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A general method to analyze the thermal performance of multi-cavity concentrating solar power receivers

Austin Fleming [a], Charles Folsom [a], Heng Ban [a,*], Zhiwen Ma [b]

[a] *Mechanical & Aerospace Engineering Department, Utah State University, 4130 Old Main Hill, Logan, UT 84322-4130, United States*
[b] *National Renewable Energy Lab, Golden, CO 80401, United States*

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Abstract

Concentrating solar power (CSP) with thermal energy storage has potential to provide grid-scale, on-demand, dispatchable renewable energy. As higher solar receiver output temperatures are necessary for higher thermal cycle efficiency, current CSP research is focused on high outlet temperature and high efficiency receivers. The objective of this study is to provide a simplified model to analyze the thermal efficiency of multi-cavity concentrating solar power receivers. The model calculates an optimal aperture flux that maximizes the local efficiency, constrained by a maximum receiver working temperature. Using this flux, the thermal efficiency, receiver temperature, and heat transfer fluid (HTF) temperature are calculated based upon an optimized flux distribution. The model also provides receiver design and HTF heat transfer requirements to achieve the necessary overall thermal efficiency. From the results, possible HTFs can be investigated to determine which ones are feasible. A case study was performed on a multi-cavity tube receiver design to demonstrate the use of the model. The case study receiver design had an effective absorptivity of 99.8%, and was modeled with conservative values for thermal constraints. It was found that a HTF with a minimum convection coefficient between 250 and 500 W m^{-2} K^{-1}, depending on the convective heat transfer to the environment, is necessary to achieve a thermal efficiency greater than 90% for the receiver. The general model can provide a design guideline for attainable thermal efficiencies of multi-cavity concentrating solar power receivers given thermal constraints and heat transfer conditions.

Keywords: Concentrated solar power; Multi-cavity receiver; Thermal analysis

1. Introduction

The key challenge to implementing grid-scale concentrating solar power (CSP) is to reduce its cost and couple it with thermal energy storage to offer on-demand, dispatchable electricity. Research has been conducted on the logistics and cost benefits of coupling CSP with thermal energy storage (Ma et al., 2014, 2011; Flueckiger et al., 2014). Through the SunShot Initiative, the US-DOE has identified areas of research to reduce the cost of CSP to compete with coal and nuclear power at a cost of 6 ¢/kW h (CSP Component Research and Development, 2015). These areas of research include: improved solar fields with respect to efficiency and cost, more advanced heat transfer fluids capable of a wide operating temperature range, improved thermal energy storage (TES) capability, cooling technology, and advanced receiver designs capable of high thermal efficiency and higher heat transfer fluid (HTF) outlet temperatures than currently available.

* Corresponding author. Tel.: +1 435 797 2098.
E-mail address: heng.ban@usu.edu (H. Ban).

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The levelized cost of energy from CSP can be decreased by reducing the manufacturing costs and increasing the efficiency of CSP plants.

The current state-of-the-art receiver is considered to be a panel receiver consisting of tubular panels with molten salt as the HTF. These have proven to be reliable and are most common in operating CSP plants today, but to date they are limited to less than 90% thermal efficiency and outlet temperatures of 565 °C (Bradshaw et al., 2002; Romero et al., 2002). While these CSP plants have improved, they still are not cost competitive to provide mainstream power production.

Many CSP power tower designs have been evaluated both with simulations and experiments. These include tubular, cavity, multi-cavity, volumetric receivers and direct absorbing receivers with working fluids including but not limited to: steam, molten salt, molten metal, gas, and particles (Behar et al., 2013; Ho and Iverson, 2014; Avila-Marin, 2011; Fleming et al., 2015; Martinek and Ma, 2014; Carotenuto et al., 1993; Near-Blackbody Enclosed Particle Receiver, 2015). Most commonly, the receiver analysis is conducted using detailed analysis tools such as computational fluid dynamics (CFD), finite element modeling, and Monte Carlo ray tracing. While these techniques are accurate, they are computationally expensive and often impose restrictions on the size of the geometry that can be modeled. This often limits the analysis to localized regions in the receiver. Some receiver designs can be approximated as an isothermal receiver; in reality however, there is a temperature distribution that varies throughout the receiver due to the varying local flux intensities and HTF temperatures. Furthermore, if the overall receiver efficiency is of interest, it is often difficult to obtain using the detailed computational analysis techniques discussed above because they require modeling the entire receiver which is computationally expensive.

