\eta_{\text{th}}$ [%] |
| Benchmark design               | 65.0       | 229            | 58.8            | -                           |
| Ideal absorber                 | 68.1       | 82             | 65.8            | 7.0                         |
| Ideal glass                    | 68.2       | 75             | 66.1            | 7.3                         |
| Ideal mirror                   | 71.9       | 228            | 65.6            | 6.8                         |
| Ideal structure                | 69.0       | 218            | 63.1            | 4.3                         |
| Ideal design                   | 83.8       | 74             | 81.8            | 23.0                        |
calculate the spectral emissivity of a representative previous-generation cermet selective coating [50]. The remaining design parameters are identical to those of the benchmark design described in Section 6.2, including the absorber tube diameter of \( D_3 = 0.06 \) m. Figure 6.17 presents the thermal efficiency as a function of \( D_3 \) and \( \psi_{\text{conc}} \) for the alternative PTC design. For consistency, the limits of \( \psi_{\text{conc}} \) were selected to be identical to those in Figure 1, so the actual alternative design with \( \psi_{\text{conc}} = 70^\circ \) is not included in the figure. Comparison with Figure 6.1 shows that both the overall shape of the contour lines and the values of \( \eta_{\text{th}} \) are similar to those of the typical high-temperature PTC design discussed in Section 6.2. This can be explained by the relative importance of competing effects. On the one hand, the alternative design exhibits a better optical performance than the typical design due to the larger absorber size relative to the aperture area of the concentrator mirror. On the other hand, the smaller aperture area reduces the solar input to the system, thereby reducing the heat gained by the HTF. Furthermore, compared to the ENEA coating used in

![Contour plot](image-url)

Figure 6.17: Contour plot of \( \eta_{\text{th}} \) as a function of \( D_3 \) and \( \psi_{\text{conc}} \) for an alternative PTC design, assuming a concentrator mirror with \( w_{\text{conc}} = 5 \) m, a synthetic oil HTF with a maximum temperature of 390°C, and a previous-generation cermet selective coating.
the previous simulations, the “Old Cermet Ave” coating has a higher total emissivity, but this is compensated by the lower overall temperatures of the alternative design, leading to slightly smaller heat losses compared to the typical design. (It should be noted that this study analyzes only the performance of the solar field. If the efficiency of a subsequent power block would be taken into account, the overall solar-to-electric efficiency of a molten-salt PTC system with temperatures around 600°C would be significantly higher than that of an oil-based design at 400°C.)

The results for the alternative design with $\psi_{\text{conc}} = 70^\circ$ and $D_3 = 0.06$ m are compiled in Table 6.8. The table indicates that the absolute values differ somewhat from the idealization study of the typical system summarized in Table 6.7. The lower overall temperatures of the alternative design lead to lower heat losses, giving higher thermal efficiencies and larger improvements compared to the typical PTC system. Nevertheless, the same overall trends are observed: The improvement potentials of the selective coating and the glass envelope are similar and larger than the improvement potential of the concentrator mirror, and that the potentials of these three components are significantly larger than the improvement potential of the supporting structure, which is around 4% in both designs.

Table 6.8: Simulation results for the alternative PTC design, assuming $w_{\text{conc}} = 5$ m, $\psi_{\text{conc}} = 70^\circ$, a synthetic oil HTF with a maximum temperature of 390°C, and a previous-generation selective coating. Results shown for the initial design, individual idealizations, and all idealizations combined.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>$\eta_{\text{opt}}$ [%]</th>
<th>$q'_{\text{loss}}$ [W/m]</th>
<th>$\eta_{\text{th}}$ [%]</th>
<th>$\Delta\eta_{\text{th}}$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Benchmark design</td>
<td>64.9</td>
<td>216</td>
<td>57.7</td>
<td>-</td>
</tr>
<tr>
<td>Ideal absorber</td>
<td>68.6</td>
<td>29</td>
<td>67.5</td>
<td>9.8</td>
</tr>
<tr>
<td>Ideal glass</td>
<td>68.1</td>
<td>34</td>
<td>66.8</td>
<td>9.1</td>
</tr>
<tr>
<td>Ideal mirror</td>
<td>71.3</td>
<td>216</td>
<td>64.0</td>
<td>6.3</td>
</tr>
<tr>
<td>Ideal structure</td>
<td>69.0</td>
<td>208</td>
<td>62.1</td>
<td>4.4</td>
</tr>
<tr>
<td>Ideal design</td>
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<td>28</td>
<td>83.2</td>
<td>25.5</td>
</tr>
</tbody>
</table>
6.6 Conclusions

