Coupled Optical-Thermal-Flow Analysis of a Pressurised Cavity-Air-Receiver for Concentrating Solar Power

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Introduction

The increasing energy demands of the world population is predominantly met by fossil fuel-based energy sources which leaves enormous amount of carbon footprints in the environment. This has led to increase in the average earth temperature by acting as a greenhouse gas. To reduce the carbon footprint and at the same time meeting the energy demands of the population, focus is towards harnessing renewable energy sources. Solar energy has enormous potential A solar thermal power plant comprises of an optical system, heat source and power block. Optical system collects direct sunlight and concentrates it onto the receiver. The receiver intercepts the concentrated sunlight and converts it to heat. Finally, the heat is converted to the thermal energy of the working fluid that drives the power cycle, thus generating electricity. However, currently the cost of electricity generation from concentrating solar power is much higher compared to conventional fossil fuels [1]. The reason being low efficiency of CSP plants which may be attributed to lower source temperatures. Currently, researchers are looking for efficient supercritical carbon dioxide (s-CO2) based closed loop Brayton cycle. As direct heating of s-CO2 is difficult owing to sharp variation in its thermophysical properties, indirect heating using pressurized air could be a possible option. This study deals with coupled optical, computational fluid dynamics and heat transfer analysis of a cavity air receiver used for converting concentrated solar radiation to heat using pressurized air. COMSOL Multiphysics® 5.3a was used to perform the coupled analysis.

Theory

Concentrated sunlight from the optical concentrator, Scheffler dish is intercepted by the receiver. Receiver absorbs and converts radiation to thermal energy of the working fluid used for the power cycle. Efficient receiver provides higher temperatures at which heat transfer occurs thus increasing the exergy and leading to increased power cycle efficiency. Receiver modelling helps in providing guidelines for such efficient receiver designs. To analyse the input energy to the cavity receiver, ray tracing analysis is carried out. The flux map obtained from the ray tracing analysis is then coupled to carry out the flow and heat transfer analysis providing the velocity and temperature distributions in the receiver. The outlet air temperature is analysed as a receiver performance indicator.

In this study, a cavity-air-receiver is analysed, although there are various receiver configurations possible. The receiver design is basically a cylindrical cavity with one end dome closed while the other end open. The annular space is filled with porous ceramic for increasing the surface area for heat transfer. Cavity receiver gets heated after absorbing the concentrated solar radiation. Part of the heat transfer occurs by conduction to the porous ceramic and rest is lost as reradiation from cavity surface as well as by natural convection to the ambient.

Optical modelling

$$\frac{d\boldsymbol{k}}{dt} = -\frac{\partial \omega}{\partial q}\,;\ \frac{d\boldsymbol{q}}{dt} = \frac{\partial \omega}{\partial k}$$

where **k** is wave vector, **q** is position vector and ω is angular frequency.

Geometrical optics module is used to model the concentrated solar radiation falling on the cavity surface. The Scheffler concentrator geometry is imported as CAD file from SOLIDWORKS which is defined as illuminated surface boundary with the initial ray direction vector and total source power specified. The cavity surface is specified as deposited ray power surface. This ray tracing study is carried out in separate component.

Flow Modelling

$$\nabla.\mathbf{u} = \mathbf{0}$$

$$\nabla.\left[-p\mathbf{I} + \frac{\mu}{\epsilon_p}\left(\nabla\mathbf{u} + (\nabla\mathbf{u})^T\right)\right] - \left(\frac{\mu}{k}\boldsymbol{u}\right) = \mathbf{0}$$

Here $\boldsymbol{u}$ is velocity vector, p is pressure, μ is dynamic viscosity, ϵ_p is porosity, k is thermal conductivity.

Brinkman equations are used to model flow through annular porous region. Pressurized air at 20 bar and 300 K is passed through the porous ceramic. Mass flow rate is specified at the inlet. The porous medium is assumed Silicon Carbide with geometric properties like permeability, porosity and pore size obtained from [2].

Heat Transfer Modelling

Solid phase: $$\nabla.\left(\theta_p k_s \boldsymbol{\nabla} \mathrm{T}_s\right) + h_{\mathrm{sf}}(\mathrm{T_f} - \mathrm{T_s}) = \mathbf{0}$$

Here $\theta_p$is 1- ϵ_p, k_s is solid thermal conductivity, h_{sf} is interstitial heat transfer coefficient, T_s is solid temperature distribution, $\mathrm{T_f}$ is fluid temperature distribution.

Fluid phase: $$(1 - \theta_\mathbf{p})\rho c_p \boldsymbol{u}_\mathrm{f}.\nabla \mathrm{T_f} + \nabla.\mathbf{q} = \nabla.\left((1 - \theta_p)k_\mathrm{f}\boldsymbol{\nabla}\mathrm{T_f}\right) + h_{\mathrm{sf}}(\mathrm{T}_s - \mathrm{T_f})$$

Here ρ is fluid density, c_p is fluid heat capacity at constant pressure, $\mathbf{q}$ is conductive heat flux.

Closure equation: $$Q_s = \frac{q_{sf}}{\theta_p}(\mathrm{T_f} - \mathrm{T}_S) \quad ; \; Q_\mathrm{f} = \frac{q_{sf}}{1-\theta_p}(\mathrm{T_s} - \mathrm{T_f});$$

Here, q_{sf} is the volumetric heat transfer coefficient.

