Doctoral Thesis

On-sun demonstration and heat transfer analysis of a modular pressurized air solar receiver

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ON-SUN DEMONSTRATION AND HEAT TRANSFER ANALYSIS OF A MODULAR PRESSURIZED AIR SOLAR RECEIVER

A thesis submitted to attain the degree of DOCTOR OF SCIENCES of ETH ZURICH (Dr. sc. ETH Zurich)

presented by
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2015
Abstract

A high-temperature high-concentration pressurized air solar receiver is considered for driving a power generation Brayton cycle. The modular design consists of an open-end dome-end cylindrical SiC cavity surrounded by a concentric annular reticulated porous ceramic (RPC) foam lined with insulating material, all contained in a stainless steel pressure vessel with a secondary concentrator attached to its windowless aperture. Concentrated solar energy is absorbed by the cavity and transferred to the pressurized air flowing across the RPC by combined conduction, convection, and radiation.

In this thesis, the demonstration of a full-scale solar receiver prototype is elaborated. On-sun experiments were performed at the solar research facility of the Weizmann Institute of Science, Israel, for up to 47 kW of concentrated solar radiative power input in the absolute pressure range of 2 – 6 bar. The peak outlet air temperatures reached 1206 °C for an average solar concentration ratio of 2500 suns. A notable thermal efficiency – defined as the ratio of the enthalpy change of the air flow divided by the solar radiative power input through the aperture – of 91% was achieved at 700 °C and 4 bar.

Concurrently, a numerical model was developed to elucidate the major heat loss mechanisms and to study the impact on the solar receiver performance caused by changes in process conditions, material properties, and geometry for 50 kW solar radiative power input. The governing mass, momentum and energy conservation equations were numerically solved for steady-state conditions by coupled Monte Carlo and finite volume techniques. Model validation was accomplished with experimental results. For all investigated pressures in the range 4 – 15 bar, an optimum design point was found at an outlet air temperature of 665 °C, yielding a thermal efficiency of 80%. For the maximum allowed cavity temperature of 1600 °C, the outlet air temperature and thermal efficiency are 1345 °C and 51%, respectively. For all operating points, reradiation is the
dominant heat loss. At the ~80% thermal efficiency level, relative improvements of up to 11% are possible via geometry and material optimization.

Subsequently, the data obtained from the numerical simulations was extended to higher pressures to fully capture the receiver’s performance potential in gas turbine applications. The thermal performance of a solar power tower using an array of high-temperature pressurized air solar receivers was analyzed for a simple Brayton, a recuperated Brayton, a combined Brayton-Rankine cycle, and a dry steam reheat combined cycle. While the power cycle’s heat-to-work efficiency increases with higher turbine inlet temperatures and pressures, the receiver’s solar-to-heat efficiency decreases at higher operating temperatures and pressures. Using a solar receiver array with a preheat stage, a peak solar cycle efficiency – defined as the ratio of the cycles net work output divided by the solar radiative power input through the receiver’s aperture – of 39% was achieved in a combined cycle operating at a receiver outlet air temperature of 1300 °C and a Brayton cycle pressure ratio of 9 driving a Rankine cycle operating at 100 bar and 550 °C superheated steam.
Zusammenfassung


Parallel wurde ein numerisches Modell entwickelt, das den Prototypen abbildet. Das Modell kam zum Einsatz, um die vorherrschenden Wärmeverlustmechanismen bei 50 kW Eingangsleistung aufzuzeigen und die Receiverleistung in einem erweiterten Spektrum an Betriebsbedingungen, sowie bei veränderter Materialbeschaffenheit und Receiver-Geometrie zu studieren.

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First of all, I would like to express my deepest gratitude towards Prof. Dr. Aldo Steinfeld for entrusting me with this exciting, challenging project and for his supervision, guidance, and support.

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Nomenclature

Latin characters

1 – 5  process points in gas turbine cycle
A – F  process points in steam turbine cycle
A – E  fitting coefficients, -
\(A\)  area, m\(^2\)
\(a\)  semi-major axis, m
B  bottom
C  center
\(C\)  concentration ratio, suns
\(C, c\)  costs, $
\(C_p\)  pressure coefficient, -
\(c_p\)  specific heat capacity, J kg\(^{-1}\) K\(^{-1}\)
\(D\)  flange diameter, m
d  beam-down mirror diameter, m
d\(_{\text{nom}}\)  nominal pore diameter, mm
F  focus
\(f\)  center-to-focus distance, m
\(f\)  cost share, -
\(f\)  receiver stage power share, -
\(f\)  serial-to-parallel slab ratio, -
h  specific enthalpy, kJ kg\(^{-1}\)
h  tower height, m
I  investment cost, $
I_{\text{DNI}}  direct normal irradiance, W m\(^2\)
k  conductivity, W m\(^{-1}\) K\(^{-1}\)
L  latitude, °deg
\(l\)  load factor, -
\( \dot{m} \)  
\( \text{mass flow rate, g s}^{-1} \)

\( N \)  
\( \text{average number of reflections, -} \)

\( p \)  
\( \text{absolute pressure, bar} \)

\( p \)  
\( \text{electricity price, $} \)

\( \Delta p \)  
\( \text{pressure drop, mbar} \)

\( \dot{Q} \)  
\( \text{power, heat, kW} \)

\( \dot{Q}_{\text{in}} \)  
\( \text{solar radiative power input, kW} \)

\( q_{\text{in}} \)  
\( \text{specific solar radiative energy input, kJ kg}^{-1} \)

\( \dot{q}'' \)  
\( \text{radiative heat flux, convective heat flux, kW m}^{-2} \)

\( R \)  
\( \text{revenue, $} \)

\( R \)  
\( \text{specific gas constant of air, 287 J kg}^{-1} \text{ K}^{-1} \)

\( R \)  
\( \text{target radius, m} \)

\( R^2 \)  
\( \text{coefficient of determination, -} \)

\( r \)  
\( \text{radial coordinate, m} \)

\( s \)  
\( \text{specific entropy, kJ kg}^{-1} \text{ K}^{-1} \)

\( s \)  
\( \text{specific surface area, m}^{-1} \)

\( T \)  
\( \text{top} \)

\( T \)  
\( \text{temperature, °C} \)

\( \overline{T} \)  
\( \text{mean temperature, °C} \)

\( \Delta T \)  
\( \text{temperature difference, °C} \)

\( t \)  
\( \text{thickness, mm} \)

\( t \)  
\( \text{time, s} \)

\( \Delta t \)  
\( \text{time interval, s} \)

\( \dot{W} \)  
\( \text{power, kW} \)

\( w \)  
\( \text{specific work, kJ kg}^{-1} \)

\( v \)  
\( \text{fluid velocity, m s}^{-1} \)

\( v \)  
\( \text{specific volume, m}^{3} \text{ kg}^{-1} \)

\( x \)  
\( \text{axial coordinate, m} \)

\( x \)  
\( \text{vapor quality, -} \)

\( z \)  
\( \text{vertical coordinate, m} \)
Greek characters

\( \alpha \) solar altitude angle, °deg
\( \alpha \) tilt angle, °deg
\( \beta \) absorption coefficient, m\(^{-1}\)
\( \gamma \) air heat capacity ratio, -
\( \gamma \) solar azimuth angle, °deg
\( \varepsilon \) eccentricity, -
\( \varepsilon \) heat exchanger effectiveness, -
\( \varepsilon \) total hemispherical emissivity, -
\( \eta \) efficiency, -
\( \theta_{\text{in}} \) acceptance angle, °deg
\( \theta_{\text{out}} \) exit angle, °deg
\( \mu \) kinematic viscosity, kg m\(^{-1}\) s\(^{-1}\)
\( \Pi \) pressure ratio, -
\( \rho \) density, kg m\(^{-3}\)
\( \rho \) reflectivity, -
\( \sigma \) angular surface dispersion error, mrad
\( \sigma \) Stefan-Boltzmann constant, 5.67\( \cdot 10^{-8} \) W m\(^{-2}\) K\(^{-4}\)
\( \tau \) excess temperature, °C
\( \Phi \) generic parameter/variable
\( \varphi \) polar coordinate, m
\( \varphi \) porosity, -

Subscripts

1 – 3 position in receiver air flow
1 upper
2 lower
abs absorbed
amb ambient
ap approach
avg average
c compressor
cav cavity
corr corrected
Nomenclature

w    water
w    winter solstice
+    upper RPC location
–    lower RPC location

Dimensionless groups

Nu    Nusselt number
Pr    Prandtl number
Ra    Rayleigh number
Re    Reynolds number

Abbreviations

BAF    air flow guiding baffles
BC    Brayton cycle
BD    beam-down
CC    combined Brayton-Rankine cycle
CFD    computational fluid dynamics
CPC    compound parabolic concentrator
CPU    central processing unit
CSP    concentrating solar power
DIN    Deutsches Institut für Normung
DNI    direct normal irradiance
DPLS    direct pore-level simulation
E    East
ETH    Eidgenössische Technische Hochschule (Swiss Federal Institute of Technology), Zürich, Switzerland
FV    finite volume
GAST    Gas-cooled solar tower
HE    heat engine
HF    heliostat field
HRSG    heat recovery steam generator
HT    high-temperature
HTF    heat transfer fluid
HX  heat exchanger
IR   infra-red
LT   low-temperature
MC   Monte Carlo ray-tracing
MM   mismatch
N    North
PCU  power conversion unit
PPI  pores per inch
PSI  Paul Scherrer Institut, Villigen, Switzerland
RC   recuperated Brayton cycle
RHCC reheated combined Brayton-Rankine cycle
RPC  reticulate porous ceramic
S    South
SCH  Scheffler dish
TT   tower top
WIS  Weizmann Institute of Science, Rehovot, Israel

Al₂O₃  alumina
SiO₂  silica
SiC   silicon carbide
SiSiC reaction-bonded silicon infiltrated α-silicon carbide and β-silicon carbide composite
SSiC  sintered α-silicon carbide
1 Introduction

1.1 Motivation

The law of energy conservation dictates that the total energy of an isolated system remains constant. While energy can be converted or stored in many different forms (e.g. chemical, electrical, mechanical, thermal), it can neither be created nor destroyed. Assuming Earth to be a closed system, its energy balance is solely dictated by radiative energy transfer to and from its surroundings, *i.e.* empty space. The Sun is the only radiation source in Earth’s proximity that has a significant influence on its energy balance. Thus, all forms of energy stored on Earth origins from the Sun, *i.e.* biomass and fossil fuel, which also counts as converted biomass. Exceptions are radioactive decay, residual heat stored inside Earth, and the lunar part of the tidal potential. Intermediate forms of energy such as wind and the atmospheric water cycle are also fueled by sunlight due to evaporation processes and the irregularly distributed heating of the land masses.

The increasing energy demand of Earth’s population led to an excessive exploitation of the stored fossil sunlight, which was accumulated over hundreds of millions of years. The fear of running out of fuels has been around for centuries, however no reliable predictions could be made so far. Keeping in mind the risks and uncertainties of nuclear energy, the direct and indirect (via wind, hydro, biomass) energy conversion of solar radiation is the strongest option for a sustainable energy future.

While sunlight suffers from a comparably low power density, *i.e.* $<1000 \text{ W/m}^2$ at sea level, it is technically simple to concentrate it with optical devices. Following Stefan-Boltzmann law, highly concentrated solar radiation leads to high absorber temperatures and hence provides high-quality thermal energy with high exergetic value to drive a heat engine.
1.2 Concentrated solar power towers

State-of-the-art concentrated solar power (CSP) technologies using solar towers can be divided into two fundamental groups: (a) solar power is directly added to the working fluid, which gets heated up and is then expanded in a turbine, or (b) solar power is added to an intermediate heat transfer fluid (HTF), which then heats the working fluid at a later stage. Judged from recent development in commercial CSP plants, direct steam generation and molten salt as HTF appear to be the most mature technologies in these two categories, cf. Table 1-1:

<table>
<thead>
<tr>
<th>CSP plant</th>
<th>First operation</th>
<th>Power (MWel / MWth)</th>
<th>HTF</th>
</tr>
</thead>
<tbody>
<tr>
<td>Abengoa: PS 10 (Spain)</td>
<td>2006</td>
<td>10 / 55</td>
<td>saturated steam</td>
</tr>
<tr>
<td>Abengoa: PS 20 (Spain)</td>
<td>2009</td>
<td>20 / 110</td>
<td>saturated steam</td>
</tr>
<tr>
<td>eSolar: Sierra Sun Tower (USA)</td>
<td>2010</td>
<td>5 / 2 × 10</td>
<td>superheated steam</td>
</tr>
<tr>
<td>Torresol: Gemasolar (Spain)</td>
<td>2011</td>
<td>19.9 / 220</td>
<td>molten salt</td>
</tr>
<tr>
<td>BrightSource: Coalinga (USA)</td>
<td>2013</td>
<td>0a / 29</td>
<td>saturated steam</td>
</tr>
<tr>
<td>BrightSource: Ivanpah (USA)</td>
<td>2013</td>
<td>3 × 130 / 3 × 300</td>
<td>superheated steam</td>
</tr>
<tr>
<td>SolarReserve: Crescent Dunes (USA)</td>
<td>2015</td>
<td>110 / 565</td>
<td>molten salt</td>
</tr>
</tbody>
</table>

a process heat
Despite this promising development in CSP plants, the current technologies are limited in their peak temperatures to ~650 °C for superheated (or supercritical) steam [2] and 565 °C for molten salt [3]. Improvements towards higher temperatures are expected [4]. For an ideal reversible heat engine working between said $T_{\text{hot}}$ and ambient, the corresponding Carnot efficiencies, $\eta_{\text{Carnot}}$, are limited to ~0.65:

$$\eta_{\text{Carnot}} = 1 - \frac{T_{\text{amb}}}{T_{\text{hot}}}$$  \hspace{1cm} (1.1)

Essentially, $\eta_{\text{Carnot}}$ provides a qualitative rating of the exergetic value of the heat added to the heat engine at $T_{\text{hot}}$. If $T_{\text{hot}}$ can be increased to >1000 °C, $\eta_{\text{Carnot}}$ exceeds 0.8. Gas turbines typically operate at temperatures exceeding 1000 °C [5]. In CSP systems based on solar towers, significant improvement in terms of the solar-to-electricity efficiency may be achieved by supplying high-temperature heat to drive a gas turbine, e.g. in a combined Brayton-Rankine cycle [3,6]. Figure 1-1 shows a schematic of a solar combined cycle, including a combustor allowing round-the-clock operation in hybrid configuration.

1.3 Solar-driven gas turbines

Several technology development programs like GAST (1981 – 1987) [7], CONSOLAR (2001 – 2004) [8], and SOLGATE (1998 – 2002) [9] have been carried out to successfully demonstrate the feasibility of solar gas turbine systems. SOLUGAS is an ongoing European project aiming at the demonstration of the first pilot solar hybrid gas turbine power plant at commercial size [10].

Table 1-2 gives a short overview on the key parameters of these previous developments. A preliminary system analysis indicated that for becoming competitive compared to solar steam-based Rankine-cycles, temperatures in the range of 700 – 900 °C and moderate pressures up to 6 bar are technically feasible for the integration of a solar receiver with a gas turbine.
**Figure 1-1 – Schematic of the solar combined Brayton-Rankine cycle**

**Table 1-2 – Key parameters of solar-driven gas turbine development programs**

<table>
<thead>
<tr>
<th>Program</th>
<th>Temperatures (°C)</th>
<th>Pressures (bar)</th>
<th>Power</th>
</tr>
</thead>
<tbody>
<tr>
<td>GAST</td>
<td>~800 – 1000</td>
<td>9 – 10</td>
<td>20 MW_{el}</td>
</tr>
<tr>
<td>CONSOLAR</td>
<td>~300 – 1200</td>
<td>8 – 20</td>
<td>&gt;470 kW_{th}</td>
</tr>
<tr>
<td>SOLGATE</td>
<td>~550 – 1000</td>
<td>6.5 – 15</td>
<td>&gt;230 kW_{el}</td>
</tr>
<tr>
<td>SOLUGAS</td>
<td>~800</td>
<td>10</td>
<td>&gt;4 MW_{el}</td>
</tr>
</tbody>
</table>
1.4 Pressurized air receivers

The required temperatures and pressures of the working fluid at the gas turbine inlet are typically in the range 700 – 1400 °C and 3 – 35 bar [5]. To meet these temperature and pressure requirements, several pressurized air receivers have been developed. Most promising previous design concepts of solar receivers were based on directly irradiated volumetric absorbers that required the use of transparent windows [11–14], which are critical and troublesome components [15,16]. Alternatively, an indirectly irradiated windowless solar receiver was proposed at ETH. The design, fabrication, and experimental testing of a lab-scale solar receiver for ~3 kWth solar radiative power input was reported [17,18]. To find promising candidates for a receiver scale-up, various cavity dimensions and ceramic materials were tested on a solar tower under concentrated solar radiative power inputs of ~37 kWth [19].

1.5 Thesis outline

This thesis was carried out in the framework of a joint ALSTOM – ETH project entitled “Solar-driven Gas Turbine”. The project aimed at the fabrication and experimental on-sun testing of a full-scale solar receiver, numerical heat transfer modeling, and power cycle integration studies.

The engineering concept, design, and fabrication of the solar receiver prototype are described in Chapter 2. Chapter 3 explains the setup and Chapter 4 presents the experimental results.

A numerical heat and mass transfer model of the receiver is described in Chapter 5. Chapter 6 presents the model predictions and receiver performance maps.

Chapter 7 provides a power cycle analysis with integrated solar receiver and brief economic considerations. Finally, Chapter 8 concludes this thesis with a summary and outlook.


2 Solar Receiver

2.1 Working principle

A schematic of the indirectly irradiated solar receiver concept is shown in Figure 2-1. The design consists of a cylindrical, cavity-type opaque absorber, which efficiently absorbs incoming solar radiation. The absorbed heat is then conducted through the cavity wall and heats a reticulated porous ceramic (RPC) foam layer surrounding the cavity. Due to the high surface area within the porous medium, the heat is efficiently transferred to the pressurized air flowing across the RPC via combined conduction, convection, and radiation. Heat losses towards the surroundings are reduced by a layer of non-porous insulating material, all contained in a metallic pressure vessel. A compound parabolic concentrator (CPC, [20]) at the cavity aperture further increases the solar concentration ratio and reduces the aperture size and consequently reradiation losses [17–19].

---

1 Material of this Chapter has been published in:
2.2 Receiver prototype

The modular design of the solar receiver is schematically shown in Figure 2-2. Key dimensions and materials are listed at the end of this Chapter in Table 2-1 and Table 2-2, respectively. The prototype consists of an open-end dome-end cylindrical cavity surrounded by a concentric RPC foam. The cavity is made of sintered α-silicon carbide (SSiC), which has a relative high thermal conductivity and exhibits reasonable thermal stress resistance at elevated temperatures \([19,21]\). The RPC is also made of silicon carbide (SiC), whose effective heat transport properties have been determined by direct pore-level numerical simulations on 3D tomographic scans, \(cf.\) Section 5.2. The cavity and RPC, lined with \(\text{Al}_2\text{O}_3-\text{SiO}_2\) insulation material, are contained in a pressure vessel made of 316L stainless steel designed for pressures up to 9 bar at 25 °C and 6 bar at 120 °C steel temperature. At elevated temperatures, pressures higher than 6 bar had to be avoided due to pressure vessel regulations \([22–24]\). The ceramic cavity is sealed and held in place by a stuffing box made of Inconel 617, using high-temperature silica-based sealing materials. The stuffing box mechanism is self-sealing if pressure is applied. While the cavity is manufactured in one piece \([19]\), the RPC and insulation are made from stacked rings, \(cf.\) Figure
2-2. A CPC is incorporated at the cavity aperture to boost the solar concentration ratio, $C$, to peak values exceeding 3000 suns and to reduce the aperture size. Cold air enters the receiver near the aperture to keep the cavity sealing, front plate, and CPC cooled and to enhance the heat transfer at the hottest part of the cavity [19]. The hot air exits through an axial outlet located on the rear part of the vessel.

All inner receiver parts (cavity, RPC, and insulation) can be easily accessed and replaced by removing the back part of the pressure vessel near the dome-end of the cavity allowing experimental investigation of multiple RPC configurations inside the same receiver vessel. Due to manufacturing tolerances and the stacking concept, gaps between the inner parts were inevitable. To mitigate the possibilities of the fluid to bypass the RPC via the gaps between the insulation blocks, Al$_2$O$_3$-SiO$_2$-based insulation wool was squeezed into the most prominent gaps.