The optimum concentrator rim angle and absorber tube diameters with respect to the thermal efficiency of the solar field were determined for a typical current-generation, high-temperature PTC design. The geometric optimization revealed the improvement potential of the PTC system without implementing additional components or new materials. The results indicate that a maximum thermal efficiency is reached for a rim angle of 110° coupled to a slightly smaller absorber diameter than that used in conventional systems at the expense of a heavier structure and additional material costs. Idealizing the optical properties of the selective coating, the glass envelope, and the concentrator mirror resulted in improvements in the thermal efficiency of about 6.8-7.3%, while a system without shading from the support structure gives an increase of around 4.3%. Combining all idealized components would allow increases of up to 23.0% in the thermal efficiency. These conclusions are robust in the sense that similar results were obtained for an older-generation, lower-temperature design in addition to the current-generation, high-temperature design.

The use of several secondary mirror designs was also investigated. The improvements in thermal efficiency relative to the benchmark design range from 0.8% for the tailored seagull to 1.6% for the aplanatic mirror. These gains in thermal efficiency are smaller than those obtained with material improvements. However, these gains can be put into perspective by comparing them with the potential increase in thermal efficiency of a system with no mirror surface and tracking errors, where an improvement of 1.2% was determined for $D_3 = 0.06$ m (see Table 6.4). It should be noted that while the best performance was achieved using aplanatic mirrors, even very simple secondary mirror designs such as the reflective glass surface and the reflective annulus insulation deliver increases in thermal efficiency of 1.0% and 0.9%, respectively.

Most of the improvements presented here rely on ideal behavior and materials that may be difficult or impossible to achieve in practice. Some of the
secondary-mirror designs also present challenges concerning practicality, manufacturability, and cost. In addition, the benefit of adding secondary optics appears questionable, as factors influencing the optical performance may outweigh better thermal performance. Nevertheless, the results presented here point to the most promising directions for development to increase the thermal efficiency of PTC systems. To determine the most cost-effective approach, dedicated structural, manufacturing, and economic analyses must be carried out. Such analyses are beyond the scope of this work.
7 Summary

The central contribution of this thesis is the development and application of a detailed 3D heat transfer model for PTC systems. An innovation of this model is the implementation of MC ray tracing to determine not only the incident solar radiation, but also the exchange of thermally emitted radiation. An accurate description of the incident radiation merely allows the optical performance of a PTC design to be assessed. Using MC ray tracing for emitted radiation, however, generates an accurate and detailed description of the radiative heat losses, as non-uniform temperature distributions, semi-transparent glass, and the effects of shielding by supports and possible secondary optics are considered. In addition, the 3D representation of heat transfer allows for the analysis of the circumferential distribution and peak values of surface temperature and heat flux. Together with the ability to use detailed spectral data for the optical properties, the model presented in this thesis therefore enables a full optical and thermal analysis of PTC systems.