This study focuses on developing an analysis technique that can quantify the thermal performance of CSP receivers quickly without the need for extensive detailed computational analysis. The technique developed will also allow for quantitative comparisons between receiver designs using different geometries, working fluids, and sizes. An application of this technique will be demonstrated on a case study for a receiver design to show how it can be applied to a specific geometry and operating conditions. In this case study it will be demonstrated how more accurate methods, such as Monte Carlo ray tracing and CFD, can be incorporated in the technique to increase the accuracy.

2. Method and procedure

The technique development is generalized as much as possible to be applicable to many multi-cavity receiver geometries, HTFs, and operating conditions. The motivation for this technique is to develop a method with the ability to rapidly analyze the thermal performance of a CSP receiver design. Providing this ability enables engineers and researchers to easily determine the feasibility of a receiver design to meet design requirements. Furthermore, it can be used to guide researchers to focus on specific areas where design improvements can be made, or which components the receiver performance is most sensitive to. The technique development process will first establish the assumptions and constraints of the procedure. Next, the analysis procedure will be discussed along with the motivation for each result. Finally, the implications and applications of the technique will be discussed.

The only receiver cavity geometry constraint for the technique presented here is an indirect absorbing receiver in which the aperture area (A_{aper}) of the receiver and the heat transfer area of the HTF (A_{htf}) can be clearly defined, such that $A_{htf} \geqslant A_{aper}$. If the receiver consisted of a flat plate with solar flux incident on one side and the HTF on the other, then the aperture area and heat transfer area would be the same. However, in other instances where the surface absorbing radiation is not at the aperture of the receiver, a larger A_{htf} can be obtained. For example, if the absorbing surface was a series of parallel tubes (diameter D and length L) with the edges touching, the aperture area for one tube would be $D{\cdot}L$, whereas the heat transfer area would be $\pi DL/2$, since only one side of the tube is illuminated. High thermal efficiency is possible with a multi-cavity receiver by utilizing a heat transfer area much larger than the aperture area. This results in a lower surface temperature by spreading the flux across the heat transfer area thereby reducing the radiation and convection losses to the environment. In this study nothing else is specified about the geometry of receivers other than the ratio of A_{htf}/A_{aper} (referred to as the area ratio, A_{ratio}, for the remainder of the paper). By keeping the analysis general, the results are comparable to many different designs of surface absorbing receivers.

This simplified model evaluates the energy balance in a small control volume (CV) at the aperture of the receiver, with no consideration to the geometry of the receiver (see Fig. 1). Since the CV being evaluated is small, the aperture flux, receiver temperature, and HTF temperature can be considered a constant throughout the CV. In the CV, the working fluid is assumed to be the same temperature as it was at the inlet. A grid convergence study was performed on the size of the CV to ensure the accuracy of the constant T_{htf} and T_R inside the CV assumption. Eq. (1) is therefore derived by performing an energy balance throughout the CV, assuming energy in as flux from the solar field and losses due to re-radiation of the receiver, natural convection losses to the environment, and energy transferred to the HTF.

$$A_{aper}q''_{aper}\eta_{absorb} = \varepsilon_{eff}\sigma A_{aper}(T_R^4 - T_{sur}^4) + h_{amb}A_{aper}(T_R - T_{amb}) + h_{htf}A_{htf}(T_R - T_{htf}) \quad (1)$$

Here, A_{aper} is the aperture area of the receiver, q''_{aper} is the flux from the solar field, η_{absorb} is the absorption efficiency of the receiver, ε_{eff} is the effective emissivity of the receiver