---

Figure 2-2 – Schematic of the solar receiver prototype. The modular design consists of a cylindrical SiC cavity surrounded by a concentric annular RPC foam contained in a stainless steel pressure vessel, with a CPC attached to its windowless aperture.
Due to the decreasing insulation thickness in the conical outlet part of the pressure vessel, the highest thermal strain was observed at the welding between the outlet flange and the reducer cone. To support gas exit temperatures beyond the thermal limit of the materials, the DN50 outlet flange is water-cooled via a copper tube inlay and air-cooled by two industrial vents. Since the outlet air temperature is measured upstream of these cooled parts, the thermal performance of the receiver is not biased.

<table>
<thead>
<tr>
<th>Part</th>
<th>Dimension</th>
</tr>
</thead>
<tbody>
<tr>
<td>CPC inlet aperture diameter</td>
<td>Ø 634 mm</td>
</tr>
<tr>
<td>CPC outlet aperture diameter</td>
<td>Ø 147 mm</td>
</tr>
<tr>
<td>Cavity inner diameter</td>
<td>Ø 250 mm</td>
</tr>
<tr>
<td>Cavity length</td>
<td>500 mm</td>
</tr>
<tr>
<td>Cavity wall thickness</td>
<td>7 mm</td>
</tr>
<tr>
<td>Receiver diameter</td>
<td>DN 400 PN 10</td>
</tr>
<tr>
<td>Outlet flange</td>
<td>DN 50 PN 16</td>
</tr>
<tr>
<td>Outlet pipe</td>
<td>2 &quot;</td>
</tr>
<tr>
<td>Inlet tubes</td>
<td>½ &quot;</td>
</tr>
</tbody>
</table>

\[ a \text{ DIN 2632, } b \text{ DIN 2576 } \]

<table>
<thead>
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<th>( T (\degree \text{C}) )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cavity</td>
<td>Halsic S(^{a}) – Pressureless sintered ( \alpha )-SiC (SSiC)</td>
<td>1600</td>
</tr>
<tr>
<td>RPC</td>
<td>Reaction-bonded silicon infiltrated ( \alpha )-SiC and ( \beta )-SiC composite (SiSiC)</td>
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</tr>
<tr>
<td>Insulation</td>
<td>Insulform 1200(^{b}) – ( \text{Al}_2\text{O}_3 ) (47%), ( \text{SiO}_2 ) (52%)</td>
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</tr>
<tr>
<td>Pressure vessel</td>
<td>X2CrNiMo17-12-2 (316L)</td>
<td>550</td>
</tr>
<tr>
<td>Front plate, stuffing box</td>
<td>Inconel Alloy 617 (N06617)</td>
<td>1050</td>
</tr>
<tr>
<td>Sealings</td>
<td>Klinger – Milam PSS(^{c})</td>
<td>900</td>
</tr>
</tbody>
</table>

\[ a \text{ Ref. [25], } b \text{ Ref. [26], } c \text{ Ref. [27] } \]
3 Experimental setup

3.1 Solar tower

Experimentation was carried out at the solar tower of the solar research facility of the Weizmann Institute of Science (WIS) in Rehovot, Israel. The solar receiver system was located ~30 m above the heliostat field. Figure 3-1 shows a photograph of the experimental setup, cf. Appendix A. The fixed CPC was truncated [13,19,20], with an acceptance angle, $\theta_{in} = 12^\circ$deg and an exit angle, $\theta_{out} = 65^\circ$deg, tilted downward by $\alpha = 26.6^\circ$deg from the horizontal and 23.6 $^\circ$deg towards north-east; hence, it could only accept the reflected sunlight from the 13 heliostats highlighted in Figure 3-2. Serving as primary concentrators, each heliostat consisted of 20 flat specular mirror facets aligned in such a way that they approach a spherically curved surface with paraxial focus located at the CPC inlet. Table 3-1 summarizes the heliostat field characteristics.

For on-sun experimentation, a reduced set of ten heliostats was used to obtain the desired solar power input, cf. Figure 3-2. While the heliostats 104, 206, 308, and 408 were canted directly to the CPC inlet aperture, heliostat 204 was canted to an adjacent experiment also located on the 7th floor 4 m east of the CPC. The heliostats 304, 306, 406, 506, and 508 were canted to a beam-down mirror mounted on the western side of the solar tower with focus at a height of ~48 m, cf. Figure A-3 in the Appendix. This difference in slant range caused a reduction in the solar power delivery of said heliostats. Experimentation was carried out for a maximum solar radiative power input through the cavity aperture, $\dot{Q}_{in} = 47$ kWth, determined by water calorimeter measurements, and Monte Carlo ray-tracing, cf. Sections 3.3 and 3.4.

---

Figure 3-1 – Photograph of the experimental setup during operation. Located ~30 m above the heliostat field, the receiver is attached to a water-cooled CPC. The cooling system of the CPC is seen in the background and the northern part of the heliostat field is seen through the window [28].

Table 3-1 – Specifications of the solar research facility of the WIS [29]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Latitude</td>
<td>31° 54' 40&quot; N</td>
</tr>
<tr>
<td>Longitude</td>
<td>34° 49' 5&quot; E</td>
</tr>
<tr>
<td>Time zone</td>
<td>UTC + 02:00</td>
</tr>
<tr>
<td>Standard time meridian</td>
<td>30° E</td>
</tr>
<tr>
<td>Elevation</td>
<td>44.6 m</td>
</tr>
<tr>
<td>Tower height</td>
<td>56.5 m</td>
</tr>
<tr>
<td>Target height</td>
<td>30.4 m</td>
</tr>
<tr>
<td>Tower base dimensions</td>
<td>12 × 15 m</td>
</tr>
<tr>
<td>Number of heliostats</td>
<td>64</td>
</tr>
<tr>
<td>Number of facets per heliostat</td>
<td>20</td>
</tr>
<tr>
<td>Total reflective surface area</td>
<td>3514 m²</td>
</tr>
<tr>
<td>Reflective surface area per heliostat</td>
<td>54.91 m²</td>
</tr>
<tr>
<td>Heliostat dimensions(^a)</td>
<td>7.84 × 7.85 m</td>
</tr>
<tr>
<td>Facet dimensions</td>
<td>1.56 × 1.76 m</td>
</tr>
</tbody>
</table>

\(^a\) including gaps between facets
Figure 3-2 – WIS heliostat field layout [29]. The blue ellipse indicates the intersection of the CPC acceptance cone and the horizontal. Only the highlighted heliostats were used for the experiments; they are canted to the CPC inlet aperture (green), another experiment on the 7th floor (yellow), and a beam-down mirror attached to the western side of the solar tower at ~48 m height (orange).

3.2 Receiver setup

Figure 3-3 shows a simplified schematic of the experimental setup. A full piping and instrumentation diagram of the experimental setup is later provided in Figure 3-5. Dry compressed air at ambient temperature was supplied by three screw compressors (Howden Compressor, Airmac). A cold air bypass was used to quench the hot outlet air to below 300 °C by mixing immediately after the solar receiver outlet to meet exhaust gas safety requirements and to reduce the material requirements of the downstream instrumentation. Air flow rates were monitored by two Coriolis-type mass flow meters (Endress+Hauser, Promass 80F) located between the compressors and the inlets to the solar...
Experimental setup

receiver ($\dot{m}_1$) and redundantly, at the outlet after the outlet-bypass mixing ($\dot{m}_3$), cf. Figure 3-3 and Figure 3-5. The cold bypass flow rate ($\dot{m}_2$) was monitored by a thermic mass flow meter (Endress+Hauser, T-mass A150). Two valves (Metso, Neles) were used to control the bypass air flow ratio and the system pressure, $p$. To increase the outlet air temperature (measured before mixing, cf. Figure 2-2) at constant solar power input, the air mass flow rate through the solar receiver, $\dot{m} = \dot{m}_1 - \dot{m}_2$, was reduced step-wise from ~150 g/s to ~10 g/s, while the bypass flow rate was increased accordingly.

![Flow schematic of the experimental setup](image)

Temperatures were measured with type-K thermocouples, located at inlet, outlet, exhaust pipe, upper and lower part of the RPC, and the cavity sealing. To improve accuracy, the thermocouple measuring the outlet air temperature, $T_{\text{out}}$, was shielded by a $\varnothing 1$ mm Al$_2$O$_3$ tube. Figure 3-4 and Figure 3-5 indicate the thermocouple locations:
(a) inlet air temperature, $T_{in}$
(b) outlet air temperature (shielded), $T_{out}$
(c) upper RPC temperature, $T^+$
(d) lower RPC temperature, $T^-$
(e) exhaust air temperature, $T_{ex}$
(f) cavity sealing temperature, $T_{cav}$

Figure 3-4 – Locations of the thermocouples and pressure sensors

For the determination of the receiver’s thermal efficiency, later defined in Equation (4.7), only $T_{in}$ and $T_{out}$ were of importance. The other temperatures were monitored for supervision. The pressure was monitored by two pressure transmitters (Keller, 33 X) placed before and after the RPC. A pop-off valve was installed to cope with overpressure exceeding 6 bar, cf. Figure 2-2 and Figure 3-5. As an additional safety measure, the temperature of the pressure vessel surface was monitored by an infra-red (IR) camera (FLIR, A320).

The direct normal irradiance (DNI), $I_{DNI}$, was measured by a pyrheliometer (Eppley, NIP) located at the northern end of the heliostat field.
Figure 3-5 – Piping and instrumentation diagram of the experimental setup. The controlled parameters are indicated in brackets.
Pressure levels of $p = 2, 4$ and $6 \text{ bar}$ were experimentally investigated. They were set with an average accuracy of $\pm 1.2\%$, except for the $2 \text{ bar}$ case, in which high $\dot{m}$ led to an increased pressure drop through the piping downstream of the solar receiver resulting in system pressures of $\sim 2.2 \text{ bar}$. Pressure increase/decrease during heating/cooling was adjusted by opening/closing the valves.

Experimental on-sun testing was split into three experimental campaigns, each lasting around one month taking place in April, August, and October 2013. Each campaign investigated a different RPC configuration:

(a) 10 pores per inch (PPI), April 2013
(b) 20 PPI, August 2013
(c) 10 PPI with five air flow guiding baffles (BAF), October 2013

Figure 3-6 shows photographs of the 10 PPI and 20 PPI RPC foams. The air flow guiding baffles were made from a silicate-based sealing-paste (Coltogum 1500) and alternately covered either the upper or lower half of the RPC annulus, spaced 75 mm, cf. Figure 3-7. Effective foam properties were determined via direct pore-level simulations, cf. Section 5.2.
3.3 Calorimeter measurements

To account for seasonal changes of the solar power input, two sets of water calorimeter measurements were taken at the beginning of each experimental campaign. The calorimeter consists of a black coated cavity wrapped in copper cooling tubes and an insulation layer. The cooling water flow rate was measured using a digital paddle wheel flow sensor (Blue-White Industries, Digiflo F 2000), while the inlet and outlet water temperatures were measured by type-J thermocouples, cf. Figure 3-8. The water-calorimeter was mounted with its aperture attached to the exit of the CPC. Two measurements of each individual heliostat were taken: the first while all the other heliostats were in stow position, and the second while the other heliostats were set to tracking the Sun but reflecting the sunrays to a dummy target located 30 m east of the CPC entrance in mid-air. This double-measurement approach was used to determine the shading/blocking losses from the net power delivered by the heliostats.

The solar power input of each heliostat was calculated using Equation (3.1) and enthalpy values of water [30]:

$$\dot{Q}_\text{in} = \dot{m}_w (h_{\text{out},w} - h_{\text{in},w})$$  \hspace{1cm} (3.1)

Figure 3-8 – Piping and instrumentation diagram of the water calorimeter setup
\( \dot{Q}_\text{in} \) exhibited significant variation throughout the day, mainly due to the intermittency of the DNI and the tracking heliostat field, \textit{cf.} Figure 3-10. The latter manifests itself in varying cosine losses, shading/blocking, spillage due to astigmatic aberration effects, and tracking accuracy \cite{31}. Secondary effects were wind, path attenuation, and mirror soiling. Furthermore, the CPCs reflectivity was deteriorated due to age and dust deposition. To calculate \( \dot{Q}_\text{in} \), a Monte Carlo ray-tracing (MC) model was developed and experimentally validated with the calorimetric measurements.

### 3.4 Monte Carlo ray-tracing model

A 3D MC simulation using an in-house code \cite{32} was applied to numerically calculate the radiative heat transfer from each heliostat through the CPC into the calorimeter aperture. Figure 3-9 shows the MC model domain. Uniformly distributed sunrays subtending a solid half angle of 4.65 mrad are incident on the heliostats from the sun position at a given time and day, determined from known sun-path correlations \cite{33}. Tracking heliostats are modeled as two specularly reflecting sectors of a sphere with rectangular outline and paraxial focus located at the CPC inlet. The CPC was modelled by eight conical sections approximating its theoretical \( \theta_{\text{in}}/\theta_{\text{out}} \) shape \cite{20}. Non-tracking heliostats neighboring the tracking heliostats and the solar tower were modeled as absorbing black rectangles to fully assess shading/blocking effects, \textit{cf.} Figure 3-2 for the full heliostat field layout.

The heliostat reflectivity, \( \rho \), and a Rayleigh-distributed angular surface dispersion error, \( \sigma \), were adjusted individually per heliostat, while the reflectivity of the secondary concentrator \( \text{\textit{i.e.} CPC} \) was set to 0.6 with a probability of diffuse reflection of 0.1, \textit{cf.} Figure A-6 in the Appendix.
The agreement between the calorimetrically measured and the simulated radiative power inputs was ±8.7%. Because of aging after >30 years of operation, the heliostat tracking system caused an offset between the heliostat’s projected image and the target. This aiming offset had to be corrected manually prior to experimentation, which constituted a significant non-quantifiable source of error, cf. Appendix B. To improve accuracy, calorimeter measurements were also carried out directly on the solar receiver under cold conditions using air at maximum mass flow rate, referred to as calibration points. The calibration points were used to accurately shift the $\dot{Q}_{in}$ determined by MC to match the daily manual offset correction, cf. Figure 3-10. It is noted that peak $\dot{Q}_{in}$ occurs around 15:00 instead of noon, because the CPC’s optical axis points to the North-east, cf. Figure 3-2.
Figure 3-10 – Exemplary plot of the solar radiative power input vs. local time for mid-April. The solid line represents the calibrated $\dot{Q}_{in}$, using two calibration points (diamonds), while the dashed line represents $\dot{Q}_{in}$ predicted by the MC model.
4 Experimental results

The pressure drop, $\Delta p$, outlet air temperature, $T_{out}$, the corresponding thermal power, $Q_{th}$, and finally the resulting thermal efficiency, $\eta_{th}$, are of main importance for determining the solar receiver performance. A steady-state criterion for all variables $\Phi$ (e.g. $T$, $\dot{m}$, $Q_{th}$) was defined as:

$$\Delta \Phi(\Delta t) \leq 1\%$$ (4.1)

for $\Delta t \geq 60$ s. Measurement fluctuations were accounted for in the error bars of the efficiency plots. Figure 4-1 shows a typical experimental run at the 2 bar pressure level on a clear day with peak $I_{DNI} = 911$ W/m². With $\dot{m}$ set to maximum, $\dot{Q}_{in}$ was increased step-wise by introducing the heliostats one by one from 11:20 to 11:50. The two calibration points are at 11:50 and 15:00 at conditions with maximum $\dot{m}$, cf. previous Section. $T_{out}$ was increased by reducing $\dot{m}$ step-wise. The peak temperature registered during that particular run was 1090 °C.

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3 Material of this Chapter has been published in:
Experimental results

Figure 4-1 – Representative experimental run at \( p = 2 \text{ bar} \). The heliostats were introduced one by one after 11:20. The two calibration points are at 11:50 and 15:00.

4.1 Pressure drop

\( \Delta p = p_{\text{in}} - p_{\text{out}} \) was measured across the RPC, *cf.* Figure 3-4. The RPC is the core heat transferring element of the solar receiver and thus, its effect on \( \Delta p \) is of major interest. As expected, the three investigated RPC configurations (10 PPI, 20 PPI, and 10 PPI + baffles) showed different pressure drop characteristics. Figure 4-2 shows \( \Delta p \) across the RPC as a function of \( \dot{m} \) at the three investigated pressure levels for all three RPC configurations. Peak measured \( \Delta p \) was 54 ±4.8 mbar and occurred at \( \dot{m} = 117.4 \text{ g/s} \) and \( p = 2 \text{ bar} \) with the 20 PPI configuration. This extreme value corresponds to a static pressure loss of 2.7%, which is still relatively low compared to turbine combustor units where usually 2 – 8% of static pressure is lost [5]. As expected, \( \Delta p \) decreases with increasing \( p \) and decreasing \( \dot{m} \) due to its inherent dependency on the fluid velocity, \( v^2 \), as shown in Figure 4-3. Due to high noise in the \( p \)-transmitter signals, no error bars are indicated in the following Figures for better readability.
Figure 4-2 – Pressure drop across the RPC vs. mass flow rate for the three RPC configurations: 10 PPI, 20 PPI, and 10 PPI + baffles (BAF).

Figure 4-3 – Pressure drop across the RPC vs. the fluid velocity for the three RPC configurations: 10 PPI, 20 PPI, and 10 PPI + baffles (BAF). The dashed lines represent linear fits for the corresponding pressure levels.
A suitable approach to distinguish this dependency is to compare the pressure loss with the dynamic pressure of the flow, $p_{dyn}$. The pressure coefficient, $C_p$, is defined as:

$$ C_p = \frac{\Delta p}{p_{dyn}} = \frac{2 \cdot \Delta p}{\rho v^2} \quad (4.2) $$

The corrected mass flow, $\dot{m}_{corr}$, applied in compressor maps and turbine performance charts [34], reflects the mass flow that passes through a device (e.g. compressor, bypass duct, etc.) at atmospheric standard conditions at sea-level (DIN ISO 2533: 15 °C, 1 atm):

$$ \dot{m}_{corr} = \dot{m} \sqrt{\frac{T_{avg}}{288.15 \text{ K}}} \cdot \sqrt{\frac{p_{stat}}{1.01325 \text{ bar}}} \quad (4.3) $$

$$ T_{avg} = \frac{T_{in} + T_{out}}{2} \quad (4.4) $$

Figure 4-4 shows $C_p$ vs. $\dot{m}_{corr}$. $C_p$ is larger in the 20 PPI and the 10 PPI + baffles configurations compared to the 10 PPI case, while the five additional baffles have a similar effect as doubling the PPI. For all data points, $C_p$ increases by an average factor of $1.56 \pm 0.19$ when switching from 10 PPI to 20 PPI or the baffle-configuration. The plot suggests a transition from turbulent to laminar flow at $\dot{m}_{corr} \approx 0.02$. However, the corresponding Reynolds numbers were below 1000 and thus, this hypothesis cannot be sufficiently confirmed.
Figure 4-4 – Pressure coefficient across the RPC vs. corrected mass flow for the three RPC configurations: 10 PPI, 20 PPI, and 10 PPI + baffles (BAF)

4.2 Temperatures

Temperature $T_{\text{out}}$ is strongly dependent on $\dot{m}$, which was controlled by changing the bypass air flow ratio using the corresponding valves, cf. Figure 3-5. Figure 4-5 shows $T_{\text{out}}$ vs. $\dot{m}$. $\dot{Q}_{\text{in}}$ was at least 20 kW th, but with large variations between the data points, which makes comparison difficult. The peak $T_{\text{out}} = 1206 \pm 0.1 \, ^\circ\text{C}$ was obtained at $p = 2$ bar and $\dot{m} = 6.8 \, \text{g/s}$. For safety reasons, higher pressures were not realized for $T_{\text{out}} > 1100 \, ^\circ\text{C}$, as pressure vessel surface temperatures, $T_s$, of up to 395 °C were recorded near the outlet flange, cf. Figure 4-6 and Table 4-1. Using all data points, $T_{\text{out}}$ can be correlated to $\dot{m}$ by a power law, with $R^2 = 92.72\%$:

$$T_{\text{out}} = 29600 - 27800\dot{m}^{0.01136} \quad (4.5)$$
Experimental results

Figure 4-5 – Outlet air temperature vs. air mass flow rate for the three RPC configurations: 10 PPI, 20 PPI, and 10 PPI + baffles (BAF). The dashed line indicates the approximate trend by a power law fit, cf. Equation (4.5). The error bars are within the size of the markers.