To ensure that the model accurately predicts the performance of PTC systems, simulation results were compared to experimental data of historical test setups. The heat transfer model accurately predicted heat loss, thermal efficiency, and glass temperatures from on- and off-sun experiments. To produce useful and representative results from the analysis of PTC systems, appropriate system parameters need to be specified in the heat transfer model. For this purpose, a study was performed to find optimum operating conditions that can be used in the evaluation and optimization of PTC systems. The analysis showed that common design points like the equinox or the solstice significantly over- or under-predict the yearly average thermal efficiency of the PTC solar field by as much as 11.5%. It was found that using weighted averages of DNI and incidence angles, operating conditions can be generated that match the
actual yearly efficiency very well, lying within 0.3% irrespective of the type of PTC system used and its geographical location. In addition to saving computational effort, the determined operating conditions provide a design point where the optimization study can be performed. This proposed approach using weighted average values has great potential not only to facilitate optimization studies, but to assess the expected yield of any kind of PTC system more accurately than commonly used methods. The use of weighted averages may even be applicable to evaluate the estimated yearly performance of other CSP systems, such as CRS, CPV, or linear Fresnel. The validity of this approach could easily be verified if a detailed heat transfer model that considers variations in the incidence angle is available for a specific CSP application and weighted averages of DNI and incidence angle are used in the analysis.

After developing the detailed 3D heat transfer model, validating its accuracy, and finding appropriate design points, it was used in the development of a novel PTC receiver as part of the HITECO research project. In a first step, the optical and thermal performance of new selective coatings was evaluated and their use in the new PTC design was assessed. The spectral data of the candidate coatings were used in the simulations of a generic PTC solar field and the results were compared to existing selective coatings. The analysis showed that the best candidate coating leads to thermal efficiencies that are 0.5% higher than comparable high-temperature selective coatings. In a second step, an experimental off-sun test setup was analyzed assuming that the best selective coating is used. The setup consists of two 6 m sections of the new receiver prototype that are heated to temperatures up to 600°C to assess the heat loss and other performance values. The heat transfer model was used to predict the heat loss at different absorber tube temperatures, annulus pressure levels, and gas compositions inside the receiver. Based on the simulations, the maximum required power output of the heating system and the expected temperatures in the main system components were assessed. Using the findings of the modeling studies, the specifications of the test equipment were determined and appropri-
ate components were selected. To enable uniform heat generation in the absorber tube and to simplify the test setup, a high-current impedance heating system was selected. Currents up to 1000 A are directly passed through the absorber tube. The receiver prototype also includes a novel vacuum system, with which the pressure inside the annulus can be controlled, reducing long-term heat loss in the solar field. Using a preliminary version of a selective coating, a first set of off-sun test data was generated and compared to simulation results of the detailed heat transfer model. The model was validated as its results agree well with the experimental data. These preliminary tests proved the feasibility of the manufacturing and testing process.

The development efforts in the HITECO project include the optimization of the proposed new receiver design. For this purpose, the optimization potential of a typical high-temperature PTC system was analyzed by idealizing the behavior and the properties of the absorber tube, the glass envelope, the concentrator mirror, and the supporting structure. In the analysis, the effect of an absorber or an envelope with ideal optical properties on the thermal efficiency of a PTC solar field was evaluated. Sigmoid functions proved to be an effective way to model and optimize the spectral reflectivity of selective coatings. The improvement achieved by using ideal concentrator mirrors and supporting structure was also analyzed. The increases in thermal efficiency for these idealized system components ranged from 4.3% to 7.3% compared to a selected benchmark PTC design, while a combination of all idealizations resulted in an increase of 23%. The analysis of an alternative PTC design has shown that the idealization approach is robust in that it predicts similar increases in thermal efficiency. Even though these improvements may not be fully achievable in practice, they indicate the most promising paths for research and development. In a second study, the benefit of using secondary optics to boost the optical efficiency and reduce heat losses was investigated. Simple additions, such as a reflective glass surface or reflective annulus insulations, as well as more sophisticated secondary mirror systems like aplanats or tailored secondar-
ies were analyzed. The use of secondary mirrors increased the thermal efficiency by up to 1.6% and reduced the circumferential temperatures gradients in the absorber tube. The mechanical and economic feasibility of secondary optics was not part of the study.