Heat transfer in solid and fluid physics modules are used to obtain air and porous solid temperature distribution in the receiver. The reason for different temperature distributions is that, initially the receiver is heated until it reaches a steady state temperature then the pressurized air at ambient temperature is passed, thus the incoming cold air sees the heated porous solid. Thus, cold fluid initially enters at 300K while the porous solid at steady state temperature is at 350K. Heat transfer in solid module has heat flux as boundary condition which couples the flux map obtained from the ray tracing analysis and models convective as well as re-radiative heat loss from the receiver cavity surface. This coupling between ray tracing and heat transfer is carried out using the general extrusion tool, defined in the component where ray tracing analysis is carried out.

Simulation Results

Ray Tracing Results

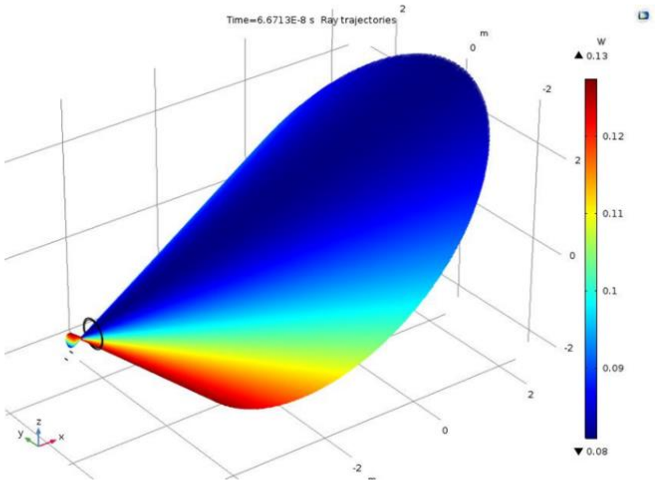

FIGURE 1. Ray tracing analysis of the Scheffler dish and the receiver

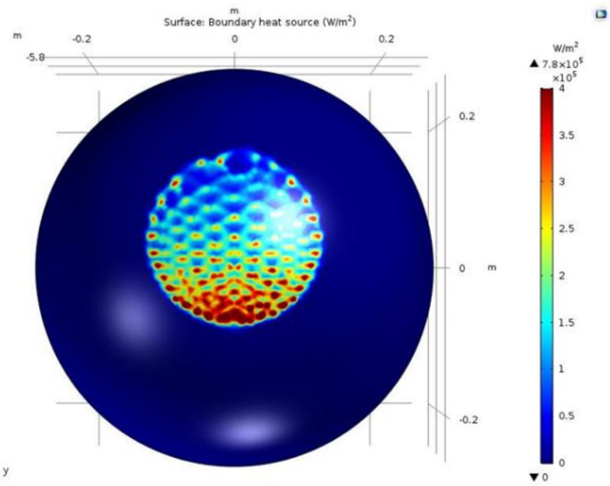

FIGURE 2. Flux map obtained on the curved cavity surface

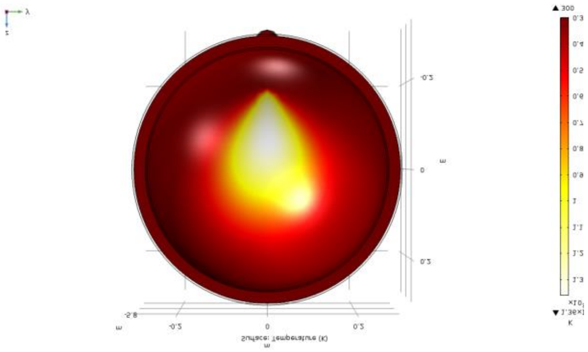

FIGURE 3. Air temperature distribution for ṁ = 0.002 kg/s

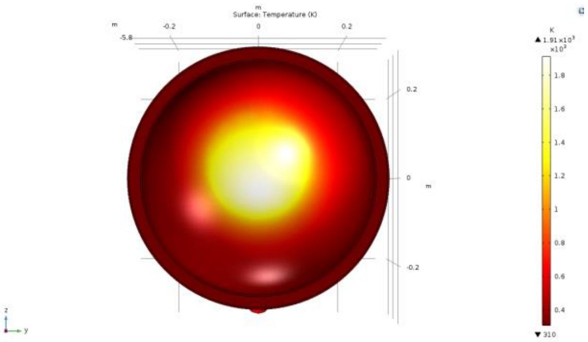

FIGURE 4. Porous medium temperature distribution for ṁ = 0.002 kg/s

Mass Flow Rate[kg/s]	**Air Outlet Temperature[K]**	**Porous Solid Temperature[K]**
0.002	426.0	604.9
0.004	412.7	597.7
0.006	377.5	593.9
0.008	340.5	504.3

Table 1. Tabulated porous solid and outlet air temperatures for a range of mass flow rates

Conclusions

The primary concentrator used for the study, namely Scheffler dish was geometrically modelled for ray tracing analysis. Ray tracing analysis captures non-uniform flux distribution on the receiver surface as opposed to a constant flux value being used as boundary condition in the thermal analysis. For a range of flow rates, the air temperature distribution was observed for a given porosity. Loss modelling of the receiver includes natural convection loss and radiative loss from the cavity surface. Local Thermal Non-Equilibrium modelling of the porous medium accurately captures the convective heat transfer within the porous domain compared to equilibrium modelling. Enhancement of convective heat transfer within the porous domain can directly affect the performance of the receiver thus appropriate porous absorbers must be fabricated with good specific surface area, pore diameter and permeability for increasing the receiver efficiency.

References

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