Alternatively, a double exponential fit was found, which results in the same $R^2 = 92.72\%$:

$$T_{out} = 632.2e^{-0.1067m} + 957.7e^{-0.01127m}$$ (4.6)

Figure 4-6 shows a characteristic thermogram of the receiver pressure vessel during on-sun testing. The hottest part was always located at the smallest diameter of the reducer cone. Peak surface temperature recorded was $T_s = 395$ °C. A second hot spot was spotted on the top part of the pressure vessel.
Knowing characteristic peak outlet air temperatures and pressures allows comparisons. Table 4-1 summarizes peak temperatures of the outlet air and the receiver surface and also gives a short comparison to existing receiver designs. For a detailed list of alternative receiver designs, cf. Ref. [11].

Table 4-1 – Peak receiver outlet air and surface temperatures

<table>
<thead>
<tr>
<th>Receiver</th>
<th>$p$ (bar)</th>
<th>$T_{out}$ (°C)</th>
<th>$T_s$ (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>ETH (3 kW$_{th}$)</td>
<td>5</td>
<td>1062$^a$</td>
<td></td>
</tr>
<tr>
<td>ETH (50 kW$_{th}$)</td>
<td>2</td>
<td>1206</td>
<td>395</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>1104</td>
<td>272</td>
</tr>
<tr>
<td></td>
<td>6</td>
<td>960</td>
<td>228</td>
</tr>
<tr>
<td>DIAPR$^*$</td>
<td>20</td>
<td>1201$^b$</td>
<td></td>
</tr>
<tr>
<td>REFOS$^*$</td>
<td>15</td>
<td>800$^c$</td>
<td></td>
</tr>
<tr>
<td></td>
<td>6.5</td>
<td>960$^d$</td>
<td></td>
</tr>
</tbody>
</table>

$^a$ Ref. [18]; $^b$ Ref. [13]; $^c$ Ref. [14]; $^d$ Ref. [35]

$^*$ designs with window

Figure 4-6 – Characteristic IR thermogram of the solar receiver during on-sun operation for $T_{out} = 816$ °C
The ETH receiver was operated at the highest reported temperature and also exhibited the highest temperature difference, $\Delta T$. Depending on the envisioned power cycle, a certain combination of high temperature and pressure is required, cf. Chapter 7. Thus, a direct evaluation of the receiver technologies is difficult. Anyhow, alleviating the window is considered as major advancement.

To underline the effects of the receiver alignment, Figure 4-7 displays $T^+$ vs. $T^-$, cf. Figure 3-4. Due to the almost horizontal mounting of the receiver, the upper part of the RPC always exhibited higher temperatures than the lower part. This effect could be reduced by using the 10 PPI + baffles RPC configuration forcing the air flow into an up-and-down flow pattern.

![Figure 4-7 – RPC temperature at top vs. bottom of the three RPC configurations: 10 PPI, 20 PPI, and 10 PPI + baffles (BAF), cf. Figure 3-4. No error bars are shown due to the unknown uncertainty of the unshielded thermocouples.](image-url)
4.3 Efficiencies

The solar receiver performance is given by its thermal efficiency, $\eta_{th}$, defined in Equation (4.7) as the enthalpy change of the air flow, $\dot{Q}_{th}$, using enthalpy values of air from Ref. [36], divided by the solar radiative power input through the aperture, $\dot{Q}_{in}$:

$$\eta_{th} = \frac{\dot{Q}_{th}}{\dot{Q}_{in}} = \frac{\dot{m}(h_{out} - h_{in})}{\dot{Q}_{in}}$$  \hfill (4.7)

Thus, $\eta_{th}$ gives the capability of the solar receiver to absorb and convert concentrated solar energy into high-temperature heat contained in the air flow.

Figure 4-8 – Figure 4-10 show $\eta_{th}$ as a function of $T_{out}$, $p$, and $\dot{m}$ for the three RPC configurations. $\dot{m}$ is calculated from the inverse fit of $T_{out}$ vs. $\dot{m}$, cf. Figure 4-5 and Equation (4.5). $\eta_{th}$ decreases with increasing $T_{out}$ because of increased heat losses, especially reradiation through the aperture. This decrease becomes significant for $T_{out} > 800$ °C as $\eta_{th}$ drops to an average of 60% for all configurations and pressures, and to 25% when $T_{out}$ reaches ~1200 °C. The scattering in the values of $\eta_{th}$ is attributed to varying heat losses at different $\dot{Q}_{in}$. 
Experimental results

Figure 4-8 – Thermal efficiency vs. outlet air temperature for the 10 PPI RPC configuration, with $\dot{m}$ from Equation (4.5)

Figure 4-9 – Thermal efficiency vs. outlet air temperature for the 20 PPI RPC configuration, with $\dot{m}$ from Equation (4.5)
Figure 4-10 – Thermal efficiency vs. outlet air temperature for the 10 PPI + baffles RPC configuration, with $\dot{m}$ from Equation (4.5)

Figure 4-11 shows $\eta_{th}$ vs. the specific solar radiative energy input, $q_{in}$, defined as $\dot{Q}_{in}$ normalized by $\dot{m}$, which results in reduced variance in $\eta_{th}$:

\[
q_{in} = \frac{\dot{Q}_{in}}{\dot{m}}
\]  

(4.8)

The corresponding $T_{out}$ can be estimated by multiplying $q_{in}$ by $\eta_{th}$ while assuming a constant air heat capacity, $c_{p,air} = 1$ kJ/(kg·K), and neglecting $T_{in} = T_{amb}$. Similarly, Figure 4-12 depicts $T_{out}$ vs. $q_{in}$, in which $\eta_{th}$ can be estimated by dividing $T_{out}$ by $q_{in}$. In both Figures, the manually added blue bands approximate the envelopes of all data. In Figure 4-12 an accurate logarithmic fit with $R^2 = 95.5\%$ was found:

\[
T_{out} = 311.31 \cdot \ln(q_{in}) - 1474.6
\]  

(4.9)
Experimental results

Figure 4-11 – Thermal efficiency vs. specific solar radiative energy input for the three RPC configurations: 10 PPI, 20 PPI, and 10 PPI + baffles (BAF) and blue envelope approximation. For better readability, error bars were omitted.

Figure 4-12 – Outlet air temperature vs. specific solar radiative energy input for the three RPC configurations: 10 PPI, 20 PPI, and 10 PPI + baffles (BAF) and blue envelope approximation. The dashed line indicates a logarithmic fit, cf. Equation (4.9). For better readability, error bars were omitted.
The lower $\eta_{th}$ at higher $T_{out}$ is compensated by the higher exergy value of the high-temperature air flow. This can be expressed through the ideal solar heat engine efficiency, $\eta_{solar HE}$, defined as $\eta_{th}$ multiplied by the Carnot efficiency of an ideal heat engine operating between $T_{out}$ and ambient temperature, $T_{amb} = 25$ °C:

$$\eta_{solar HE} = \eta_{th} \cdot \left(1 - \frac{T_{amb}}{T_{out}}\right) \quad (4.10)$$

Figure 4-13 – Figure 4-15 show $\eta_{solar HE}$ as a function of $T_{out}$, $p$, and $\dot{m}$ for the three RPC configurations. $\eta_{solar HE}$ peaks at 63.3%, which corresponds to $\eta_{th} = 91.2\%$ for $T_{out} = 699$ °C and $p = 4$ bar in the 20 PPI configuration.

Figure 4-16 shows $\eta_{solar HE}$ vs. $T_{out}$ for all data points and a trend line. Also shown is $\eta_{Carnot}$ for an ideal solar receiver without any losses, i.e. $\eta_{th} = 1$. Up to ~900 °C, a very high $\eta_{th}$ sustains the high exergetic performance. Most potential for improvement is found in the upper temperature range: even slight improvements of $\eta_{th}$ can lead to high gains in $\eta_{solar HE}$, indicated by the increasing gap between the two lines.
Figure 4-13 – Ideal solar heat engine efficiency ($\eta_{\text{th}} \times \eta_{\text{Carnot}}$) vs. outlet air temperature for the 10 PPI RPC configuration, with $\dot{m}$ from Equation (4.5)

Figure 4-14 – Ideal solar heat engine efficiency ($\eta_{\text{th}} \times \eta_{\text{Carnot}}$) vs. outlet air temperature for the 20 PPI RPC configuration, with $\dot{m}$ from Equation (4.5)
Figure 4-15 – Ideal solar heat engine efficiency ($\eta_{th} \times \eta_{Carnot}$) vs. outlet air temperature for the 10 PPI + baffles RPC configuration, with $\dot{m}$ from Equation (4.5).

Figure 4-16 – Ideal solar heat engine efficiency ($\eta_{th} \times \eta_{Carnot}$) vs. outlet air temperature, with $\dot{m}$ from Equation (4.5). The solid line shows the theoretical maximum (i.e. Carnot efficiency) for $T_{amb} = 25$ °C, while the dashed line represents a 3$^{rd}$-order polynomial fit.
Focusing on the $T_{\text{out}}$ target range of 700 – 900 °C, Table 4-2 summarizes the average $\eta_{\text{th}}$ and $\eta_{\text{solar HE}}$ for the three RPC configurations and the three pressure levels. The peak average $\eta_{\text{th}} = 74.3\%$, 60.8\%, and 61.1\%, and the corresponding $\eta_{\text{solar HE}} = 53.7\%$, 43.3\%, and 43.9\% were obtained at average $T_{\text{out}} = 804$, 777, and 805 °C at $p = 6$, 2, and 4 bar for the 10 PPI, 20 PPI, and the 10 PPI + baffles configurations, respectively. For all configurations and pressure levels, the average $\eta_{\text{th}} = 60.2\%$ and $\eta_{\text{solar HE}} = 43.8\%$ were obtained at an average $T_{\text{out}} = 799$ °C.

As seen in the pressure level average, increasing $p$ has a beneficial effect on the efficiency, attributed to the longer residence time of the air inside the receiver. It is noted that $T_{\text{in}} = T_{\text{amb}}$ and pressurization does not lead to an increase in $T_{\text{in}}$. Similarly, the configuration average of the 10 PPI configuration displays higher efficiencies compared to the other RPC configurations. Most probably the other two configurations suffer from more air that flows through gaps between RPC and insulation (cf. Figure 2-2) because of the increased flow resistance when using baffles or smaller pore size (20 PPI). It is noted that the average values in the 700 – 900 °C target range do not fully reflect the performance in the whole spectrum of possible operating conditions.

Table 4-3 lists the peak values of the solar radiative power input, $\dot{Q}_{\text{in}}$, air enthalpy change, $\dot{Q}_{\text{th}}$, outlet temperature, $T_{\text{out}}$, pressure drop, $\Delta p$, system pressure, $p$, direct normal irradiance, $I_{\text{DNI}}$, and solar concentration ratio, $C$.

Throughout this analysis, the ±8.7\% error in $\dot{Q}_{\text{in}}$ is the main source of uncertainty in the determination of $\eta_{\text{th}}$ and $\eta_{\text{solar HE}}$. Non-quantifiable errors of particular importance are the manual heliostat aiming offset adjustments and wind, both affecting the heliostat tracking accuracy, cf. Appendix B.
Table 4-2 – Average thermal efficiency and ideal heat engine efficiency ($\eta_{\text{th}} \times \eta_{\text{Carnot}}$) for all data points with $T_{\text{out}}$ in the range 700 – 900 °C and $T_{\text{amb}} = 25$ °C. Each cell contains the average efficiencies and temperature for a specific RPC configuration and pressure level. The values in brackets indicate the number of data points available to calculate the averages.

<table>
<thead>
<tr>
<th>Configuration→ $p$-level ↓</th>
<th>a) 10 PPI</th>
<th>b) 20 PPI</th>
<th>c) 10 PPI + baffles</th>
<th>$p$-level average</th>
</tr>
</thead>
<tbody>
<tr>
<td>2 bar</td>
<td>$\eta_{\text{th}}$</td>
<td>69.0%</td>
<td>60.8%</td>
<td>44.3%</td>
</tr>
<tr>
<td></td>
<td>$\eta_{\text{solar HE}}$</td>
<td>50.0%</td>
<td>43.3%</td>
<td>31.1%</td>
</tr>
<tr>
<td></td>
<td>$T_{\text{out}}$</td>
<td>818 °C (5)</td>
<td>777 °C (4)</td>
<td>736 °C (2)</td>
</tr>
<tr>
<td>4 bar</td>
<td>$\eta_{\text{th}}$</td>
<td>58.9%</td>
<td>59.4%</td>
<td>61.1%</td>
</tr>
<tr>
<td></td>
<td>$\eta_{\text{solar HE}}$</td>
<td>46.8%</td>
<td>42.9%</td>
<td>43.9%</td>
</tr>
<tr>
<td></td>
<td>$T_{\text{out}}$</td>
<td>804 °C (9)</td>
<td>807 °C (5)</td>
<td>805 °C (2)</td>
</tr>
<tr>
<td>6 bar</td>
<td>$\eta_{\text{th}}$</td>
<td>74.3%</td>
<td>55.7%</td>
<td>58.1%</td>
</tr>
<tr>
<td></td>
<td>$\eta_{\text{solar HE}}$</td>
<td>53.7%</td>
<td>40.4%</td>
<td>42.2%</td>
</tr>
<tr>
<td></td>
<td>$T_{\text{out}}$</td>
<td>804 °C (3)</td>
<td>818 °C (4)</td>
<td>824 °C (2)</td>
</tr>
<tr>
<td>Config. average</td>
<td>$\eta_{\text{th}}$</td>
<td>67.4%</td>
<td>58.6%</td>
<td>54.5%</td>
</tr>
<tr>
<td></td>
<td>$\eta_{\text{solar HE}}$</td>
<td>50.2%</td>
<td>42.2%</td>
<td>39.1%</td>
</tr>
<tr>
<td></td>
<td>$T_{\text{out}}$</td>
<td>809 °C (17)</td>
<td>801 °C (13)</td>
<td>788 °C (6)</td>
</tr>
</tbody>
</table>

Table 4-3 – Summary of experimental peak values

<table>
<thead>
<tr>
<th>Operating hours (h)</th>
<th>128</th>
</tr>
</thead>
<tbody>
<tr>
<td>On-sun operating hours (h)</td>
<td>93</td>
</tr>
<tr>
<td>Steady-state measurements$^a$</td>
<td>168</td>
</tr>
<tr>
<td>$\dot{Q}<em>{\text{in}}$ (kW$</em>{\text{th}}$)</td>
<td>46.6</td>
</tr>
<tr>
<td>$\dot{Q}<em>{\text{th}}$ (kW$</em>{\text{th}}$)</td>
<td>38.7</td>
</tr>
<tr>
<td>$T_{\text{out}}$ (°C)</td>
<td>1206</td>
</tr>
<tr>
<td>$\Delta p$ (mbar)</td>
<td>53.96</td>
</tr>
<tr>
<td>$p$ (bar)</td>
<td>6.05</td>
</tr>
<tr>
<td>$I_{\text{DNI}}$ (W/m$^2$)</td>
<td>977</td>
</tr>
<tr>
<td>$C$ (suns)</td>
<td>3012</td>
</tr>
</tbody>
</table>

$^a$ valid steady-states from a total of 206 steady-state conditions, cf. Equation (4.1)
4.4 Summary

In the previous two Chapters, the design, fabrication, and experimental testing of a 50 kW$_{th}$ pressurized air solar receiver was presented. The solar receiver was safely operated for a total of 93 hours under 168 steady-state conditions. Three configurations of the RPC were tested: (a) 10 PPI, (b) 20 PPI, and (c) 10 PPI with five air flow guiding baffles. A peak outlet air temperature of 1206 °C was reached at an air mass flow rate of 6.8 g/s. The most promising performance was achieved with the 20 PPI RPC configuration at $\dot{m} = 52.8$ g/s and $p = 4$ bar, yielding 91.2% thermal receiver efficiency at $T_{out} = 699$ °C and leading to a peak ideal solar heat engine efficiency ($\eta_{th} \times \eta_{\text{Carnot}}$) of 63.3%. Special focus was given to the 700 – 900 °C outlet air temperature range where higher pressure and lower flow resistance lead to higher efficiencies. In general, thermal efficiencies of the solar receiver exceeding 80% were obtained at outlet air temperatures that were considerably higher than those of superheated steam in Rankine cycles, which indicates the high potential for competitive solar-to-electricity efficiencies.
5 Numerical heat and mass transfer model

In Chapters 2 – 4 the design, fabrication, and experimental testing of a modular solar receiver for 50 kWth solar radiative power input was presented. In this Chapter, a rigorous numerical heat transfer model is elaborated to simulate the operation of the solar receiver. The model is based on previous 2D modeling efforts [18] and expands into 3D to improve accuracy. It incorporates the effective transport properties of the RPC foam, derived by direct pore-level simulations (DPLS) of the Navier-Stokes equations using the exact 3D digital geometry measured by micro-tomography. It further incorporates spatially resolved radiative flux distributions obtained by ray-tracing the solar concentrating optics in the MC simulation described in Section 3.4. The model refinements enable to extrapolate and analyze an extended range of operating conditions, which could not be achieved experimentally: namely, outlet air temperatures beyond 1200 °C and pressures exceeding 6 bar.

The model was validated using the experimental data presented in the previous Chapter. The validated model was applied to obtain a receiver performance map through a parametric study on the operating pressure, temperature, and mass flow rate, complemented by a sensitivity analysis of material properties and geometric dimensions.

The heat transfer model applied in this study consists of two coupled simulation codes: (a) a MC simulation to compute the solar radiative power input through the aperture, $Q_{in}$, and the absorbed solar flux inside the cavity, $q_{abs}$, and (b) a finite volume (FV) simulation to compute the temperature distribution throughout the receiver, its thermal losses, and the useful thermal power delivered to the working fluid, $Q_{th}$. For the purpose of model validation, the material of this Chapter has been published in:

dimensions, material properties, and operating conditions are consistent with those of the experimental study conducted at the WIS solar tower. $10^7$ rays per heliostat were used to obtain a good resolution ($3 \times 3$ mm) of the distribution of $\dot{q}_{abs}''$. The total absorbed radiative power on the surface of the cavity is:

$$ Q_{abs} = \int \dot{q}_{abs}'' dA $$

in which the integral is substituted by sums over the discretized flux map resulting from the numerical MC analysis.

### 5.1 CFD model

Figure 5-1 shows the FV model domain. It consists of two fluid subdomains: (a) the stagnant buoyancy-driven air inside CPC and cavity, and (b) the pressurized air flowing across the RPC. The RPC itself is modeled as a porous subdomain. The remaining subdomains are solid, representing the cavity, insulation, pressure vessel, and CPC. The geometry is simplified, i.e. no flanges, sensor ports, and stacked rings are modelled, cf. Figure 2-2.

![Figure 5-1 – Finite volume model domain indicating the fluid, porous, and solid subdomains, heat transfer modes, and air flow](image-url)
The air surrounding the receiver is not modeled; thus, thermal losses on the pressure vessel surface are tackled by a heat sink, which incorporates both radiative and convective heat transfer losses, \( \dot{q}_{\text{radiation, amb}} \) and \( \dot{q}_{\text{convection, amb}} \), respectively. Following Stefan-Boltzmann law and a Nusselt correlation for natural convection, Equation (5.2) and Equation (5.3) represent the area specific radiative and convective losses, respectively. The total losses are calculated similar to Equation (5.1). The Nusselt correlation with Nu based on the flange diameter, \( D \), was determined for the exact receiver geometry in a separate CFD simulation described in Appendix C.

\[
\dot{q}_{\text{radiation, amb}} = \sigma \left( T_s^4 - T_{\text{amb}}^4 \right) \quad (5.2)
\]

\[
\dot{q}_{\text{convection, amb}} = \frac{\text{Nu}_D \cdot \kappa_{\text{air, amb}}}{D} \left( T_s - T_{\text{amb}} \right) \quad (5.3)
\]

The governing mass, momentum and energy conservation equations were solved numerically using the CFD code Ansys CFX 15.0 with ~460'000 grid elements. The convergence criterion defined in Equation (5.4) was met for all parameters, \( \Phi \), except \( \dot{Q}_{\text{convection, cav}} \).

\[
\frac{\Phi_i - \Phi_{i-100}}{\Phi_i} \leq 0.5\% \quad (5.4)
\]

Convergence was reached after \( i = 2000 – 6000 \) iterations, depending on the operating conditions and geometry, cf. Section 6.4. The converged simulation yielded energy residuals <10\(^{-2}\) and <10\(^{-5}\); and flow residuals <10\(^{-5}\) and <10\(^{-6}\) in the ambient and pressurized air domain, respectively; and imbalances <10\(^{-2}\). In order to improve the convergence of the ambient domain, namely \( \dot{Q}_{\text{convection, cav}} \), ~30'000 iterations would have been necessary but the change in temperature distributions and heat transfer rates stayed well below 0.5%, so 2000 – 6000 iterations were considered a good compromise. Computation was carried out on the ETH high-performance computing cluster Brutus using 32 CPUs and requiring up to 4 h of computational time for 2000 iterations.
5.2 Material properties

Temperature dependent material properties of the air, SiC cavity, Al₂O₃-SiO₂ insulation, and 316L stainless steel pressure vessel are summarized in Appendix D. The effective transport properties of the SiC-based RPC are listed in Table 5-1 and have been determined by DPLS. This combined experimental-numerical methodology is described in detail in Ref. [37].