In conclusion, the detailed 3D heat transfer model developed in this thesis represents a more powerful tool than models reported in the literature. The new model combines an accurate optical analysis with a detailed assessment of the thermal behavior, leading to several improvements. Most importantly, the model enables a complete analysis of the performance of virtually any kind of PTC system irrespective of the choice of receiver design and mirror geometries, the materials used for the components and the HTF, or the optional implementation of additional optical devices. This was highlighted in the development of the HITECO receiver, where the expected performance of a new PTC receiver was simulated without the need for existing empirical data. The model also enables the assessment of uncontrollable parameters like operating conditions, the optimization of system components, and the evaluation of secondary optics, as shown in several case studies. These analyses are not possible with most reported heat transfer models, as they would focus either exclusively on the optical behavior, or assess the thermal efficiency based on rough estimations of the optical performance.
8 Outlook

From the findings of the modeling studies presented in this thesis, three main points are identified that represent a potential focus for further research to improve the heat transfer model. These three points include the pursuit of additional validation opportunities, the improvement of the calculation of the convective heat transfer terms, and the coupling of other models to the existing heat transfer model. These possible improvements are discussed in the following paragraphs.

To further increase the confidence in the predicted results of the heat transfer model, more detailed validation studies than the ones presented in this thesis are desirable. This is especially true for analyses that lie outside the range of the presented experimental data and cannot be explicitly validated. For instance, heat loss at HTF temperatures above 500°C or the optical behavior of a completely new concentrator design cannot be compared to measured results for lack of availability, but are extrapolated from existing data. In this regard, the proposed off-sun test campaign outlined in Section 5.3 represents a valuable opportunity to compare additional measurement data to values predicted by the heat transfer model once the optimum selective coating is available. In addition to comparing heat transfer at conditions that have not been reported in the past, such as absorber tube temperatures of 600°C or variable annulus gas pressures and compositions, the proposed test setup also provides information about materials, optical properties, and geometries in more detail than what is usually available in the literature. In addition, in the framework of the project, an on-sun test campaign with a prototype concentrator mirror is planned to follow the mentioned off-sun heat loss tests. The findings of the off-sun tests are incorporated in the improved prototype design, which is then tested under real conditions in the field. Again, the heat transfer model will be useful in as-
sessing the specifications of the on-sun test setup, based on the predicted behavior of the system. From this second test campaign, valuable data can be extracted, with which the optical part of the model can be calibrated. This is especially important, as comprehensive data on the optical properties of concentrator mirrors are generally not available for the systems presented in the literature. As a result, there is still a lot of uncertainty associated with the modeling of the mirror behavior, such as reflectivity and dispersion due to surface errors. The planned on-sun test setup will be an opportunity to assess the optical quality of the mirrors and to assure that the implemented optical model gives an accurate representation of the behavior of PTC mirrors.

In addition to enhancing the accuracy of the optical aspects of the heat transfer model, improvements could be made in the calculation of the convection terms $q'_{\text{conv,12}}$, $q'_{\text{conv,34}}$, and $q'_{\text{conv,56}}$ (see Figure 2.1). These convective terms are calculated using empirical Nusselt number correlations, which can be found in the literature and are usually derived from simplified problems. For instance, the correlation for the convective heat transfer to the HTF inside the absorber tube ($q'_{\text{conv,12}}$, see Section 2.4) assumes uniform temperature distribution around the tube wall. In the presented 3D heat transfer model this is clearly not the case. Since no suitable correlation could be found in the literature for these types of temperature distribution, one option to assess these convection terms more accurately is the use of computational fluid dynamics (CFD). For a specific temperature or heat flux distribution along the circumference, the heat transfer to the HTF could be calculated by solving the Navier-Stokes equations. This could be done by using existing CFD software, although this would create challenges with coupling the CFD analysis to the main heat transfer model. One solution would be to perform a comprehensive parameter study with the CFD software to derive a new Nusselt number correlation from the simulation results, which could then be used in the main heat transfer model in place of the empirical correlation. Alternatively, a custom-made Navier-Stokes solver could be implemented in the existing model to determine the convection terms simultaneously with the rest of the heat transfer analysis.
Such a solver could also be used to determine the fluid behavior inside the vacuum annulus. The molecular conduction is currently determined using a simple correlation that again assumes uniform temperature distributions and very low pressures, see Section 2.5. Studies have shown that in the pressure range that is expected to be experienced in PTC systems, the heat transfer across the annular gap may be acceptably modeled in the continuum regime, while a discrete analysis only marginally increases the accuracy with significantly higher computational cost [114, 115]. As a result, the FV method could be implemented to determine the heat transfer inside the vacuum annulus for different pressures and non-uniform temperature distributions on the wall surface. The accuracy of such an approach could then be validated using the detailed measurement data from the off-sun test campaign, where a wide pressure range in the annulus will be possible.