Figure 5-2 depicts a micro-tomography scan of a 10 PPI RPC foam sample with voxel edge length of 20 μm. The effective thermal conductivity, \( k_{\text{eff RPC}} \), is expressed as a linear combination of thermal conductivities of serial and parallel slabs [38] with serial-to-parallel slab ratio, \( f = 0.3120 \):

\[
\frac{k_{\text{eff RPC}}}{k_s} = f \left( \phi \cdot \frac{k_f}{k_s} + 1 - \phi \right) + (1 - f) \left( \phi \cdot \frac{k_s}{k_f} + 1 - \phi \right)^{-1} \tag{5.5}
\]

using the porosity, \( \phi \), and the thermal conductivities of solid and fluid, \( k_s \) and \( k_f \), respectively. The effective interfacial heat transfer coefficient of the RPC is given by the Nu correlation:

\[
\text{Nu}_{d_{\text{nom}}} = 8.1647 + 0.2662 \cdot \text{Re}_{d_{\text{nom}}}^{0.8052} \cdot \text{Pr} \tag{5.6}
\]

where Re and Nu are based on the RPC’s nominal pore diameter, \( d_{\text{nom}} \). While Equation (5.5) and Equation (5.6) in conjunction with the material properties treat the conductive and convective heat transfer inside the RPC, the radiative heat transfer within the porous domain was modeled by the P₁ radiation model [39] in an emitting, absorbing, and non-scattering medium with absorption coefficient, \( \beta = 220 \text{ m}^{-1} \) [40].
Figure 5-2 – 3D surface rendering of a 10 PPI RPC foam sample used in the solar receiver prototype with voxel edge length of 20 μm

Table 5-1 – Material properties of the SiC-based RPC foam determined by DPLS

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\phi$</td>
<td>0.85812</td>
<td>-</td>
</tr>
<tr>
<td>$s^a$</td>
<td>852</td>
<td>m$^{-1}$</td>
</tr>
<tr>
<td>$d_{\text{nom}}$</td>
<td>2.47</td>
<td>mm</td>
</tr>
<tr>
<td>$f$</td>
<td>0.3120</td>
<td>-</td>
</tr>
</tbody>
</table>

$^a$ specific surface area

5.3 Model validation

The MC model was validated using water calorimetry measurements resulting in a model accuracy of ±8.7% in $\dot{Q}_{\text{in}}$, cf. Sections 3.3 and 3.4. From the known $\dot{Q}_{\text{in}}$ follows $q_{\text{abs}}^*$, which was directly applied to the FV model as a source term on the cavity surface, cf. Figure 5-1. Figure 5-3 shows a parity plot of numerically calculated and experimentally measured outlet air temperatures, $T_{\text{out}}$. 
The model agreement is within the inaccuracy in $\dot{Q}_{in}$ (indicated by dashed lines) up to $\sim 850 \, ^\circ\text{C}$, after which the model overpredicts $T_{out}$ by up to 16%, mainly because of the wind effects during those experimental data. While the model assumes stagnant buoyancy-driven air trapped inside the downward facing cavity, the experiment was affected by gusts of wind at the CPC inlet, which could neither be prevented nor monitored, cf. Appendix B. The measured static pressure loss across the solar receiver was less than 2.7% (cf. Section 4.1), which is considerably smaller than typical values of combustors and thus, $\Delta p$ is not treated any further [5].

![Figure 5-3](image.png)

Figure 5-3 – Parity plot of numerically calculated vs. experimentally measured values of $T_{out}$. The dashed lines indicate the $\pm 8.7\%$ inaccuracy attributed to the error in $\dot{Q}_{in}$.

After model validation, $\dot{q}_{abs}''$ was circumferentially averaged and normalized to $\dot{Q}_{in} = 50 \, \text{kW}_{th}$ in order to simulate an idealized time-independent solar power input, cf. Figure 5-4. The first peak in $\dot{q}_{abs}''$ at $x \approx 75 \, \text{mm}$ is caused by the given CPC geometry with $\theta_{out} = 65^\circ$, cf. Section 3.1. Also, direct irradiation from the heliostat field is absorbed at the dome-end of the cavity, accounts for $\sim 18\%$ of $\dot{Q}_{abs}$, and starts at $x = 375 \, \text{mm}$ where the curvature of the dome-end meets the cylindrical part of the cavity, cf. Figure 5-1. It is noted that the $x$-axis is displayed reversely to be consistent with the $x$-direction previously defined in Figure 5-1.
5.4 Inlet conditions

Opposed to the experiments, in which pressurized air at ambient temperature was fed to the receiver, the simulation is supposed to analyze the situation where air is heated due to compression (e.g. in a Brayton cycle). For non-ideal compression of an ideal gas [34]:

\[ T_{in} = T_{amb} \left( \frac{p_{in}}{p_{amb}} \right)^{\frac{y-1}{y \eta_c}} \]  

where \( T_{amb} \) and \( p_{amb} \) are at ambient conditions, \( T_{in} \) and \( p_{in} \) are at the inlet conditions to the solar receiver, and \( \eta_c \) is the isentropic compressor efficiency of a multi-stage compressor. For industrial compressors, \( \eta_c = 0.88 – 0.92 \) is reported [5]; thus, an average value of \( \eta_c = 0.9 \) was chosen to determine \( T_{in} \) for the simulation cases. For compression ratios below 15 and \( T_{amb} = 25 \degree C \), the perfect gas assumption does not deviate from values of dry air [34].

Figure 5-4 – Circumferentially averaged flux distribution for \( \dot{Q}_{in} = 50 \) kW along the \( x \)-direction of the cavity
Table 5-2 summarizes the investigated pressure levels and the resulting $T_{in}$ for $\gamma = 1.4$, $p_{amb} = 1.013$ bar, and $T_{amb} = 25$ °C.

Table 5-2 – Inlet air temperatures after compression with $\gamma = 1.4$, $\eta_c = 0.9$, $p_{amb} = 1.013$ bar, and $T_{amb} = 25$ °C

<table>
<thead>
<tr>
<th>$p$ (bar)</th>
<th>$T_{in}$ (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>4</td>
<td>188</td>
</tr>
<tr>
<td>6</td>
<td>251</td>
</tr>
<tr>
<td>8</td>
<td>301</td>
</tr>
<tr>
<td>10</td>
<td>343</td>
</tr>
<tr>
<td>15</td>
<td>428</td>
</tr>
</tbody>
</table>
6 Model results

The pressure drop, $\Delta p$, the outlet air temperature, $T_{out}$, and the resulting thermal efficiency, $\eta_{th}$, are of major interest. As shown in Section 4.1, $\Delta p$ was very small compared to combustors in gas turbines, thus $\Delta p$ is not a critical parameter and is not further discussed. Furthermore, the RPC and insulation domain are each modeled as one piece, opposed to the experiments where they consisted of stacked rings with gaps and a staircase-shaped curvature approximation at the dome-end (cf. Figure 2-2 and Figure 5-1), which caused inaccurate model predictions of $\Delta p$, compared to the experiments.

It is noted that the simulation results for $T_{out}$ and consequently $\eta_{th}$ presented in this Chapter are not compatible with the experimentally obtained values because: (a) $\dot{Q}_{in}$ is normalized to 50 kWth, (b) $\dot{q}_{abs}^{n}$ is circumferentially averaged (cf. Figure 5-4), and (c) the simulated $T_{in}$ is pressure dependent (cf. Table 5-2).

6.1 Temperatures

Figure 6-1 shows $T_{out}$ as a function of $\dot{m}$, for an absolute pressure, $p = 4, 6, 8, 10, \text{ and } 15$ bar and $\dot{Q}_{in} = 50$ kWth. $\dot{m}$ was varied from 20 – 180 g/s, resulting in $T_{out}$ reaching up to 1420 °C in the 15 bar case. Figure 6-2 shows the excess temperature, $\tau$, as a function of $T_{out}$. $\tau$ represents the normalized excess temperature occurring in the cavity that is not converted into useful energy:

$$\tau = \frac{T_{cav} - T_{out}}{T_{cav}} \quad (6.1)$$

Figure 6-1 – Outlet air temperature vs. mass flow rate at five pressure levels for $\dot{Q}_m = 50$ kW. The dashed lines represent 5th-order polynomial fits, while the solid line shows the fit from Equation (4.5) for non-uniform $\dot{Q}_m$ and $T_m = T_{\text{amb}}$, cf. Figure 4-5.

Figure 6-2 – Normalized excess temperature vs. outlet air temperature for $\dot{Q}_m = 50$ kW. The dashed line indicates a linear trend, while the solid line represents the SiC material temperature limit of 1600 °C.
For all cases, \( \tau \) is above 0.103, which indicates that conduction across the cavity is the limiting heat transfer mode. The average cavity temperature, \( \overline{T_{\text{cav}}} \), exceeds the material limit of 1600 °C for SiC [25] if \( T_{\text{out}} > 1345 \) °C is reached.

Figure 6-3 shows the temperature profile along the vertical cross-section of the receiver, for \( p = 15 \) bar, \( \dot{Q}_{\text{in}} = 50 \) kW, and \( \dot{m} = 60 \) g/s, yielding \( T_{\text{out}} = 970 \) °C. It is noted that within the porous domain, only the fluid temperature is displayed causing the adjacent insulation to appear at higher temperature than the air flow.

![Exemplary temperature distribution along the vertical cross-section of the receiver.](image)

Figure 6-3 – Exemplary temperature distribution along the vertical cross-section of the receiver, for \( p = 15 \) bar, \( \dot{Q}_{\text{in}} = 50 \) kW, and \( \dot{m} = 60 \) g/s. Within the porous domain only the fluid temperature is displayed.

### 6.2 Efficiencies

The performance of the solar receiver is defined by its thermal efficiency, \( \eta_{\text{th}} \), defined in Equation (4.7). Figure 6-4 shows \( \eta_{\text{th}} \) as a function of \( T_{\text{out}} \) for the five investigated pressure levels. As expected, \( \eta_{\text{th}} \) decreases with \( T_{\text{out}} \), mainly because of reradiation losses through the aperture. The peak \( \eta_{\text{th}} = 86.3\% \) is reached for the lowest \( T_{\text{out}} = 433 \) °C at 4 bar. \( p \) only has a small influence on \( \eta_{\text{th}} \) for low \( T_{\text{out}} \), while the effect becomes more pronounced at higher \( T_{\text{out}} \). This decrease in \( \eta_{\text{th}} \) at increasing \( p \) is attributed to the increasing \( T_{\text{in}} \) causing elevated
average receiver temperatures, which lead to increasing overall heat losses throughout the receiver. For the maximum $\bar{T}_{\text{cav}} = 1600 \, ^\circ\text{C}$, $\eta_{\text{th}} = 50.5\%$ is reached at $T_{\text{out}} = 1345 \, ^\circ\text{C}$.

Figure 6-4 – Thermal efficiency vs. outlet air temperature for five pressure levels

Figure 6-5 shows $\eta_{\text{th}}$ vs. the specific solar radiative energy input, $q_{\text{in}}$, defined in Equation (4.8). Assuming a constant $c_{p,\text{air}} = 1 \, \text{kJ/(kg·K)}$, the corresponding $\Delta T$ can be estimated by multiplying $q_{\text{in}}$ by $\eta_{\text{th}}$. Similarly, Figure 6-6 shows $\Delta T$ vs. $q_{\text{in}}$, in which $\eta_{\text{th}}$ can be estimated by dividing $\Delta T$ by $q_{\text{in}}$. Since this $q_{\text{in}}$-dependency allows comparison of experimental and simulation values, in both Figures, the envelope approximations from Figure 4-11 and Figure 4-12 were added and show reasonable agreement. It is noted that the envelope from Figure 4-12 had to be adjusted to reflect $\Delta T$ instead of $T_{\text{out}}$. Figure 6-5 and Figure 6-6 also show that for a given $q_{\text{in}}$, $\Delta T$ and consequently $\eta_{\text{th}}$ decreases with increasing $p$. However, $T_{\text{out}}$ still increases with increasing $p$ due to the increase in $T_{\text{in}}$, which manifests in Figure 6-7 (cf. Table 5-2).
Figure 6-5 – Thermal efficiency vs. specific solar radiative energy input at five pressure levels. The dashed lines represent exponential fits. The blue area shows the envelope approximation of the experimental values from Figure 4-11.

Figure 6-6 – Temperature difference vs. specific solar radiative energy input at five pressure levels. The dashed lines represent 5th-order polynomial fits. The blue area shows the envelope approximation of the experimental values from Figure 4-12, adjusted for $\Delta T$. 
Figure 6-7 – Outlet air temperature vs. specific solar radiative energy input at five pressure levels. The dashed lines represent 5th-order polynomial fits.

The lower $\eta_{th}$ at high $T_{out}$ is compensated by higher exergy values of the air flow at higher temperatures. This is expressed through $\eta_{solar \ HE}$, defined in Equation (4.10). Figure 6-8 shows $\eta_{solar \ HE}$ vs. $T_{out}$ for the five investigated pressure levels. $\eta_{solar \ HE}$ peaks at 53.8%, which corresponds to $\eta_{th} = 79.6\%$ for $T_{out} = 665 \degree C$. Regardless of $p$, the optimum operating range is found to be $\sim 600 – 900 \degree C$, while temperatures above $\sim 1200 \degree C$ should be avoided due to the steep decrease in $\eta_{th}$ and $\eta_{solar \ HE}$ and the proximity to the cavity material temperature limit, cf. Figure 6-2.
Figure 6-8 – Ideal solar heat engine efficiency ($\eta_{\text{th}} \times \eta_{\text{Carnot}}$) vs. outlet air temperature at five pressure levels. The solid line indicates the Carnot limit for $T_{\text{amb}} = 25$ °C, while the dashed line shows the trend line from Figure 4-16 for non-uniform $\dot{Q}_{\text{in}}$ and $T_{\text{in}} = T_{\text{amb}}$.

### 6.3 Heat losses

An energy balance was used to assess the overall energy transport throughout the whole receiver setup and to determine the heat losses. The main energy loss mechanisms are (cf. Figure 5-1): reflection losses at the cavity due to its non-perfect behavior as a blackbody, $\dot{Q}_{\text{reflection}}$, reradiation losses through the aperture, $\dot{Q}_{\text{reradiation}}$, convective losses at the cavity surface, $\dot{Q}_{\text{convection,cav}}$, conductive losses towards the water-cooled CPC (kept at 75 °C), $\dot{Q}_{\text{conduction}}$, and radiative and convective heat losses at the pressure vessel surface, $\dot{Q}_{\text{radiation,amb}}$ and $\dot{Q}_{\text{convection,amb}}$, respectively. All energy losses are summarized in Figure 6-9 as a function of $T_{\text{out}}$, with $\dot{Q}_{\text{th}}$ being the remaining thermal power available for conversion in a heat engine (e.g. Brayton cycle):

$$
\dot{Q}_{\text{th}} = \frac{\dot{Q}_{\text{in}} - \dot{Q}_{\text{reflection}}}{\dot{Q}_{\text{abs}}} - \dot{Q}_{\text{reradiation}} - \dot{Q}_{\text{convection,cav}} - \dot{Q}_{\text{conduction}} - \dot{Q}_{\text{radiation,amb}} - \dot{Q}_{\text{convection,amb}}
$$

(6.2)
Figure 6-9 – Contribution of heat loss mechanisms as a function of the outlet air temperature for $Q_{in} = 50$ kW

The plot shows that for all considered values of $T_{out}$, $\dot{Q}_{\text{reradiation}}$ represents the largest loss and accounts for $\sim 50\%$ of all the losses combined. To some extent, $\dot{Q}_{\text{reradiation}}$ can be reduced by a secondary concentrator designed for smaller acceptance angles, which would result in a smaller cavity aperture but requires changes in the heliostat field layout. $\dot{Q}_{\text{conduction}}$ was calculated for a 10 mm insulation layer between receiver front and CPC and contributed to up to 7$\%$ of the losses, which could be reduced by increasing the thermal resistance or thickness of the CPC insulation. Similarly, $\dot{Q}_{\text{radiation,amb}}$ and $\dot{Q}_{\text{convection,amb}}$, can be reduced by improving the insulation in radial direction. For a side-by-side arrangement in a solar receiver cluster design, $\dot{Q}_{\text{radiation,amb}}$ and $\dot{Q}_{\text{convection,amb}}$ would only occur at the edges of the cluster, which results in a significant reduction of those losses, cf. Section 7.4. $\dot{Q}_{\text{convection,cav}}$ was calculated for a downward facing receiver, tilted by $25^\circ$ from the horizontal. The impact of the tilt angle, $\alpha$, is later shown in the sensitivity analysis. Since the cavity does not behave as a perfect blackbody, a constant $\dot{Q}_{\text{reflection}} = 1.2\%$ was determined, which is equivalent to an apparent aperture emissivity of 0.988.
Figure 6-10 shows the distribution of the local air temperature, $T$, and the corresponding power, $\dot{Q}$, which is added to the air flow while following a streamline through the receiver. $T$ and $\dot{Q}$ exhibit a steep increase right when the air hits the hot RPC at $x \approx 0.06$ m and flatten after leaving the RPC at $x \approx 0.55$ m. It is observed that across the whole receiver length the heat is added at a very constant rate, which indicates that the receiver length was not chosen too long. The spatial derivative of $\dot{Q}$ follows $\dot{q}_{\text{abs}}''$, cf. Figure 5-4. It is noted that the performance along an arbitrary streamline differs from the total results and that the $x$-axis is displayed reversely to be consistent with the $x$-direction previously defined in Figure 5-1.

Figure 6-10 – Air temperature, heat addition to the air flow and its spatial derivative, and absorbed solar radiative flux along the path of a streamline through the receiver, with approximate linear time dependency for $p = 8$ bar, $\dot{m} = 100$ g/s, and $\dot{Q}_{\text{in}} = 50$ kW
6.4 Sensitivity and parameter analysis

The simulation for $p = 8$ bar, $\dot{m} = 100$ g/s, and $\dot{Q}_{in} = 50$ kW$_{th}$, which results in peak $\eta_{\text{solar HE}} = 53.8\%$ at $T_{out} = 665$ °C and $\eta_{th} = 79.6\%$ was chosen as reference case at baseline dimensions, cf. Table 6-1. The deviations of $\eta_{th}$ from this reference case were analyzed for varying material properties and geometric dimensions shown in Figure 6-11 and Figure 6-12, respectively. The relative property change is defined as:

$$\Delta \Phi = \frac{\Phi - \Phi_{\text{ref}}}{\Phi_{\text{ref}}} \quad (6.3)$$

The thermal conductivity of cavity and insulation, $k_{\text{SiC}}$ and $k_{\text{Al}_2\text{O}_3-\text{SiO}_2}$, respectively, and the serial-to-parallel slab ratio, $f$ (cf. Equation (5.5)), the specific surface area, $s$, and the nominal pore diameter, $d_{\text{nom}}$, of the RPC were varied ±20%. The variation of the RPC porosity is constrained by $\phi \leq 1$. Material property interdependencies (e.g. $f$ vs. $\phi$) were not considered. The impact on efficiency of $d_{\text{nom}}$ and $s$ are below the model accuracy and not displayed. Changes in $k_{\text{SiC}}$, $k_{\text{Al}_2\text{O}_3-\text{SiO}_2}$, and $f$ have an impact below 1%, while reducing $\phi$ could lead to a gain in $\eta_{th}$ of up to 6.3%.

The cavity length, $L_{\text{cav}}$, and thickness of cavity, RPC, and insulation, $t_{\text{cav}}$, $t_{\text{RPC}}$, and $t_{\text{ins}}$, respectively, were varied ±50% from the baseline values provided in Table 6-1. Reducing the cavity thickness $t_{\text{cav}}$ leads to the largest increase in $\eta_{th}$, as conduction across the cavity wall is the limiting heat transfer mode. It is noted that $t_{\text{cav}}$ is constrained by the mechanical stability of the cavity under high pressure and temperature loads.

<table>
<thead>
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</tr>
<tr>
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<td>7</td>
</tr>
<tr>
<td>$t_{\text{RPC}}$</td>
<td>20</td>
</tr>
<tr>
<td>$t_{\text{ins}}$</td>
<td>79</td>
</tr>
</tbody>
</table>
Figure 6-11 – Sensitivity analysis of the material properties. Values for $\varphi$ have to be multiplied by 10. Trends are indicated by dashed lines.

Figure 6-12 – Parameter study of the geometric dimensions. Values for $\alpha$ have to be divided by 10. Trends are indicated by dashed lines.
At the ~80% thermal efficiency level (reference case), a relative efficiency increase of up to 11.2% might be realized by reducing $t_{\text{cav}}$ and $t_{\text{RPC}}$, and increasing $L_{\text{cav}}$ and $t_{\text{ins}}$. An additional decrease in $\phi$ also has a beneficial effect on $\eta_{\text{th}}$, as long as the pores remain connected and the air flow is not obstructed. In reality, physical variations in $\phi$ necessitate the determination of a new Nu correlation via DPLS due to the inherent interdependency of $\phi$, $d_{\text{nom}}$, and $f$.

Reversing the flow results in an improvement of the heat transfer to the air flow due to less steep temperature gradients but $T_{\text{cav}}$ is increased. With $\dot{Q}_{\text{rerradiation}}$ being the major heat loss mechanism (cf. Figure 6-9), such an increase overcompensates the heat transfer enhancements and leads to a reduction in the receiver performance of ~0.1%. Another drawback of the reverse flow configuration is the high temperature near the cavity sealing unless the sealing can be moved closer to the cooled CPC, cf. Figure 2-1.

6.5 Summary

In the previous two Chapters, a coupled MC-FV numerical heat transfer model of the solar receiver was presented. The model was validated with experimental data and applied to examine receiver operating conditions that could not be tested experimentally, especially at higher inlet air pressures and temperatures. The optimum operating conditions were found for $T_{\text{out}} \approx 600 – 900 \, ^\circ\text{C}$, resulting in thermal efficiencies ranging ~70 – 80% and an ideal solar heat engine efficiency ($\eta_{\text{th}} \times \eta_{\text{Carnot}}$) of 53.8% for 665 °C and $\eta_{\text{th}} = 79.6\%$. Reradiation was found to be the main heat loss mechanism, while conduction across the cavity wall was the limiting heat transfer mode. A sensitivity analysis showed that variations in material properties have a negligible effect on the receiver performance. A parametric study indicated that a significant relative efficiency increase of up to 11.2% might be achieved at the ~80% thermal efficiency level by reducing $t_{\text{cav}}$ and $t_{\text{RPC}}$, and increasing $L_{\text{cav}}$ and $t_{\text{ins}}$. Heat losses occurring at the pressure vessel surface can be significantly reduced for a large-scale solar receiver cluster design, consisting of an array of multiple modular solar receiver units arranged side-by-side. Reversing the air flow did not improve the receiver performance.
7 System integration study

An explicit correlation of $\eta_{th}$ as a function of $T_{in}$ and $T_{out}$ was derived using the FV model results from the previous Chapter. The results were expanded to $p = 30$ bar to cover a larger range of potential operating conditions. Since the mechanical stability of the SiC cavity was only certified to up to $p = 10$ bar, this extrapolation to 30 bar was not shown earlier but is included in the theoretical study presented in this Chapter. For large-scale power plant integration, a solar receiver cluster design is proposed consisting of an array of multiple modular solar receiver units arranged side-by-side, each attached to a hexagon-shaped secondary concentrator in a honeycomb structure [41] following a spherical fly-eye optical configuration [42], cf. Figure 7-11. Consequently, the losses occurring at the pressure vessel, $\dot{Q}_{\text{radiation,amb}}$ and $\dot{Q}_{\text{convection,amb}}$ were set to zero, cf. Figure 6-9 and Equation (6.2). Figure 7-1 shows the 2nd-order polynomial fit for $\eta_{th}$ used for the subsequent analyses:

$$\eta_{th} = A + B \cdot T_{in} + C \cdot T_{out} + D \cdot T_{in} \cdot T_{out} + E \cdot T_{out}^2$$

(7.1)

with $A = 0.9494$, $B = 7.23 \cdot 10^{-5}$, $C = -9.316 \cdot 10^{-5}$, $D = -1.537 \cdot 10^{-7}$, $E = -6.53 \cdot 10^{-8}$; and $R^2 = 0.9939$. The receiver inlet and outlet air temperatures are given by the corresponding temperature in the cycle analysis.
Figure 7-1 – Thermal receiver efficiency vs. receiver inlet and outlet air temperature. Simulation values are displayed by dots, the surface represents the 2nd-order polynomial fit of Equation (7.1).

7.1 Power cycle overview

Four simplified cycles were analyzed: (a) a simple Brayton cycle (BC) and (b) a recuperated Brayton cycle (RC), both schematically shown in Figure 7-2, (c) a combined Brayton-Rankine cycle (CC) and (d) a dry steam reheat CC (RHCC), both schematically shown in Figure 7-3. The corresponding T-s diagrams are shown in Figure 7-4 and Figure 7-5 for an exemplary BC operating between \( T_1 = T_{\text{amb}} = 25 \, ^\circ\text{C} \) and \( T_3 = T_{\text{out}} = 1000 \, ^\circ\text{C} \) with pressure ratio \( \Pi = 8 \), defined as:

\[
\Pi = \frac{p_2}{p_1} \tag{7.2}
\]

with \( p_1 = p_{\text{amb}} \) at the compressor inlet. The shown Rankine cycle exhibits temperatures ranging from \( T_A = 46 \, ^\circ\text{C} \) to \( T_C = 550 \, ^\circ\text{C} \) with steam pressure \( p_B = 100 \, \text{bar} \) and condensation pressure \( p_A = p_D = 0.1 \, \text{bar} \). Wet steam, characterized by the vapor ratio (i.e. vapor quality), \( x < 1 \), can have an adverse effect on turbine expander blades with decreasing \( x \). This drawback can be
prevented by adding a reheat and a secondary expansion stage as indicated in Figure 7-5, cp. Figure 7-3 (d).

All investigated cycles use the following simplifications:

(a) Single shaft compressors and turbines
(b) No bleeding of pressurized cooling air from compressor
(c) No bleeding of hot air or steam from turbines for preheat or regeneration
(d) No mixing of fluids inside heat exchangers and regenerators
(e) Adiabatic compressor, turbine and heat exchanger housings

Figure 7-2 – Cycle schematics: (a) simple Brayton, (b) recuperated Brayton cycle
Figure 7-3 – Cycle schematics: (c) combined Brayton-Rankine cycle, (d) reheat Brayton-Rankine cycle
Figure 7-4 – T-s diagram of an exemplary (recuperated) Brayton cycle. The dotted lines represent ideal isentropic compression or expansion. The blue area indicates the temperature band available for recuperation.

Figure 7-5 – T-s diagram of an exemplary combined Brayton-Rankine cycle. The black line indicates the two-phase liquid-vapor region. The dotted lines represent isentropic expansions. A reheat and secondary expansion stage are indicated by the dashed lines.
While a reheat and a secondary expansion stage are suitable for the Rankine cycle, the concept of adding a reheat stage becomes less attractive in a solar Brayton cycle. The reheat would occur in an additional solar receiver, which would then operate at significantly higher average temperatures leading to increased thermal losses, cf. Figure 7-1. Additionally, the pressure drop would increase at the lower reheat stage pressure due to higher fluid velocities. Thus, this concept was not further analyzed. Also, adding more than two reheat stages to the Rankine cycle is unnecessary from an economical point of view [43].

In this study, the required heliostat field design and performance is not accounted for. Nevertheless, it is clear that the condenser necessarily has to use a dry cooling concept to prevent cloud generation, which would have a negative impact on the optical performance of any heliostat field.

### 7.2 Power cycle calculations

For simplicity, the variables are labeled according to the process points (1 – 5, A – F), cf. Figure 7-2 – Figure 7-5. A summary of the assumptions and constraints for the cycle parameters is provided in Table 7-1 at the end of this Section. In the following analysis, enthalpy values, $h$, entropy values, $s$, and specific volume, $v$, for air, water, and steam were taken from Refs. [30,36]. For the working fluid, a relative pressure drop, $\Delta p = 5\%$, is presumed when passing through the receivers and both sides of the heat exchangers. This $\Delta p$ is considered rather conservative since at high $\Pi$, lower $\Delta p$ are expected.

**Brayton cycle** – Ambient air is compressed in a non-ideal compressor (1 $\rightarrow$ 2). For non-ideal compression of an ideal gas [34]:

$$T_2 = T_1 \cdot \Pi^{c_p \eta_c}$$

in which $T_1$ and $p_1$ are at ambient conditions, $T_2$ and $p_2$ are the inlet conditions to the solar receiver array, and $\eta_c = 0.9$ is the isentropic compressor efficiency, cf. Section 5.4. The compressed air is then fed to the receiver array and heated to a desired temperature $T_3$ (2 $\rightarrow$ 3).
Despite the low pressure drop characteristics of the receiver (cf. Section 4.1), a relative pressure loss penalty is applied to account for piping and instrumentation of the receiver array:

\[ p_3 = (1 - \Delta p) p_2 \]  

(7.4)

Next, the working fluid is expanded in a non-ideal gas turbine (3 → 4). For non-ideal expansion of an ideal gas [44]:

\[ T_4 = T_3 \cdot \frac{p_3}{p_4}^{\frac{-R \eta_{gt}}{\epsilon_p}} \]  

(7.5)

where \( p_4 = p_1 \) if an open cycle is assumed. For industrial gas turbines, \( \eta_{gt} > 0.9 \) are reported near their design point [5]; thus, \( \eta_{gt} = 0.9 \) was chosen to determine \( T_4 \). The compressor and gas turbine powers are defined as:

\[ \dot{W}_c = \dot{m} \left( h_2 - h_1 \right) \]  

(7.6)

\[ \dot{W}_{gt} = \dot{m} \left( h_3 - h_4 \right) \]  

(7.7)

respectively, with the air mass flow rate, \( \dot{m} \), being set according to the desired \( T_3 \). The BC efficiency, \( \eta_{BC} \), is defined as:

\[ \eta_{BC} = \frac{\dot{W}_{gt} - \dot{W}_c}{Q_{th}} \]  

(7.8)

**Recuperated cycle** – Thermal power to preheat the receiver flow is extracted by a heat exchanger (HX) located between 4 → 5 and transferred to the working fluid between 2 → 2'. The temperature at position 2' is found via the HX effectiveness, \( \epsilon \).
A typical \( \varepsilon = 0.9 \) for gas turbine regenerators was chosen [45]:

\[
\varepsilon = \frac{T_{2'} - T_2}{T_4 - T_2} \tag{7.9}
\]

Since recuperation can only occur if \( T_{2'} > T_2 \), a sufficiently high \( T_4 \) is required. In the RC, \( p_4 > p_5 \) due to the additional flow resistance inside the HX:

\[
p_5 = (1 - \Delta p) p_4 \tag{7.10}
\]

\[
p_{2'} = (1 - \Delta p) p_2 \tag{7.11}
\]

Again, the RC efficiency, \( \eta_{RC} = \eta_{BC}, \text{ cf. Equation (7.8).} \)

**Combined cycle** – The bottom Rankine cycle incorporated in the CC operates between \( p_A \) and \( p_B \), with \( p_B \) usually ranging from 100 – 170 bar [46], a typical value of 100 bar was chosen according to Ref. [47]. The water is pressurized by a pump (A \( \rightarrow \) B). \( T_A, h_A \) and \( v_A \) are defined as saturation conditions at \( p_A \). \( h_B \) is found via:

\[
h_B = h_A + v_A (p_B - p_A) \tag{7.12}
\]

To determine the water mass flow rate, \( \dot{m}_w \), and the superheated steam temperature, \( T_C \), a heat recovery steam generator (HRSG) has been analytically analyzed to design a Rankine cycle that can be operated using only the rejected heat of the BC (4 \( \rightarrow \) 5). As shown in Figure 7-6, the HRSG (B \( \rightarrow \) C) is assumed to operate without losses to the surroundings and is split into three parts [48]: (a) the economizer, heating the pressurized water close to its evaporation temperature with a threshold called the approach temperature difference, \( \Delta T_{ap} \), (b) the evaporator, generating live steam with a certain pinch temperature difference, \( \Delta T_{pinch} \), between the hot and cold side (air and steam side, respectively), and (c) the superheater, further heating the dry steam.

Given \( \Delta T_{pinch} = 20 \, ^\circ C \) and \( \Delta T_{ap} = 10 \, ^\circ C \) [5], \( T_C \) and \( \dot{m}_w \) are found by solving the three energy conservation equations in each part of the HRSG, summarized as:
\[ \dot{m}_w (h(T_C) - h_B) = \dot{m} (h_4 - h_3) \]  

paired with the 2nd law of thermodynamics to ensure that the two lines in Figure 7-6 do not intersect. The Figure shows an exemplary HRSG diagram for a CC, in which the Brayton side operates at \( T_3 = 1000 \, ^\circ\text{C} \) and \( \Pi = 6 \) resulting in \( T_4 = 608 \, ^\circ\text{C} \) (red line); the Rankine side operates at \( p_A = 0.1 \, \text{bar} \) and \( p_B = 100 \, \text{bar} \) and reaches \( T_C = 383 \, ^\circ\text{C} \) (blue line). \( T_C \) is typically found in the range of \( 500 - 600 \, ^\circ\text{C} \) [46] and limited to \( \sim 650 \, ^\circ\text{C} \) due to piping material limitations for austenitic steels [2]. To cope with changes in \( T_4 >> 600 \, ^\circ\text{C} \), \( \dot{m}_w \) is increased accordingly, consequently leading to a larger Rankine cycle.

Figure 7-6 – Fluid temperature vs. relative heat transfer in a heat recovery steam generator consisting of (a) economizer, (b) evaporator, and (c) superheater. The hot air flow (red line) heats, evaporates, and superheats the water-steam flow (blue line).
To account for pressure losses in the HRSG, \( p_4 \) and \( p_5 \) are treated according to Equation (7.10) and:

\[
p_C = (1 - \Delta p) p_B
\]  

(7.14)

Using the entropy value for \( s_C \), \( h_D \) can be calculated (\( C \rightarrow D \)):

\[
h_D = h_C - \eta_{st} (h_C - h_{D,is})
\]  

(7.15)

with the isentropic steam turbine efficiency, \( \eta_{st} = 0.9 \), for shrouded reaction blading [49] and the enthalpy reached for ideal isentropic expansion, \( h_{D,is} \), from \( p_C \) to \( p_D = p_A = 0.1 \text{ bar} \) (i.e. \( \eta_{st} = 1 \)). The corresponding \( T_D = T_A \) now lies within the two-phase liquid-vapor region with vapor quality, \( x \), defined as:

\[
x = \frac{h_D - h_{D,l}}{h_{D,v} - h_{D,l}}
\]  

(7.16)

with \( h_{D,l} \) and \( h_{D,v} \) defined for the pure liquid and vapor phase, respectively. The pump and the steam turbine powers are defined as:

\[
\dot{W}_p = \frac{\dot{m}_w (h_B - h_A)}{\eta_p}
\]  

(7.17)

\[
\dot{W}_{st} = \dot{m}_v (h_C - h_D)
\]  

(7.18)

respectively, with \( \dot{m}_v = \dot{m}_w \), and the pump efficiency, \( \eta_p = 0.85 \), which had a negligible impact on the system performance. Including the topping Brayton cycle, the CC efficiency, \( \eta_{CC} \), is defined as:

\[
\eta_{CC} = \frac{\dot{W}_{gt} + \dot{W}_{st} - \dot{W}_c - \dot{W}_p}{\dot{Q}_{th}}
\]  

(7.19)
Reheat combined cycle – After the first expansion (C → D'), the RHCC requires an additional superheater (i.e. reheater) between D' → E followed by an additional expander stage (E → F). To avoid entering the two-phase region, both expansion stages are constrained by $x = 1$, cf. Figure 7-5. In an iterative approach, $p_{D'} = 13$ bar was found, which fulfills both constraints for all analyzed conditions. With expansion from $p_C$ to $p_{D'}$, $h_{D'}$ is found according to Equation (7.15) via $s_C$.

The additional pressure drop in the reheater is accounted for by:

$$p_E = (1 - \Delta p) p_{D'} \tag{7.20}$$

$$p_5 = (1 - 2\Delta p) p_4 \tag{7.21}$$

on the steam and air side, respectively. Determining $T_C$ and $\dot{m}_w$ requires the combined HRSG-reheater energy balance:

$$\dot{m}_w (h_C - h_B + h_E - h_{D'}) = \dot{m} (h_4 - h_5) \tag{7.22}$$

with $h_E = h_C = h(T_C)$. $h_E$ is determined via the reheat temperature $T_E = T_C$ and $p_E$, and $h_F$ via expansion to $p_F$ similar to Equation (7.15) via $s_E$. The steam turbine powers now become:

$$\dot{W}_{st} = \dot{m}_v (h_C - h_{D'} + h_E - h_F) \tag{7.23}$$

with $\dot{m}_v = \dot{m}_w$, resulting in the RHCC efficiency, $\eta_{RHCC} = \eta_{CC}$, cf. Equation (7.19). Considering the 2nd law of thermodynamics, the cycle was chosen such that the reheater is directly heated by the hot air flow exiting the gas turbine prior to the HRSG, cf. Figure 7-3. From a manufacturing point of view this is advantageous since the pressure difference between the two fluids is lower at these higher temperatures, which allows a reduction of the wall thicknesses of the heat exchanger tubes inside the reheater.

Finally, Table 7-1 provides a summary of the assumptions and constraints for the cycle parameters.
Table 7-1 – Summary of cycle parameters and constraints

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</table>

7.3 Idealized cycle results

Starting with a cycle provided with ideal, lossless combustion instead of the solar receiver, Figure 7-7 – Figure 7-10 show the power cycle efficiency, $\eta_{cycle}$ vs. $\Pi$ and $T_3$ for an idealized BC, RC, CC, and RHCC using the parameters defined in Table 7-1.

The non-solar BC shows highest efficiency for maximized $\Pi$ and $T_3$. $\eta_{BC}$ benefits from high $\Pi$ and $T_3$ until the compressor work required for high $\Pi$ becomes too large. Opposed to the BC, the non-solar RC profits from low $\Pi$ because of increased heat recuperation attributed to the higher exhaust temperatures; thus, a secondary peak at low $\Pi$ is observed if the requirement $T_4 > T_2' > T_2$ is met, allowing high internal heat exchange, $\dot{Q}_{recovery}$, cf. Figure 7-4. At a certain level, increasing $\Pi$ leads to $T_2 \approx T_4$, resulting in the HX to become obsolete.
The non-solar CC shows significantly higher cycle efficiencies than the BC and RC. Similar to the BC, the CC shows highest efficiency for maximized $T_3$ but only allows $\Pi < 22$. For each $T_3$, a certain $\Pi$ threshold is found at which $x_{\text{min}}$ is not reached and the Rankine cycle becomes obsolete. Then, the $\eta_{\text{CC}}$ curves almost converge with the original $\eta_{\text{BC}}$ curves with an offset attributed to the additional $\Delta p$ in the HRSG. Shifting the sharp decline towards higher $\Pi$ can be achieved by reducing $x_{\text{min}}$ and $p_B$ or increasing $p_D$. Compared to the BC and RC, the CC can theoretically also be operated at $\Pi \approx 1$, in which case the gas turbine does not produce any work and the hot air is used as heat transfer fluid to heat the steam in the Rankine cycle. Thus, at $\Pi \approx 1$, $\eta_{\text{CC}}$ does not reach zero but reflects the efficiency of the Rankine cycle operating at the corresponding $T_4$.