Another major convective term in the heat transfer analysis is convection from the glass envelope to the atmosphere \( (q'_{\text{conv,56}}) \), modeled using an empirical Nusselt number correlation derived for a cross flow around an isolated cylinder, see Section 2.6. Identical to the other convective terms, the assumptions of the used correlations of a uniform temperature distribution and a free-standing cylinder are not entirely valid for the case at hand. In particular, the concentrator mirror causes a flow field of the wind around the receiver that, depending on the orientation of the PTC system, is significantly different than the flow field around a cylinder in a free stream. Together with the non-uniform temperature distribution, the consequent convective heat transfer may in reality vary much more than predicted by the current heat transfer model. A CFD analysis of the flow field could produce valuable information about the range of heat transfer that is experienced from a PTC receiver. A preliminary review of the literature has shown that only little information is available on such analyses [116-118]. Similar to convection inside the absorber tube, a comprehensive parameter study could yield a more accurate correlation between flow and receiver properties and the convective heat transfer. The correlation would also
include additional system parameters that have not been previously considered, namely the orientation of the PTC system during the day and the proximity of adjacent mirrors, as well as their effect on heat transfer. The proposed on-sun test campaign mentioned above could serve as a source for experimental data to validate such a CFD analysis.

The third possible avenue of improvement concerns the coupling of additional models to the 3D heat transfer model, enabling simulations that go beyond optical and thermal analyses. For instance, to utilize the full potential of this heat transfer model of a solar field, a detailed representation of the power block could be added. This would allow for optimizations of the solar field in terms of electric output, instead of solely the thermal output. This addition of such a power block model would also serve as a basis for a potential economic model that may be implemented. The assessment of economic aspects, such as the cost of materials, assembly, or maintenance, would enable a financial evaluation of a particular PTC system and allow for an optimization in terms of economic parameters, such as the levelized cost of electricity production. As mentioned in Chapter 6, considering all relevant factors like economic characteristics in addition to optical, thermal, mechanical performance values enables a better evaluation of the feasibility of a particular system. In this regard, calculating the mechanical stress acting on key components of the PTC system could also yield valuable information. Assessing the forces induced by temperature gradients or wind loads can provide quantitative data on the required material properties and their costs, determine the limiting geometrical dimensions of the system, and highlight possible benefits of achieving lower temperature gradients, for instance through the use of secondary optics.

Apart from improving system components and implementing secondary optics, the heat transfer model could also be used to investigate the feasibility of more unconventional processes, using PTC technology. For instance, different HTF materials can easily be implemented in the model to study the effect of using air, CO₂, liquid metals, water/steam, or particle suspensions as a means to
transfer and store thermal energy. Furthermore, the model could be used to evaluate the feasibility of using PTC systems for low-temperature chemical processes, such as photocatalytic water detoxification, methanol reforming, or ammonia dissociation. Finally, similar to the study on secondary optics, new receiver designs, for example for CPV applications, can easily be implemented in the heat transfer model, assessing their optical and thermal performance.
Appendix A