The reheat combined cycle shows a steeper incline in efficiency at low $\Pi$ but cannot reach the same efficiency level as the CC. This reduction in performance is attributed to the additional pressure drop inside the reheater. Similar to the CC, a $\Pi$ threshold is found, at which $x < 1$ is reached in at least one of the steam turbines, which violates the dry steam constraint ($x = 1$). Again due to the increased pressure drop, the curves do not fully converge with the original $\eta_{\text{BC}}$ curves. Compared to the CC, the $\Pi$ threshold is shifted towards higher $\Pi$ (i.e. lower $T_4$) because the RHCC can extract more sensible heat from the vapor region of the reheat stage, than the sensible and/or latent heat the CC can extract before $x_{\text{min}}$ is reached. Proof can be found in a water-steam enthalpy chart, or Ref. [30]. Again, reducing $p_B$ or increasing $p_D$ shifts the efficiency decline towards higher $\Pi$. 
Figure 7-7 – Brayton cycle efficiency vs. pressure ratio without solar receiver (i.e. $\eta_{th} = 1$) for turbine inlet air temperatures ranging 700 – 1400 °C

Figure 7-8 – Recuperated Brayton cycle efficiency vs. pressure ratio without solar receiver (i.e. $\eta_{th} = 1$) for turbine inlet air temperatures ranging 700 – 1400 °C
Figure 7-9 – Combined Brayton-Rankine cycle efficiency vs. pressure ratio without solar receiver (i.e. $\eta_{th} = 1$) for turbine inlet air temperatures ranging 700 – 1400 °C

Figure 7-10 – Reheat combined cycle efficiency vs. pressure ratio without solar receiver (i.e. $\eta_{th} = 1$) for turbine inlet air temperatures ranging 700 – 1400 °C
7.4 Solar cycle results

Preliminary studies showed that placing the power block on the ground causes significant pressure and temperature losses when transporting the compressor discharge air up, and the hot receiver outlet air down [50], cf. Figure 1-1. To remain in a temperature range suitable for stainless steel, steel temperatures $<600$ $^\circ$C are desired [44]. To overcome such material limitations, a co-annular piping system was proposed consisting of an inner pipe transporting the hot receiver outlet air downward surrounded by an annular pipe transporting the cold receiver inlet air upward. Depending on the diameter ratio of the pipes, operating pressure, and temperature, this optimized air transport concept resulted in absolute thermal losses ranging $\sim 0.05 – 0.1\%$ per meter pipe length (i.e. tower height, $h$). Considering a solar tower height of $>120$ m, $\sim 12\%$ thermal losses are expected during transport for a receiver outlet air temperature, $T_{\text{out}} = 900$ $^\circ$C. Consequently, installing the gas turbine expander on tower top is considered the only option to overcome such limitations [35,51]. In such a setup, the turbine rotational axis can be aligned along the tower axis and axial bearings can be used with the potential of reducing vibrations. While the gas turbine is located on tower top, the steam turbine, condenser, and water pump remain on ground level. Due to the large size of the HRSG (large internal heat exchange surface areas), the hot exhaust air from the gas turbine has to be transported back down to the ground level. Given state-of-the-art concepts for steam-generation solar power towers [47,52,53], the transport of hot pressurized steam $<600$ $^\circ$C is not considered as major challenge. Because of the lower overall heat transfer coefficient of air compared to steam, no additional penalties for the transport of the hot exhaust air are incorporated in the subsequent analysis, in which $T_i > T_C$.

Following the idealized non-solar cycle efficiencies, $\eta_{BC}$, $\eta_{RC}$, and $\eta_{CC}$ (cf. Equation (7.19)), the solar power cycle efficiencies are defined as:

$$
\eta_{\text{cycle}} = \eta_{\text{th}} \cdot \eta_{\text{cycle}} = \frac{-\dot{W}_{st} + \dot{W}_{c} - \dot{W}_{p}}{Q_{\text{in}}} \tag{7.24}
$$
The subscript “cycle” may refer to either BC, RC, CC, or RHCC. It is noted that $\dot{W}_{st} = \dot{W}_{p} = 0$ for the BC and RC cases.

For real solar tower applications, a receiver cluster arranged in a fly-eye configuration has been proposed previously [31,42], while in this study, a flat honeycomb structure is analyzed [41]. Figure 7-11 shows the proposed receiver array consisting of 85 receivers attached to hexagon-shaped CPCs arranged side-by-side. This array can be divided into 43 high-temperature (HT) receiver-CPC units located at the center in the high flux region (high $\dot{Q}_{in}$), and 42 low-temperature (LT) units receiving lower $\dot{Q}_{in}$ located peripherally [54]. The LT units are used to preheat the air, which is then fed to the HT units where it is heated to the desired $T_3$. This concept has three-fold advantages: (a) the HT units located in the center of the array benefit from the elimination of the thermal losses at the pressure vessel surface ($\dot{Q}_{\text{radiation,amb}}$ and $\dot{Q}_{\text{convection,amb}}$) because they are in contact with each other, increasing the receiver performance by a percentage of up to 18% compared to a single receiver (cf. Figure 6-9, $T_{\text{out}} = 1400 \, ^\circ\text{C}$), (b) the LT units located at the edges of the array still suffer from these losses, but since they operate at lower temperatures, the losses are less severe and can be minimized by additional insulation, and (c) the spillage of the incoming flux is captured by the LT units. The only disadvantage of using a two-stage serial receiver concept is the increase in $\Delta p$. Table 7-2 summarizes all occurrences of $\Delta p$.

<table>
<thead>
<tr>
<th>Table 7-2 – Pressure drop occurrences</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Receiver array</strong></td>
</tr>
<tr>
<td>Single-stage</td>
</tr>
<tr>
<td>Two-stage</td>
</tr>
<tr>
<td><strong>Locations of $\Delta p$ in Brayton cycle for air</strong></td>
</tr>
<tr>
<td><strong>$\Delta p$ in Rankine cycle for steam</strong></td>
</tr>
</tbody>
</table>
7.4.1 Single-stage receiver array

The solar cycle efficiencies for a single-stage receiver array with 85 receivers in parallel, in which no distinction between LT and HT receivers is made, are shown in Figure 7-12 – Figure 7-15 as a function of $\Pi$ and $T_3$ for a solar BC, RC, CC, and RHCC using the parameters defined in Table 7-1 and $\eta_{\text{th}}$ from Equation (7.1).

The cycle efficiency of the solar BC reaches a plateau of $\eta_{\text{solar BC}} > 28\%$ for $\Pi > 14$ and $T_3 > 900$ peaking at $\eta_{\text{solar BC}} \approx 30.1\%$ for $\Pi = 30$ and $T_3 = 1100$ °C. Following the trend of the non-solar BC a high $\Pi$ is beneficial. The solar RC clearly benefits from heat recuperation and exhibits a distinct peak at $\eta_{\text{solar RC}} \approx 30.3\%$ for $\Pi = 3.5$ and $T_3 = 700$ °C. The trend shows that lowering $T_3$ might significantly improve $\eta_{\text{solar RC}}$. However, lowering $T_3$ to below 600°C obviates the need of a pressurized air receiver, since state-of-the-art steam
receivers already perform well in this region [3,47,52,53]. Keeping $T_3$ low can also reduce material cost and prolong the receiver’s lifespan.

The cycle efficiency of the solar CC shows a plateau of $\eta_{\text{solar CC}} > 37\%$ for $\Pi = 4 – 16$ and $T_3 = 1100 – 1300 \, ^\circ\text{C}$. The peak $\eta_{\text{solar CC}} \approx 37.9\%$ is reached at $\Pi = 6.5 – 10$ and $T_3 = 1200$, right before $x_{\text{min}}$ exceeds 0.8. Similar to the solar CC, the solar RHCC shows a plateau of $\eta_{\text{solar RHCC}} > 36\%$ for $\Pi = 4 – 13$ and $T_3 = 1200 – 1400 \, ^\circ\text{C}$, peaking at $\approx 36.4\%$ for $\Pi = 8$ and $T_3 = 1300 \, ^\circ\text{C}$. Compared to the solar RHCC, the solar CC exhibits higher efficiencies at lower temperatures and pressures. Table 7-3 provides a summary of the peak $\eta_{\text{solar cycle}}$ and the required operating conditions, $T_3$ and $\Pi$, for the solar BC, RC, and both CCs.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>BC</th>
<th>RC</th>
<th>CC</th>
<th>RHCC</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\eta_{\text{solar cycle}}$ (%)</td>
<td>30.1</td>
<td>30.3</td>
<td>37.9</td>
<td>36.4</td>
</tr>
<tr>
<td>$T_3$ (°C)</td>
<td>1100</td>
<td>700</td>
<td>1200</td>
<td>1300</td>
</tr>
<tr>
<td>$\Pi$ (-)</td>
<td>30</td>
<td>3.5</td>
<td>6.5</td>
<td>8</td>
</tr>
</tbody>
</table>

For low $\Pi$, the gas turbine and compressor unit become smaller in size, which is beneficial when located on tower top because of the reduced weight and inertia. The increased piping diameters at low $\Pi$ are compensated by the relatively short piping lengths if mounted on tower top. Also, a smaller BC can adapt faster to transients caused by intermittency in $\dot{Q}_\text{in}$ due to e.g. clouds. To avoid problems with transients in the HRSG, it is advised to have a hybrid co-firing or heat storage option.
Figure 7-12 – Solar Brayton cycle efficiency vs. pressure ratio for receiver outlet air temperatures ranging 700 – 1400 °C

Figure 7-13 – Solar recuperated Brayton cycle efficiency vs. pressure ratio for receiver outlet air temperatures ranging 700 – 1400 °C
Figure 7-14 – Solar combined Brayton-Rankine cycle efficiency vs. pressure ratio for receiver outlet air temperatures ranging 700 – 1400 °C

Figure 7-15 – Solar reheat combined cycle efficiency vs. pressure ratio for receiver outlet air temperatures ranging 700 – 1400 °C
7.4.2 Two-stage receiver array

If two (or more) receiver stages are considered, the power share per stage, \( f \), has to be defined. In the two-stage case at hand, the power share of the receiver array’s LT stage, \( f_{LT} = 1 - f_{HT} \), can vary between 0 – 1. The previously discussed single-stage array would be equivalent to \( f_{LT} = 0 \) or 1 but operating at twice the \( \Delta p \) (cf. Table 7-2) to allow comparison of the single-stage vs. two-stage array.

Figure 7-16 shows the peak efficiencies as a function of \( f_{LT} \) for the BC, RC, and both CCs. Despite the increased \( \Delta p \), \( \eta_{solar \ BC} \) could be increased to \( \sim 30.3\% \) (+0.2%) now occurring at \( \Pi = 30 \) and \( T_3 = 1200 \, ^\circ C \) (+100 \, ^\circ C) for \( f_{LT} = 0.5 \). The effect of the increased \( \Delta p \) is represented by the efficiency reduction at \( f_{LT} = 0 \) and 1. The RC does not benefit from the serial receiver configuration because the HX already provides the necessary preheat and the receiver units already operate at higher average temperatures compared to the BC. Consequently, a RC does not make sense in a two-stage receiver array and vice-versa. For \( T_3 = 1300 \, ^\circ C \) (CC: +100 \, ^\circ C) and \( f_{LT} = 0.6 \), the two CC cases show the highest augmentations with \( \eta_{solar \ CC} = 39.3\% \) (+1.4%) and \( \eta_{solar \ RHCC} = 37.3\% \) (+0.9%) at \( \Pi = 9 \) (+2.5) and \( \Pi = 8.5 \) (+0.5), respectively. On one hand, both CC benefit from the increased performance of the BC, and on the other hand, the higher \( \Delta p \) in the BC leads to higher \( T_4 \) because less energy can be extracted in the expander, consequently leading to an increase of the Rankine cycle efficiency.

Table 7-4 summarizes the operating conditions that yield peak \( \eta_{solar \ BC} \) and the more promising \( \eta_{solar \ CC} \), with \( T_3 \) being the intermediate temperature between the LT and HT receiver. Another strategy to improve \( \eta_{solar \ CC} \) is to increase \( T_C \) to 650 \, ^\circ C \) and \( p_B \) to 165 bar. Compared to the previous increase, this additional measure leads to another increase in \( \eta_{solar \ CC} = 40.6\% \) (+1.3%) at \( \Pi = 7.5 \) (-1.5) again for \( T_3 = 1300 \, ^\circ C \) and \( f_{LT} = 0.6 \).

Despite these promising results, the \( T_{out} > 1300 \, ^\circ C \) operating range has to be handled with care because \( T_{cav} \) reaches the SiC material limit of 1600 \, ^\circ C, cf. Section 6.1. Considering a more evenly distributed \( \dot{q}_{abs} \) (cf. Figure 5-4) this may be avoided.
The suggested LT-HT configuration from Figure 7-11 can adapt according to the incoming flux delivered by the heliostat field and the required $f_{LT}$. E.g. switching the six outmost HT receivers to LT-mode already tackles the $f_{LT} = 0.6$ requirement by means of number of receivers if a constant power per receiver can be assumed. The next logical step would be designing a heliostat field that can provide the necessary preheat power share of 0.5 – 0.6.

Table 7-4 – Operating conditions of a two-stage receiver array for peak $\eta_{solar\ cycle}$

<table>
<thead>
<tr>
<th>Parameter</th>
<th>BC</th>
<th>CC</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\eta_{solar\ cycle} (%)$</td>
<td>30.3</td>
<td>39.3</td>
</tr>
<tr>
<td>$T_3$ (°C)</td>
<td>1200</td>
<td>1300</td>
</tr>
<tr>
<td>$\Pi$ (-)</td>
<td>30</td>
<td>9</td>
</tr>
<tr>
<td>$f_{LT}$ (-)</td>
<td>0.5</td>
<td>0.6</td>
</tr>
<tr>
<td>$T'_3$ (°C)</td>
<td>937</td>
<td>912</td>
</tr>
<tr>
<td>$\eta_{th\ LT}$ (%)</td>
<td>77.9</td>
<td>78.7</td>
</tr>
<tr>
<td>$\eta_{th\ HT}$ (%)</td>
<td>64.4</td>
<td>60.2</td>
</tr>
<tr>
<td>$\eta_{th\ array}$ (%)</td>
<td>71.2</td>
<td>71.3</td>
</tr>
</tbody>
</table>
7.4.3 Techno-economic considerations

Figure 7-17 and Figure 7-18 show $\eta_{\text{solar cycle}}$ vs. the specific work output, $w_{\text{out}}$, defined in Equation (7.25), for the solar BC and CC both using the two-stage receiver array at their optimum $f_{LT} = 0.5$ and 0.6, respectively.

\[ w_{\text{out}} = \frac{\dot{W}_{\text{out}}}{\dot{m}_{\text{air}}} = \frac{\dot{W}_{\text{gl}} + \dot{W}_{\text{st}} - \dot{W}_{\text{c}} - \dot{W}_{\text{p}}}{\dot{m}_{\text{air}}} \]  \hspace{1cm} (7.25)

with $\dot{W}_{\text{st}} = \dot{W}_{\text{p}} = 0$ for the BC. For a given power plant output, $\dot{W}_{\text{out}}$, the size of the power block depends on $w_{\text{out}}$ [44], while the overall efficiency, $\eta$, dictates its fuel consumption, which is equivalent to the heliostat field size in CSP applications. Thus, in solar applications $\eta$ influences parts of the initial investment cost, while in non-solar power plants it influences the variable costs accumulating during operation. From an economical point of view it is desired to have high $w_{\text{out}}$ to keep the power conversion unit (PCU) small. However, it is also desired to have high $\eta$ to reduce the required $Q_{\text{in}}$. The Figures indicate that high $w_{\text{out}}$ at high $\eta$ can be achieved in the CC, while the BC shows a more distinct trade-off behavior.

The difference in $w_{\text{out}}$ at comparably high $\eta$ is shown in Table 7-5 and Table 7-6 for the BC and CC, respectively. In a simplified economic analysis, the payback time is used to find out whether a power plant tuned for $w_{\text{out}}$ pays off faster than one optimized for $\eta$ (being the reference case), cf. Appendix E. The analysis is simplified such that the heliostat field cost and the power block cost scale linearly with $\eta$ and $w_{\text{out}}$, e.g. for a reduction in efficiency of 50%, the heliostat field cost would double to provide the same $Q_{\text{in}}$. Optimization towards $w_{\text{out}}$ is only advisable for the CC due to the comparably high cost of a CC power block. For the BC, the optimization strategy depends more on the comparably high heliostat field cost share and no deterministic statement about the optimization approach can be made at this point.
Figure 7-17 – Cycle efficiency as a function of the receiver outlet air temperatures and pressure ratios vs. specific work output of the solar Brayton cycle

Figure 7-18 – Cycle efficiency as a function of the receiver outlet air temperatures and pressure ratios vs. specific work output of the solar combined Brayton-Rankine cycle. Cases with insufficient steam production (*i.e.* *x* < 0.8) are not displayed.
Table 7-5 – Maximum $\eta_{\text{solar BC}}$ and $w_{\text{out}}$ for the two-stage receiver array

<table>
<thead>
<tr>
<th>Optimized for:</th>
<th>$\eta_{\text{solar BC}}$ (%)</th>
<th>$w_{\text{out}}$ (kJ/kg)</th>
<th>$\Delta$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\eta_{\text{solar BC}}$ (%)</td>
<td>30.3</td>
<td>27.4</td>
<td>-9.6</td>
</tr>
<tr>
<td>$w_{\text{out}}$ (kJ/kg)</td>
<td>310</td>
<td>452</td>
<td>+45.8</td>
</tr>
<tr>
<td>$\Pi$ (-)</td>
<td>30</td>
<td>18</td>
<td></td>
</tr>
<tr>
<td>$T_3$ (°C)</td>
<td>1200</td>
<td>1400</td>
<td></td>
</tr>
</tbody>
</table>

Table 7-6 – Maximum $\eta_{\text{solar CC}}$ and $w_{\text{out}}$ for the two-stage receiver array

<table>
<thead>
<tr>
<th>Optimized for:</th>
<th>$\eta_{\text{solar CC}}$ (%)</th>
<th>$w_{\text{out}}$ (kJ/kg)</th>
<th>$\Delta$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\eta_{\text{solar CC}}$ (%)</td>
<td>39.3</td>
<td>38</td>
<td>-3.3</td>
</tr>
<tr>
<td>$w_{\text{out}}$ (kJ/kg)</td>
<td>620</td>
<td>706</td>
<td>+13.9</td>
</tr>
<tr>
<td>$\Pi$ (-)</td>
<td>9</td>
<td>7.5</td>
<td></td>
</tr>
<tr>
<td>$T_3$ (°C)</td>
<td>1300</td>
<td>1400</td>
<td></td>
</tr>
</tbody>
</table>

Following the trends in CSP plant development, >100 MW absorbed solar thermal power is desired, cf. Table 1-1. The receiver cluster proposed in Figure 7-11 is suggested to be mounted on 6 – 9 sides of a solar tower, each facing its own heliostat field lobe [41]. This results in 510 – 765 receiver units each operating at ~130 kW$_{th}$. If a lower power per receiver is desired, it is suggested to mount another receiver array one level above the first to serve as preheat unit by using the incoming concentrated radiation from the far field. Figure 7-19 depicts such an envisioned solar tower configuration with separate LT and HT receiver array mounted on 9 sides of a tower. Each receiver array consists of 85 hexagon-shaped receiver-CPC units resulting in each receiver operating at ~65 kW$_{th}$, which is close to the operating range of the receiver prototype, cf. Chapters 2 – 4. Thus, the receiver developed within this study is considered a full-scale prototype.
Figure 7-19 – Advanced receiver arrays mounted on nine sides of a solar tower with low-temperature stage located on top of the high-temperature stage, inspired by [41]

7.5 Summary

In this Chapter, a power cycle analysis of a simple Brayton cycle, a recuperated Brayton cycle, a combined Brayton-Rankine cycle, and a reheat combined Brayton-Rankine cycle, all in conjunction with a pressurized air solar receiver array for CSP tower systems was presented. Using the receiver performance map obtained from the numerical heat transfer model, the goal was to find efficiencies for the combined receiver and power cycle system, whilst the optical characteristics of the heliostat field were neglected. Results of a single-stage receiver array showed promising efficiencies of 30.1%, 30.3%, 37.9%, and 36.4% for the solar BC, RC, CC, and RHCC, respectively. Compared to the BC, the RC only shows a slight improvement in $\eta_{solar\;RC}$. Depending on the overall power plant operating conditions, the additional cost of implementing a heat recovery system might not be justified. A two-stage solar receiver array divided into a low-temperature and a high-temperature part was further proposed. In such
a two-stage serial receiver arrangement, the solar BC, CC, and RHCC efficiencies could be increased to 30.3% (+0.2%), 39.3% (+1.4%), and 37.3% (+0.9%), respectively, for a preheat power share of 0.5 – 0.6 and $T_{\text{out}} \approx 1300 \, ^\circ\text{C}$. The RC did not benefit from the two-stage receiver array configuration.
8 Summary and outlook

In this thesis, a full-scale pressurized air solar receiver was developed, experimentally demonstrated, and analyzed in a numerical model. The design, fabrication, and experimental on-sun testing of a 50 kW<sub>th</sub> pressurized air solar receiver operating at 2 – 6 bar absolute pressure was presented in Chapters 2 – 4. The receiver prototype turned out to be very robust, it withstood steep transients and was very reliable. Also, the modular design was very flexible allowing experimental testing of three RPC configurations: (a) 10 PPI, (b) 20 PPI, and (c) 10 PPI with air flow guiding baffles. A peak outlet air temperature of 1206 °C was reached, setting a new record for pressurized air solar receivers, cf. Table 4-1. Most promisingly, a peak performance of 91.2% thermal receiver efficiency at <i>T</i><sub>out</sub> = 699 °C was achieved. Considering 2<sup>nd</sup> law thermodynamics, the receiver’s thermal efficiency was multiplied by the Carnot efficiency to simulate an ideal heat engine operating between the receiver outlet air temperature and ambient. The ideal operating conditions were found at an outlet air temperature range of ~600 – 900 °C. The peak ideal solar heat engine efficiency (<i>η</i><sub>th</sub> × <i>η</i><sub>Carnot</sub>) of 63.3% was also reached at <i>T</i><sub>out</sub> = 699 °C.

The numerical model presented in Chapters 5 and 6 was validated with experimental data of the on-sun tests. Operating conditions beyond the experimental limitations, especially at higher pressures and temperatures were assessed by the model. For <i>T</i><sub>out</sub> ≈ 600 – 900 °C, the optimum operating conditions were found resulting in a thermal efficiency ranging ~70 – 80%, and an ideal solar heat engine efficiency of 53.8% at 665 °C. Regardless of the operating conditions, reradiation was the main heat loss mechanism, while conduction across the cavity wall was the limiting heat transfer mode. A sensitivity analysis showed that the model was rather insensitive to variations in material properties. Thus, uncertainties in the material data have a negligible effect on the receiver performance. A parametric study indicated that by reducing the cavity and RPC thickness, and increasing the cavity length and the insulation
thickness, a significant relative efficiency increase of up to 11.2% might be achieved at the ~80% thermal efficiency level.

Having built a validated CFD model, the simulation results were extrapolated to serve as receiver performance map in a power cycle analysis. In a system integration study presented in Chapter 7, the performance characteristics of a simple Brayton cycle, a recuperated Brayton cycle, a combined Brayton-Rankine cycle, and a reheat combined Brayton-Rankine cycle were investigated, all in conjunction with a two-stage solar receiver array. For a preheat power share of ~50% and ~60% of the total input, the solar BC and CC efficiencies reached 30.3% and 39.3%, respectively. Conclusively, solar cycle efficiencies reaching 30 – 40% are considered very competitive compared to existing commercial CSP tower plants, cf. Table 1-1.

8.1 Outlook and research recommendations

Using the Carnot efficiency to represent an ideal heat engine, the experiments and the simulations both showed that the most promising receiver performance is found in the $T_{\text{out}} \approx 600 – 900$ °C operating range. By contrast, the cycle analyses showed that much higher receiver outlet temperatures are required to yield overall system optimization. Because the model overpredicts the receiver performance in the high-temperature range, the model is recommended to be adapted in order to account for the staircase-shaped RPC in the back part of the receiver (cp. Figure 2-2 and Figure 5-1) and the gaps occurring between cavity and RPC. Due to gravity and the stacked rings concept, these gaps tend to be larger at the bottom side of the cavity. Especially since no distinct gradient between $T^+$ and $T^-$ was registered in the model (cp. Figure 4-7 and Figure 6-3), further model optimization is advised (also, cf. Appendix B). Focusing on the receiver design, it is desired to avoid parasitic air flow through gaps between insulation, RPC, and cavity. Therefore, it is suggested to build a combined cavity-RPC structure in one single piece.

The constant relative $\Delta p$ of 5%, which was assumed in the system integration study is rather conservative; at high $\Pi$, low fluid velocities prevail, which reduces $\Delta p$ and consequently increases the cycle performance. Thus, future work should focus on the development of a model including a suitable piping concept of the proposed receiver array with a velocity dependent $\Delta p$. 
The power split between the LT and HT receiver units has to be certified via ray-tracing using known heliostat field geometries. Also, the solar concentration ratio required to achieve the desired intermediate temperature (temperature between LT and HT receiver) has to be certified at the edges of the receiver array. The proposed receiver array requires a multiple lobe heliostat field, including secondary optics, which are expected to achieve yearly average efficiencies of $\eta_{\text{opt}} = 0.35$ if located at 40 °deg N [41]. No statement about $\eta_{\text{solar-to-electricity}} = \eta_{\text{th}} \times \eta_{\text{cycle}} \times \eta_{\text{opt}}$ can be made so far because the part load characteristics of $\eta_{\text{cycle}}$ have to be fully assessed to account for transients in DNI, variations in cosine losses, and shading/blocking, all caused by solar intermittency.

In the present analysis only simple power cycles at design point (full load) were analyzed. It is further suggested to analyze more complex cycles using pressurized air bleeding from the compressor to provide blade cooling in the turbine and hot air and/or steam bleeding from the turbine to be used in regenerators. Also, heat storage systems, e.g. packed bed of rocks [55–57] or thermochemical storage [58], with the goal of having the Rankine cycle running continuously at full load are suggested as future research topics. Especially HRSGs do not favor transient operating conditions [48], thus either co-firing of fossil fuels or a heat storage strategy are highly recommended. The main underlying assumption of this study was to install the expander on tower top. If this requirement cannot be met, it is suggested to account for ~0.1% heat losses per meter tower height and reduce the presented efficiency values accordingly.

The current technology development towards higher firing temperatures in gas turbines continues and will exceed 1600 °C in future industrial applications [59,60]. Also the development of ultra-supercritical steam turbines is advancing [61] and BrightSource works on expanding the field of operation to 650 °C (Luz Power Tower 650). The temperature limit of SiC-based receiver cavities of 1600 °C allows the integration of the receiver concept in a wide range of such promising advanced power cycles. Consequently, efforts to drastically increase the receiver performance in the high-temperature region are highly recommended to become more competitive vs. non-solar power generation.

As a next logical step towards industrialization, the demonstration of the receiver concept in a pilot-scale receiver cluster consisting of seven receiver units is proposed, cf. Figure 8-1. Such a scale-up design requires multiple CPCs adjacent to each other. Their optical efficiency has a direct influence on the
overall power plant efficiency and has to be assessed separately. Also an evaluation of suitable solar concentrator technologies has to be performed, e.g. beam-down vs. tower top, and fly-eye vs. flat honeycomb structures, or the implementation on solar dishes has to be assessed, cf. Appendix F.

Figure 8-1 – Schematic of a receiver cluster attached to seven CPCs
Appendix A

Photographs

Figure A-1 – Front view of the receiver prototype [28]
Figure A-2 – Front view of the receiver mounted on the CPC [28]
Figure A-3 – Solar tower at the WIS seen from the north-western part of the heliostat field [28]. The beam-down mirror can be seen on the right hand side of the tower.

Figure A-4 – Tracking heliostats during experiments: (a) morning, (b) evening, seen from the 7th floor next to the CPC. In both photographs, heliostat 406 is in stow position, cf. Figure 3-2.
Figure A-5 – Glowing cavity at ~1400 °C, seen from CPC inlet

Figure A-6 – Glowing cavity at ~700 °C, seen from CPC inlet
Appendix B

Non-quantifiable errors

Disturbances occurring inside the receiver were hidden from the observer (e.g. air flowing through gaps, real temperature distribution, hot spots); thus, no statement about these phenomena can be made. This list provides a short summary on the observable but non-quantifiable errors that influenced the determination of $\dot{Q}_\text{in}$:

*Wind* – During the experiments, the wind speed near the CPC entrance was occasionally measured using a hand-held anemometer. The acquired data were of low spatial and temporal resolution and wind gusts could not be resolved correctly. In any case, the exact conditions in front of the CPC could not be measured due to the strong radiation prevailing during the experiments. Aside from the steady-state criterion (*cf.* Equation (4.1)), the wind data was used as an additional measure to dismiss data points. Especially at higher temperatures, the lower efficiencies lead to slower heating of the receiver’s thermal inertia (*cf.* Figure 4-1); thus, wind effects had a larger impact at high temperatures (*cf.* Figure 5-3).

Despite the roughly known wind conditions on the experiment floor, no statement about the wind conditions on the heliostat field could be made. Strong and gusty winds on the heliostat field have a strong influence on the tracking accuracy and worsen the solar beam quality significantly. To avoid damage, wind speed measurements at the tower top were close-looped to the heliostat tracking system to automatically send the heliostats to stow position if wind speeds exceeding 15 m/s were registered. This data could not be accessed.
**Smoke** – For high solar power inputs, parts of the floor surrounding the setup started to give off smoke. This was prevented by placing insulating heat-shield panels onto the affected floor parts. Also, after nightly precipitation (dew), the insulation tiles covering the tower front produced evaporation fumes at the beginning of the experiments. Evaporation fumes and smoke in front of the CPC entrance drastically affect the solar beam transmittance and could not be quantified. Still, their occurrence served as an additional measure to dismiss data points.

**Sun-shape** – A weather phenomenon referred to as “desert heat” occurred several times during the experimental campaigns. During this phenomenon, dusty air from the desert travels across the sky. The sky turns yellowish and the sun-shape appears to be larger, leading to a high circumsolar ratio and reduced solar beam quality [62]. Usually this effect was captured by the pyrheliometer, however early stages of such weather conditions, also caused by a hazy sky, were not fully resolved.

**Heliostat tracking offset** – After more than 30 years of operation the heliostat tracking system showed a distinct offset. This effect is mainly attributed to Earth’s gyroscopic nutation behavior and possible unevenness of the ground (e.g. sagging of heliostat foundations due to rain erosion) and leads to altered relative sun positions. The projected heliostat images had an offset of ~1 m, which is significant for a 0.63 m target diameter (i.e. CPC entrance). Prior to each experimental run, this offset was corrected manually but no statement about the operator’s accuracy can be made.

Conclusively, it is assumed that the receiver performance is actually better than determined in Chapter 4 due to the reduction in $\dot{Q}_{in}$ attributed to all these influences listed above.
Appendix C

Nusselt correlation for the pressure vessel surface

To account for convective heat losses on the receiver pressure vessel surface, a Nusselt correlation serving as a heat sink was applied on the simplified geometry in the CFD simulation, cf. Section 5.1. To find such a Nu correlation, a separate 3D CFD model fully representing the exact receiver geometry was built in Ansys CFX 15.0 resulting in:

\[
\text{Nu}_D = 3.444 + \frac{0.656}{1 + \left( \frac{0.492}{\text{Pr}} \right)^{\frac{9}{16}}} \cdot \text{Ra}_{D}^{\frac{1}{4}}
\]  \hspace{1cm} (C.1)

with \( D = 0.565 \, \text{m}, \) being the flange diameter. In a second step, this correlation was evaluated by mapping IR thermograms onto the receiver geometry to accurately predict the actual convective heat losses as shown in Figure C-1. An overall error of ±3% was found compared to the CFD simulation with the simplified geometry.
Figure C-1 – Model domain of the exact receiver geometry to simulate convective heat losses on the pressure vessel surface by mapping IR thermograms, cf. Figure 4-6
Appendix D

Material properties

Table D-1 provides temperature dependent material properties used in the CFD simulations, cf. Chapter 5.
Table D-1 – Material properties

<table>
<thead>
<tr>
<th>Param.</th>
<th>Value</th>
<th>( T ) (K)</th>
<th>Unit</th>
</tr>
</thead>
</table>
| \( c_{p,\text{air}}^a \) | \[
\left( A + B \cdot \left( \frac{C}{T_i} \right) \right) + D \cdot \left( \frac{E}{T_i} \right) \right) \cdot \frac{1}{28.966} \] | \( 50 \leq T \leq 1500 \) | J kg\(^{-1}\) K\(^{-1}\) |
| \( \mu_{\text{air}}^a \) | \[
\frac{A \cdot T_i^a}{1 + \frac{C}{T_i}} \] | \( 80 \leq T \leq 2000 \) | kg m\(^{-1}\) s\(^{-1}\) |
| \( k_{\text{air}}^a \) | \[
\frac{A \cdot T_i^a}{1 + \frac{C}{T_i} + \frac{D}{T_i^2}} \] | \( 70 \leq T \leq 2000 \) | W m\(^{-1}\) K\(^{-1}\) |
| \( c_{p,\text{SiC}}^b \) | \[
1110 + 0.15(T_i - 273) - 425 \cdot \exp \left( 3 \cdot 10^3 \cdot (T_i - 273) \right) \] | \( 273 \leq T \leq 2273 \) | J kg\(^{-1}\) K\(^{-1}\) |
| \( k_{\text{SiC}}^b \) | \[
52000 \cdot \exp \left( -1.24 \cdot 10^3 \cdot (T_i - 273) \right) \] | \( 273 \leq T \leq 2273 \) | W m\(^{-1}\) K\(^{-1}\) |
| \( c_{p,\text{Al2O3-SiO2}}^c \) | \[
1300 - 10 \cdot \exp \left( 5 - 3.3 \cdot 10^{-3} \cdot (T_i + 10) \right) \] | \( 77 \leq T \leq 1823 \) | J kg\(^{-1}\) K\(^{-1}\) |
| \( k_{\text{Al2O3-SiO2}}^d \) | \[
5.276 \cdot 10^{-2} \cdot \exp \left( 1.273 \cdot 10^{-3} \cdot T_i \right) \] | \( 473 \leq T \leq 1673 \) | W m\(^{-1}\) K\(^{-1}\) |
| \( c_{p,\text{steel}}^e \) | \[
414.7 + 0.2004 \cdot T_i \] | \( 293 \leq T \leq 1143 \) | J kg\(^{-1}\) K\(^{-1}\) |
| \( k_{\text{steel}}^e \) | \[
7.588 + 0.0125 \cdot T_i \] | \( 473 \leq T \leq 873 \) | W m\(^{-1}\) K\(^{-1}\) |
| \( \varepsilon_{\text{SiC}}^f \) | 0.9 | \( 423 \leq T \leq 1400 \) | - |
| \( \varepsilon_{\text{steel}}^g \) | 0.6 | | - |

\(^a\)Ref. [63]; \(^b\)Ref. [21]
\(^c\)exponential fit from values provided by Refs. [64,65]
\(^d\)exponential fit from values provided by Ref. [26]
\(^e\)linear fit from values provided by Ref. [66]
\(^f\)average value from Refs. [39,67]
\(^g\)average of experimental values measured by IR spectroscopy
Appendix E

Techno-economic analysis

The analyzed solar cycles show a trade-off characteristic, allowing a cycle to either be optimized for $\eta$ or $w$, cf. Subsection 7.4.3. To determine when to use which optimization approach, a simplified economic analysis using the payback time, $t_{pb}$, is suggested. The revenue, $R$, is defined as:

$$ R = w \cdot \dot{m} \cdot I \cdot p \cdot t $$  \hspace{1cm} (E.1)

with load factor, $l = 0.3$, electricity price, $p$, and time, $t$, while the cost, $C$, is defined as:

$$ C = I + c_v \cdot t $$  \hspace{1cm} (E.2)

with investment cost, $I$, and variable costs $c_v$. $I$ contains all the investment cost to purchase/build the power plant, e.g. tower, heliostats, power block, land purchase, etc., while $c_v$ contains all the costs that depend on $t$, e.g. taxes, operation and maintenance costs, wages, etc. Setting $R = C$ defines the payback time as:

$$ t_{pb} = \frac{I}{w \cdot \dot{m} \cdot l \cdot p - c_v} $$  \hspace{1cm} (E.3)

To put $c_v$ in relation to $I$, it is defined as an annual cost factor, $f_v$, of the initial investment cost:

$$ c_v = f_v \cdot I $$  \hspace{1cm} (E.4)
Using the values from Table 7-5 and Table 7-6, two cases can be analyzed: maximum $\eta$ vs. maximum $w$. The case optimized for $\eta$ is used as reference case, which determines the heliostat field size/cost and power block size/cost. Consequently, reducing $\eta$ in order to achieve higher $w$, leads to an increase of the heliostat field size and a reduction of the PCU size. Thus, $I$ optimized for high $w$ can be expressed as:

$$
I_{w=\text{max}} = (r_{\text{HF}} \cdot f_{\text{HF}} + r_{\text{PCU}} \cdot f_{\text{PCU}} + f_{\text{rest}}) \cdot I_{\eta=\text{max}}
$$

(E.5)

with $f_{\text{HF}}$, $f_{\text{PCU}}$, and $f_{\text{rest}}$ being the cost shares of the heliostat field, power conversion unit, and rest, which is not affected by $\eta$ and $w$, respectively, and

$$
r_{\text{HF}} = \frac{\eta_{\text{max}}}{\eta_{w=\text{max}}}; \quad r_{\text{PCU}} = \frac{w_{\text{max}}}{w_{w=\text{max}}}
$$

(E.6)

Combining Equations (E.3) – (E.6) and normalizing $t_{pb}$ for maximum $w$ by $t_{pb}$ for maximum $\eta$, the reduced payback time, $t_{\text{red}}$, is found:

$$
t_{\text{red}} = \frac{t_{pb_{w=\text{max}}}}{t_{pb_{\eta=\text{max}}}} - 1
$$

(E.7)

So, only if $t_{\text{red}}$ is negative, optimization for $w$ should be favored. This analysis results in four degrees of freedom:

(a) heliostat cost share, $f_{\text{HF}}$ (% of $I$)
(b) power conversion unit cost share, $f_{\text{PCU}}$ (% of $I$)
(c) variable cost factor, $f_v$ (% of $I$ p.a.)
(d) electricity price, $p$ (% of $I$ p.a.)

and load factor, $l = 0.3$, and $f_{\text{rest}}$ given by:

$$
f_{\text{rest}} = 1 - f_{\text{PCU}} - f_{\text{HF}}
$$

(E.8)
Table E-1 shows exemplary cost combinations and the suggested optimization strategy for the BC and CC cases. CC power plants have a significantly higher $f_{PCU}$ than BC power plants, thus optimization towards $w$ is favorable. When a certain combination of high $f_{HF}$ and low $f_{PCU}$ is reached, the optimization strategy for the BC changes because of its more pronounced trade-off behavior, cf. Table 7-5. Assuming $p = 20\%$ of $I \; p.a.$ results in electricity costs of $\sim 10 \; \text{¢/kWh}$ for a 40 M$ solar CC power plant rated at 30 MW$_{el}$ (yearly 79 GWh at $l = 0.3$).

<table>
<thead>
<tr>
<th>Power cycle</th>
<th>$f_{HF}$ (%)</th>
<th>$f_{PCU}$ (%)</th>
<th>$f_v$ (%/a)</th>
<th>$p$ (%/a)</th>
<th>opt.</th>
</tr>
</thead>
<tbody>
<tr>
<td>BC</td>
<td>40</td>
<td>15</td>
<td>4</td>
<td>20</td>
<td>$w$</td>
</tr>
<tr>
<td></td>
<td>50</td>
<td>10</td>
<td>4</td>
<td>20</td>
<td>$\eta$</td>
</tr>
<tr>
<td>CC</td>
<td>35</td>
<td>35</td>
<td>6</td>
<td>20</td>
<td>$w$</td>
</tr>
<tr>
<td></td>
<td>50</td>
<td>30</td>
<td>6</td>
<td>20</td>
<td>$w$</td>
</tr>
</tbody>
</table>

So far, the analysis shown above suffers from one specific restriction: it is assumed that the heliostat field and the power block cost scale linearly with $\eta$ and $w$, respectively, cf. Equation (E.6). In reality, it is expected that a power block running at higher $w$ is smaller but since it runs at higher temperature it may not be significantly cheaper. Reciprocally, to achieve twice the solar power input (if $\eta = 0.5 \times \eta_{ref}$), simply doubling the heliostat field size is not sufficient because heliostats added at the far side of the field, which is the only direction a field can grow, suffer from lower optical efficiencies [68]. Consequently, a correction factor or power factor should be added to Equation (E.6).
Appendix F

Optical analysis of three point-focus concentrator technologies

F.1 Motivation

Striving towards higher solar power plant efficiencies, higher temperatures become necessary to drive highly efficient combined cycles [3,6]. The minimum required temperature at which solar heat is added should exceed ~700 °C, or about 1000 K. To achieve such target temperatures above 1000 K, solar concentration ratios, $C$, exceeding 1000 suns are necessary to efficiently extract the useful heat [69]. Such concentration ratios can only be achieved by point-focus CSP system. Proven technologies are central receiver systems (solar towers) and parabolic dishes [1,52].