Coefficients for the finite volume discretization

For the discretized differential equation

\[ a_{n,i,j} T_{n,i,j} = a_{n,i-1,j} T_{n,i-1,j} + a_{n,i+1,j} T_{n,i+1,j} + a_{n\pm1,i,j} T_{n\pm1,i,j} + S_{n,i,j} \]  

(A.1)

the coefficients \( a \) for the neighbor nodes contain the local thermal conductivity and geometrical distances. For instance, for a node \( i \) on the outer absorber tube surface (3), see Figure 2.1, the neighboring coefficients are described by \([31]\)

\[ a_{n,i-1,j} = a_{3,i-1,j} = \frac{k_{n,i-1,i,j} \cdot \Delta r}{r_{i-1,i} \Delta \theta} = k_{23,i-1,i,j} \cdot \frac{D_3 - D_2}{3D_3 + D_2} \cdot \frac{n_{seg2}}{\pi} \]  

(A.2)

\[ a_{n,i+1,j} = a_{3,i+1,j} = \frac{k_{n,i,i+1,j} \cdot \Delta r}{r_{i,i+1} \Delta \theta} = k_{23,i,i+1,j} \cdot \frac{D_3 - D_2}{3D_3 + D_2} \cdot \frac{n_{seg2}}{\pi} \]  

(A.3)

\[ a_{n-1,i,j} = a_{2,i,j} = \frac{k_{n-1,n,i,j} \cdot r_{n-1,n} \Delta \theta}{r_n - r_{n-1}} = k_{23,i,j} \cdot \frac{D_3 + D_2}{D_3 - D_2} \cdot \frac{n_{seg2}}{\pi} \]  

(A.4)

where \( k_{23} \) is the harmonic mean of the local thermal conductivities based on the temperatures of the two nodes. The geometric distances \( r_{i-1,i}, r_{n-1,n}, \) and \( r_n - r_{n-1} \) are visualized in Figure A.1.

Figure A.1: Geometric distances of the 2D discretization mesh in polar coordinates.
The convective heat transfer for a boundary node \( T_{n,i,j} \) in contact with the HTF, the annulus gas, or the ambient air is modeled as

\[
\dot{q}'_{n,i,j} = h_{n,i,j} \cdot (T_{f,i,j} - T_{n,i,j}) \cdot r_n \Delta \theta
\]  
(A.5)

where \( h_{n,i,j} \) is the local heat transfer coefficient, \( T_{f,i,j} \) is the local fluid temperature and \( r_n \Delta \theta = \pi D_n / n_{seg2} \) is the length of the control volume boundary. The coefficient \( a_{n,i,j} \) for the node \( T_{n,i,j} \) is the sum of all its neighbor’s coefficients and the portion of the source term that depends on the node’s temperature,

\[
a_{n,i,j} = a_{n,i-1,j} + a_{n,i+1,j} + a_{n\pm 1,i,j} + h_{n,i,j} r_n \Delta \theta
\]  
(A.6)

The source \( S \) contains all source terms, for instance the radiation terms determined in the ray tracing processes and the constant parts of the convective boundary condition terms,

\[
S_{n,i,j} = S_{C,n,i,j} \Delta V_{n,i,j} + T_{f,i,j} \cdot h_{n,i,j} r_n \Delta \theta
\]  
(A.7)

where \( \Delta V_{n,i,j} \) is the control volume. The new temperature at a specific node is determined through rearranging the discretization equation,

\[
T_{n,i,j} = \frac{a_{n,i-1,j} T_{n,i-1,j} + a_{n,i+1,j} T_{n,i+1,j} + a_{n\pm 1,i,j} T_{n\pm 1,i,j} + S_{n,i,j}}{a_{n,i,j}}
\]  
(A.8)
## Appendix B

### Results for the Analysis of a Current Generation PTC System

Table B.1: Analysis of a current-generation PTC system for Golden, USA.