In this Chapter an investigation and comparison of three concentrator technologies is given for: (a) the solar tower, (b) the solar tower in conjunction with a beam-down mirror, and (c) the Scheffler dish, which is composed of two lateral parts of a parabolic dish, being a derivative of the common solar dish [70]. This analysis was carried out using the Monte Carlo ray-tracing method.

F.2 Concentrator technologies

For the analysis, the solar tower and heliostat field geometry of the solar research facility of the WIS was chosen as reference case [29]. The field consists of 64 heliostats arranged in a south-facing quarter circle with the solar tower located at its center, at latitude $L = 31.5$ °deg, cf. Figure 3-2 and Table 3-1.
Tower top – The tower top configuration refers to a target located at a height of 60 m on top of the solar tower. The target normal was set to match the average aiming vector of all 64 heliostats resulting in a downward tilt from the vertical of 35.9 °deg.

In a preliminary analysis, the flat heliostat facets were either aligned on a spherically shaped structure with a sphere diameter of twice the heliostat slant range or a parabolically shaped structure with focus length equal the slant range. No significant improvements were found when aligning the facets on a paraboloidal structure compared to a spherical. However, if the flat facets are replaced by curved facets, improvements of factor 4 by means of efficiency and concentration ratio were achieved compared to flat facets. Again, no significant difference was found between spherically and paraboloidally curved facets. Hence, the following analysis only focuses on spherically curved heliostat facets aligned on a spherically shaped heliostat structure.

Beam-down – Derived from the tower top case, two sets of beam-down configurations were investigated. Beam-down mirror configurations have the advantage of having the target located on the ground, while only a secondary reflector is mounted on the tower.

The beam-down technology uses conic section geometries with two foci: the ellipse and the hyperbola, which are rotated around the tower axis into ellipsoids and hyperboloids. While the hyperboloidal geometry has a convex mirror shape (Cassegrain reflector), the ellipsoidal case exhibits a concave shape (Gregorian reflector) \[71,72\]. Figure F-1 shows a schematic of two ellipsoidal and two hyperboloidal beam-down mirrors, which share the same center, \( C \), and focal points, \( F \). The upper focal point is located 60 m above ground (\( F_1 \)), while the lower focal point stays on ground level at the origin (\( F_2 \)). Despite having the same \( C \) and \( F \), the depicted geometries are differing in eccentricity, \( \varepsilon \), which is defined as the distance from center-to-focus, \( f \), divided by the length of the semi-major axis (center-to-vertex), \( a \):

\[
\varepsilon = \frac{f}{a} \tag{F.1}
\]
For hyperbolic curves, $\varepsilon_{\text{hyp}} > 1$, and for elliptic curves, $0 < \varepsilon_{\text{ell}} < 1$ has to be satisfied. If the curvature at the vertex of the ellipse and the hyperbola are identical, they relate to each other by:

$$\varepsilon_{\text{ell}} = \frac{1}{\varepsilon_{\text{hyp}}}$$  \hspace{1cm} (F.2)

To allow comparison, $\varepsilon$ was set according to Equation (F.2) for each pair of ellipsoidal and hyperboloidal beam-down mirrors. Figure F-1 emphasizes this relationship: the sunlight enters from the right (not displayed) and is reflected towards $F_1$ by a primary concentrator, i.e. the heliostat field. The identical curvatures of each pair of $\varepsilon$ cannot be seen by the naked eye but the rays hitting $F_2$ appear axis-symmetrical.

Figure F-1 – Schematic of quadric beam-down reflectors and their eccentricity relationship, cf. Equation (F.2). Sunlight enters from the right (not displayed), is reflected by the heliostat field, and concentrated towards the upper focus, $F_1$. Only the upper hyperbola branch is shown.
Scheffler dish – Parabolic dishes provide very high $C$ at relatively small-scale compared to tower top applications [73]. Also, they show no distinct daytime dependency of $\eta$ and $C$ caused by cosine losses. State-of-the-art parabolic dishes use two-axis tracking systems to track the sun. Therefore, the target has to be mounted at the focus in mid-air, following the movement of the tracking dish. This procedure is unpractical, since the hot working fluid has to be conducted over multiple moving joints. To overcome this issue, W. Scheffler proposed to only build two lateral parts of a parabolic dish with their foci located on the ground, referred to as Scheffler dishes [70,74]. As shown in Figure F-2, the centers of the resulting top and bottom dishes, $T$ and $B$, respectively, and the target located at their foci, $F$, are installed co-axially and parallel to Earth’s rotational axis. This results in a one-axis tracking system, in which the dishes perform one full revolution per day to track the sun, leading to a simplification of the tracking device to an ordinary clockwork.

Since the target remains at a fixed position on the ground, the shape and tilt of the paraboloidal sections have to be adjusted for seasonal changes. Figure F-2 shows a schematic of the Scheffler dish working principle for summer and winter solstices. For these two seasons, the top and bottom dish parabolas (through $T$ and $B$, respectively) differ in shape significantly because the parabola symmetry axes have to remain parallel to the incoming sunlight while BFT remains constant.
The focus and the centers of the top and bottom dishes are aligned co-axially and parallel to Earth’s axis. For seasonal changes of the solar altitude angle, $\alpha$, the curvatures of the parabola sections are adjusted: the curvature of the top dish (solid line) changes from low to high when switching from winter to summer while the bottom dish (dashed line) exhibits the opposite.

F.3 Monte Carlo ray-tracing

The MC analysis was performed using an in-house code [32]. Similar to Section 3.4, uniformly distributed sunrays subtending a solid half angle of 4.65 mrad are incident on the heliostats from the sun position at a given time and day, determined from known sun-path correlations [33]. The reflectivities of all specular surfaces is set to unity, assuming the ideal case. The target is modelled as a black body with $\rho = 0$. Shading/blocking caused by adjacent heliostats is taken into account. Characteristic surface dispersion errors, $\sigma$, in the range of 0 – 6 mrad are implemented for the specular surfaces of the primary concentrators, whereas errors up to 12 mrad are set for the beam-down mirror to account for its more complex shape.

Each heliostat facet is modelled individually, with a ray density of $10^5$ rays per facet. The target is divided into 50 radial segments to obtain a good resolution.
of the flux distribution, \( q'' \), on the target. For the Scheffler-dishes, the equivalent area and ray density of one heliostat are preserved.

All simulations were performed for a baseline situation encountered on January 21, 11:53 local time, at which the minimum cosine loss of the entire heliostat field of 1.9% is reached with the solar altitude angle, \( \alpha = 38.4^\circ \) deg. Another minimum can be found on November 22, 11:29, with the same cosine loss.

**F.4 Results**

To characterize optical concentrator systems, the achievable mean concentration ratio, \( C_{\text{mean}} \), and the optical efficiency, \( \eta \), are used. \( C_{\text{mean}} \) is defined in Equation (F.4) as a function of the target radius, \( R \):

\[
C(r, \varphi) = \frac{\dot{q}''(r, \varphi)}{I_{\text{DNI}}} \tag{F.3}
\]

\[
C_{\text{mean}}(R) = \frac{\int_0^{2\pi} \int_0^R \dot{q}''(r, \varphi) r \, dr \, d\varphi}{\int_0^{2\pi} \int_0^R \frac{I_{\text{DNI}}}{2\pi R^2} \, dr \, d\varphi} \tag{F.4}
\]

The optical system efficiency is defined similarly, using the maximum incoming solar radiative power input, \( \dot{Q}_{\text{sun}} \), defined as the direct normal irradiance, \( I_{\text{DNI}} \), multiplied by the reflective surface area of the primary concentrators:

\[
\eta(R) = \frac{\int_0^{2\pi} \int_0^R \dot{q}''(r, \varphi) r \, dr \, d\varphi}{\int_0^{2\pi} \int_0^R \frac{I_{\text{DNI}} \cdot A_{\text{refl}}}{\dot{Q}_{\text{sun}}} \, dr \, d\varphi} = \frac{\dot{Q}_{\text{bar}}(R)}{\dot{Q}_{\text{sun}}} \tag{F.5}
\]

In the numerical analysis, the integrals are replaced by sums over the discretized target flux maps resulting from the numerical MC analysis.
**Tower top** – Figure F-3 shows $C_{\text{mean}}$ and $\eta$ as a function of $R = 0 - 1.5$ m. A set of three different heliostat surface dispersion errors, $\sigma_h = 0$, 3, and 6 mrad were analyzed leading to peak $C_{\text{mean}} \approx 4370, 2990,$ and 1130, respectively. On one hand, the required $C_{\text{mean}} > 1000$ suns was reached in all three cases for $R < 0.45$ m, which means that the analyzed tower top configurations do not necessarily require a secondary concentrator to reach $C_{\text{mean}} > 1000$ suns. If desired, a secondary concentrator (e.g. CPC) can be incorporated to further boost the solar concentration ratio. On the other hand, $\eta$ becomes significantly smaller for small radii. To obtain peak $\eta = 94.8\%$, $R > 0.8$ m is required. The $\sigma_h = 6$ mrad case displays a rather poor performance by means of $C$ and $\eta$, so $\sigma_h < 6$ mrad is desired. To allow comparison of the tower top, beam-down, and Scheffler configuration, $\sigma_h = 3$ mrad was kept for all primary concentrators in the following analyses.

![Figure F-3](image)

Figure F-3 – Mean concentration ratio and optical efficiency of the tower top configuration vs. target radius and heliostat surface error. $C > 1000$ suns is highlighted.

**Beam-down** – While the tower top geometry with only one set of reflectors (i.e. heliostats) was already given, the eccentricity of the beam-down mirror served as an additional degree of freedom, which had to be assessed first. The search for the most suitable $\varepsilon$ had to be limited either to the hyperboloidal ($\varepsilon > 1$)
or ellipsoidal ($\varepsilon < 1$) case. Figure F-4 displays the resulting beam-down mirror diameter, $d$, and tower height, $h$, both defined in Figure F-1 as a function of $\varepsilon$. As Figure F-1 already suggested, an ellipsoidal beam-down concept becomes significantly larger compared to the hyperboloidal case if the upper focus $F_1$ is kept at constant height. Theoretically, if $\varepsilon \sim 1$ is approached, the baseline height of 60 m would be reached for both concepts. Due to their considerably smaller size, only the hyperboloidal beam-down mirrors are considered as a viable option.

![Figure F-4](image)

Figure F-4 – Beam-down mirror diameter and tower height for hyperboloidal ($\varepsilon > 1$) and ellipsoidal ($\varepsilon < 1$) beam-down mirrors as a function of their eccentricities, cf. Figure F-1

Compared to a tower top configuration where tower shading is small, the beam-down mirror shows significantly higher shading losses due to its larger size. Figure F-5 shows $\eta$ vs. $\varepsilon$ to elucidate the effect of shading by the beam-down mirror. For $\varepsilon$ larger than $\sim 1.75$, a steep decrease in $\eta$ can be observed. Depending on the target radius, this decrease is already observed at smaller $\varepsilon$. Since $\varepsilon = 1.75$ shows good performance also for $R = 1.0 - 1.5$ m, it was chosen as the most suitable $\varepsilon$ for a hyperboloidal beam-down mirror in conjunction with the given heliostat field.
With $\varepsilon = 1.75$ resulting in $h = 53$ m and $d = 20.3$ m, Figure F-6 displays $C_{\text{mean}}$ and $\eta$ for $R = 0 – 3.0$ m. While $\sigma_h$ was kept at 3 mrad, three different beam-down mirror surface errors, $\sigma_{BD} = 0, 6, \text{ and } 12$ mrad were analyzed, leading to peak $C_{\text{mean}} = 358$ suns for $\sigma_{BD} = 0$ mrad. $\eta$ was determined without beam-down mirror shading effects in order to allow an idealized comparison of the tower top and beam-down case. Despite $\eta$ approaching 94.8% for a sufficiently large $R$, $C_{\text{mean}}$ does not reach 1000 suns even for perfect beam-down mirror reflection. Consequently, to be applicable in a solar gas turbine setup, a tertiary concentrator located at ground level becomes unavoidable, cf. Ref. [75]. Apparently, the surface quality of the beam-down mirror is only of minor importance, allowing higher manufacturing tolerances or a faceted construction, which is beneficial for such a complex shape. The main cause of the low $C$ is the error propagation after secondary reflection at the beam-down mirror caused by $\sigma_h$ and $\sigma_{\text{sun}} > 0$ and the increased optical path length leading to decreased solar beam quality.
Scheffler dish – The analysis of the Scheffler dish only investigates the bottom dish during the baseline situation. The top dish shows the same results as the bottom dish, but its occurrence is shifted by half a year. Figure F-7 shows $C_{\text{mean}}$ and $\eta$ for $R = 0 – 0.3$ m. A set of three different dish surface errors, $\sigma_d = 0$, 3, and 6 mrad were analyzed, leading to peak $C_{\text{mean}} \approx 8990$, 5370, and 2400, respectively. In the investigated range of $\sigma_d$, the Scheffler dish meets the required concentration ratio. The Scheffler dish shows much higher cosine losses because of the larger angle between target and Sun compared to the tower top case. Peak $\eta = 56.2\%$ is already reached at $R = 0.1$ m, at which $C_{\text{mean}}$ drops below 1200 but still reached the requirement. Due to its one-axis tracking principle, the cosine loss remains constant during the day, opposed to a heliostat field where it changes throughout the day.

Having the advantage of the target located on the ground, the disadvantage is found in the dish curvature, which has to be adjusted to account for seasonal changes. To assess the sensitivity of the dish curvature, simulations were performed in which the curvature does not match the current day by up to seven days, referred to as mismatch (MM). With $\sigma_d$ being set to zero, a mismatch
by one day has a similar effect as a surface error of 3 mrad. Increasing the mismatch to one full week, the performance of the Scheffler dish is reduced drastically because the focus misses the center of the target and is shifted to higher $R$, cf. Figure F-7. Because of this significant mismatch effect, $\sigma_d = 6$ mrad was considered as reasonable reference case for comparison.

Figure F-7 – Mean concentration ratio and optical efficiency of the bottom Scheffler dish vs. target radius and mirror surface error. The dotted lines represent mismatched mirror curvatures, which have not been adjusted for the specified number of days for $\sigma_d = 0$ mrad. $C > 1000$ suns is highlighted.

*Comparison* – Figure F-8 shows a comparison of the three analyzed concentrator concepts: tower top (TT), beam-down (BD), and Scheffler dish (SCH) for $R = 0 – 1.5$ m. The surface area of the Scheffler dish was already set equivalent to one heliostat surface area and only the $\varepsilon = 1.75$ beam-down mirror geometry with same upper focus location as the tower top case are shown. The surface errors were set to 3 mrad for heliostats and 6 mrad for the more complex beam-down mirror and Scheffler dish.
A comparison of the tower top and the beam-down concept shows that the tower top concept outperforms the beam-down mirror’s solar concentration ability by about one order of magnitude. A similar effect is found in the efficiency plot. Thus, a tertiary concentration with minimum $C > 10$ is required in a beam-down application.

The Scheffler dish almost reaches the same performance as the tower top configuration by means of $C$. However, if the surface quality was better than 6 mrad, the Scheffler dish would perform better than the tower top concept. The down side is the decrease of $C$ to below 1000 for $R > 0.1$ m. It appears that the Scheffler dish shows a very steep increase in $\eta$, and thus outperforms the tower top concept for $R < 0.56$ m, while for $R > 0.56$ m the tower top shows the highest performance by means of $C$ and $\eta$. It is noted that the heliostat field geometry and the Scheffler dish surface were chosen arbitrarily, hence Figure F-9 contradicts this finding by providing a comparison of the concentrator concepts using the target area normalized by the total reflective primary surface area. Conclusively, the tower top configuration outperforms the other concepts in $C$ and $\eta$ for all target sizes. Table F-1 summarizes the peak values of $C_{\text{mean}}$ and $\eta$. 

Figure F-8 – Comparison of the mean concentration ratio and optical efficiency of the tower top (TT), beam-down (BD), and Scheffler dish (SCH) concept vs. target radius, with parameters summarized in Table F-1. $C > 1000$ suns is highlighted.
and the parameters: surface quality, \( \sigma \), of the heliostat, beam-down, and Scheffler dish mirrors; and eccentricity, height, and diameter of the beam-down mirror, \( \varepsilon \), \( h \), and \( d \), respectively.

![Graph showing comparison of mean concentration ratio and optical efficiency](image)

Figure F-9 – Comparison of the mean concentration ratio and optical efficiency of the tower top (TT), beam-down (BD), and Scheffler dish (SCH) concept vs. target-to-reflector area fraction, with parameters summarized in Table F-1. \( C > 1000 \) suns is highlighted.

Table F-1 – Peak concentration ratios, peak efficiencies, and parameter summary for the three analyzed optical concepts

<table>
<thead>
<tr>
<th>Parameter</th>
<th>tower top</th>
<th>beam-down</th>
<th>Scheffler</th>
</tr>
</thead>
<tbody>
<tr>
<td>Peak ( C_{\text{mean}} )</td>
<td>4372</td>
<td>359</td>
<td>8985</td>
</tr>
<tr>
<td>Peak ( \eta )</td>
<td>94.8%</td>
<td>94.8%</td>
<td>56.2%</td>
</tr>
<tr>
<td>Peak ( C_{\text{mean}} ) (in comparison)</td>
<td>2992</td>
<td>206</td>
<td>2403</td>
</tr>
<tr>
<td>( \sigma ) (in comparison)</td>
<td>3 mrad</td>
<td>6 mrad</td>
<td>6 mrad</td>
</tr>
<tr>
<td>( \varepsilon )</td>
<td></td>
<td></td>
<td>1.75</td>
</tr>
<tr>
<td>( h )</td>
<td></td>
<td></td>
<td>53 m</td>
</tr>
<tr>
<td>( d )</td>
<td></td>
<td></td>
<td>20.3 m</td>
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F.5 Summary and conclusion

Using the Monte Carlo ray-tracing method, the optical performance of a tower top, beam-down, and Scheffler-dish concept were analyzed by means of solar concentration ratio and optical efficiency. While the tower top and beam-down concepts were based on the heliostat field and tower geometry of the WIS, the Scheffler dish size was chosen arbitrarily to be equivalent to the size of one heliostat. The paramount goal was to reach concentration ratios exceeding 1000 suns to evaluate the applicability of the optical system if used to drive gas turbine cycles operating at >1000 K solar thermal heat input.

Most promising results were obtained with the tower top concept, which outperforms the beam-down concept by up to one order of magnitude. Still, the main advantage of the beam-down concept is that the receiver array and power block can be built on the ground, but requiring a tertiary concentrator. Since all the calculations were performed using a mirror reflectivity of $\rho = 1$, the drawback of using secondary or tertiary concentrators can be found in $\eta$, which scales $\propto \rho^N$, with $N$ being the average number of reflections. If multiple serial reflections become necessary or if concentrators are used that allow more than one reflection, such as CPCs [20], the beam-down concept will show a decline in the average $\eta$. Considering the increased convective heat losses of such a receiver, whose aperture faces upward, the beam-down concept cannot compete against tower top and Scheffler dish applications.

The main advantages of the Scheffler dish are its promising concentration ratios, the target, which can be located on the ground, and its daytime independent performance. Main drawback is the comparably low optical efficiency, which is further deteriorated by the necessary daily adjustments of the mirror curvature. Technically, this curvature adjustment might be achievable by using evacuated mirror foils [76,77]. However, the Scheffler dish concept cannot be considered as a proven state-of-the-art technology yet and difficulties may arise. So far, only the bottom dish of the Scheffler configuration has been analyzed. If both, the upper and lower Scheffler dish shall be used, a dual-cavity receiver concept is required to accept rays coming from two directions, cf. Figure F-2. Considering the desired power range of a power plant, the Scheffler dish is
a modular design, which can be scaled by combining multiple dishes. Depending on the dish size, net absorbed power levels of up to 50 kW\textsubscript{th} per dish are considered reasonable [74].

Conclusively, the tower top configuration shows the most promising results for large-scale gas turbine applications. Keeping in mind that dishes are limited to smaller sizes than heliostat fields, the Scheffler dish still provides a good option for small-scale power generation plants but is very sensitive to mismatches of the mirror curvatures.
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References


References


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