<table>
<thead>
<tr>
<th>Design point</th>
<th>$I_{DNI}$ [W/m²]</th>
<th>$\theta$ [°]</th>
<th>$\eta_{th}$ [%]</th>
<th>$\eta_{th} - \eta_{th,y}$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\eta_{th,y}$ (Empirical DNI)</td>
<td>-</td>
<td>-</td>
<td>62.9</td>
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<tr>
<td>$\eta_{th,y}$ (TMY Data)</td>
<td>-</td>
<td>-</td>
<td>61.7</td>
<td>-</td>
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<tr>
<td>Empirical DNI, YTA</td>
<td>782</td>
<td>25.9</td>
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<tr>
<td>TMY data, YTA</td>
<td>687</td>
<td>26.5</td>
<td>66.3</td>
<td>4.6</td>
</tr>
<tr>
<td>Solstice, noon</td>
<td>979</td>
<td>16.3</td>
<td>72.5</td>
<td>9.6</td>
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<tr>
<td>Equinox, noon</td>
<td>971</td>
<td>40.1</td>
<td>55.2</td>
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<tr>
<td>Equinox, DTA</td>
<td>801</td>
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<tr>
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<tr>
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<td>32.2</td>
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<td>0.2</td>
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Table B.2: Analysis of a current-generation PTC system for Keahole, USA.

<table>
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<tr>
<th>Design point</th>
<th>$I_{DNI}$ [W/m²]</th>
<th>$\theta$ [°]</th>
<th>$\eta_{th}$ [%]</th>
<th>$\eta_{th} - \eta_{th,y}$ [%]</th>
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<tbody>
<tr>
<td>$\eta_{th,y}$ (Empirical DNI)</td>
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<td>-</td>
<td>69.5</td>
<td>-</td>
</tr>
<tr>
<td>$\eta_{th,y}$ (TMY Data)</td>
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<td>-</td>
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<tr>
<td>Empirical DNI, YTA</td>
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<td>17.4</td>
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</tr>
<tr>
<td>TMY data, YTA</td>
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<td>70.6</td>
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<tr>
<td>Solstice, noon</td>
<td>834</td>
<td>3.7</td>
<td>76.4</td>
<td>6.9</td>
</tr>
<tr>
<td>Equinox, noon</td>
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<td>20.1</td>
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<td>Equinox, DTA</td>
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Table B.3: Analysis of a current-generation PTC system for Atacama, Chile.

<table>
<thead>
<tr>
<th>Design point</th>
<th>$I_{DNI}$ [W/m$^2$]</th>
<th>$\theta$ [$^\circ$]</th>
<th>$\eta_{th}$ [%]</th>
<th>$\eta_{th} - \eta_{th,y}$ [%]</th>
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<tr>
<td>$\eta_{th,y}$ (Empirical DNI)</td>
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<td>-</td>
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<tr>
<td>Empirical DNI, YTA</td>
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<tr>
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<tr>
<td>Equinox, noon</td>
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<td>4.0</td>
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<td>817</td>
<td>22.1</td>
<td>69.4</td>
<td>0.1</td>
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Appendix C

Material properties used in the heat transfer model

Table C.1: Material properties of air [119]; $T$: temperature, $\rho$: density, $c_p$: specific heat capacity, $k$: thermal conductivity, $\nu$: kinematic viscosity, $Pr$: Prandtl number.

<table>
<thead>
<tr>
<th>$T$ [K]</th>
<th>$\rho$ [kg/m$^3$]</th>
<th>$c_p$ [J/kgK]</th>
<th>$k$ [W/mK]</th>
<th>$\nu$ [m$^2$/s]</th>
<th>$Pr$ [-]</th>
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<tr>
<td>100</td>
<td>3.6050</td>
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<tr>
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<tr>
<td>260</td>
<td>1.3580</td>
<td>1006</td>
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<td>1.214E-05</td>
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</table>
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