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Numerical and experimental evaluation and optimization of ceramic foam as solar absorber – Single-layer vs multi-layer configurations

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HIGHLIGHTS

- The thermal efficiency of ceramic foam absorbers depends strongly on their porosity.
- The highest possible porosity should be chosen for future solar receiver designs.
- A flat efficiency maximum can be observed for cell densities between 30 and 50 PPI.
- An optimized single-layer configuration is the absorber of choice.
- Multi-layer configurations do not have any advantage regarding thermal performance.

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ABSTRACT

This work targets the numerical and experimental evaluation of ceramic foam as solar absorber material for solar thermal power generation. Two different 1-D model types with local thermal non-equilibrium (LTNE) have been developed independently at CENER and Fraunhofer-IKTS. The modeling of radiation propagation inside the foam is considered via two approaches. One approach is based on a discrete-ordinate solution of the radiation transport equation; the other imposes the solar flux defining an exponential attenuation, as derived from Bouguer's law, and considering thermal radiation transport according to Rosseland's diffusion approximation.

Both models have been successfully checked for consistency against experimental data obtained at a 4 kW solar simulator. Then, the models have been run applying automatic scripts, performing a large number of parameter variations, optimizing for the absorber thermal efficiency. It is important to note that single, double and triple layer absorber configurations have been studied, since previous works found that a decreasing porosity in direction of flow can enhance absorber performance.

The parametric optimization studies have shown that the porosity of the foam is strongly related to the obtained thermal efficiency. The higher the porosity of the foam, the higher is also the absorber thermal efficiency. A broad plateau-like efficiency maximum can be observed for cell densities between 30 and 50 PPI (pores per inch). When applying a multi-layer configuration, no significant correlation can be observed between efficiency and the properties of the second or third layer. Only the parameters of the first layer seem to determine the thermal performance. This leads to the conclusion that an optimized single-layer configuration is the absorber of choice. If necessary, a second layer could be applied to satisfy mechanical stability aspects.

1. Introduction

Solar thermal power, also known as concentrated solar power (CSP) or solar thermal electricity (STE), is a renewable energy sector with great potential, as it directly harnesses the abundant amount of solar energy incident on planet earth. CSP plants capture the sun's direct normal irradiation (DNI), concentrate it onto a receiving surface, and transform the absorbed heat into mechanical work and subsequently, into electric energy by using state-of-the-art thermodynamic power cycles. Unlike other renewable energy sectors (such as wind or

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Nomenclature

A	area (m^2)
$A_{surface}$	surface area for convective heat transfer (m^2)
$A_{receiver}$	aperture area of receiver (m^2)
d_c	cell diameter (m)
d_h	hydraulic diameter of foam (m)
d_w	window diameter (m)
Hg	Hagen number (–)
h_{abs}	absorber thickness (m)
$h_{ambient}$	convective heat transfer coefficient between ambient air and receiver front face (W/(m^2 K))
$h_{a,i}$	upstream specific enthalpy at the left boundary of control volume i (J/kg)
$h_{b,i}$	upstream specific enthalpy at the right boundary of control volume i (J/kg)
h_{fi}	convective interstitial heat transfer coefficient between solid and fluid at node i (W/(m^2 K))
h_i	specific enthalpy of control volume i (J/kg)
h_v	convective volumetric heat transfer coefficient between solid and fluid (W/(m^3 K))
I_0	solar flux incident on the absorber's front face (W/m^2)
I_i	solar flux incident at node i (W/m^2)
K	extinction coefficient (m^{-1})
k_f	thermal conductivity of fluid (W/(m K))
k_{eff}	effective thermal conductivity (W/(m K))
k_{con}	conductive contribution of effective conductivity (W/(m K))
k_{rad}	radiative contribution of effective conductivity (W/(m K))
k_s	pure solid thermal conductivity (W/(m K))
l	edge length of tetrakaidecahedron – idealized cell geometry (m)
m_i	total fluid mass inside the control volume i (kg)
$\dot{m}_{a,i}$	mass flow at left boundary of control volume i, if entering positive else negative (kg/s)
$\dot{m}_{b,i}$	mass flow at right boundary of control volume i, if leaving positive else negative (kg/s)
Nu	Nusselt number (–)
p_i	pressure within control volume i (Pa)
q	total radiative flow in Discrete Ordinate Model (W/m^2)
$q_{in,sol}$	incident solar radiation flow in Discrete Ordinate Model (W/m^2)
$\dot{Q}_{con}$	convective heat loss from absorber front surface (W)
$\dot{Q}_{net,\,i}$	net heat flow over the control volume i boundary (W)
$\dot{Q}_{rad}$	radiative heat loss from absorber front face (W)
$\dot{Q}_{rad,V}$	volumetric radiative thermal heat source in Discrete Ordinate Model (W/m^2)
R_i	area ratio at node section i (–)
s	radiation path position variable in Discrete Ordinate Model (m)
S_v	specific surface area (m^2/m^3)
T	temperature (K)
$T_{env,in/out}$	ambient temperature at absorber inlet, resp. outlet side (K)
T_{si}	solid temperature at node i (K)
t	time (s)
t_s	strut thickness (m)
U_i	total internal energy of control volume i (J)
u_i	specific internal energy of control volume i (J/kg)
V_i	total volume of the control volume i (m^3)
v_i	flow velocity within control volume i (m/s)
v_s	superficial flow velocity – empty tube velocity (m/s)
v	interstitial velocity (m/s)
z	coordinate in direction of air flow, absorber depth (m)
Δp	pressure drop (Pa)
ΔL	flow length (m)
ε_o	open porosity (–)
ε	thermal emittance (–)
$\Phi(e_i,e_j)$	scattering function between incoming (direction vector e_i) and outgoing ray (direction vector e_j) (–)
Ψ	empirical parameter (–)
σ	Stefan-Boltzmann constant (W/(m^2 K^4))
ρ_i	density of fluid within control volume i (kg/m^3)
μ_f	dynamic viscosity of fluid (kg/(m s))
μ_i	polar angle cosine for discretized ray direction with index i in Discrete Ordinate Model (–)
	weighting factor for discretized ray direction with index i in Discrete Ordinate Model (–)
κ_α	absorption coefficient (m^{-1})
κ_s	scattering coefficient (m^{-1})

photovoltaic power), solar thermal power plants can provide dispatchable power by means of thermal energy storage and/or hybridization with renewable or conventional fuels. Given the abundant amount of solar power available for terrestrial solar collectors (85 PW) [1], which exceeds the current world's power demand (15 TW) several thousand times [1], CSP is a highly promising alternative to conventional fossil-fuel or nuclear technology, setting new standards in terms of environmental impact, sustainability, safety, and thus quality of life.

The majority of today's CSP plants are based on the parabolic trough collector technology that has been established on commercial level since the 1980s (SEGS plants in California, USA [2]). Back then, annual solar-to-electric conversion efficiencies achieved values up to 10.6% [2] and are nowadays still not higher than 14–15% [3]. Achieved peak solar-to-electric conversion efficiencies are in the range of 20–25% [4]. Fundamentally, this efficiency constraint is due to the limited operating temperature (≈400 °C) of the applied heat transfer fluid (thermal oil) [5].

Clearly, the move to other heat transfer fluids that enable higher operating temperatures is a must. Viable options are for example molten salts (upper limit at about 600 °C [6]) and air. Another option is the direct heating of the power cycle's working fluid in the solar receiver (e.g. direct steam generation). However, also a solar collector/receiver technology change is inevitable, since higher receiver operating temperatures are only feasible with high area concentration ratios [7]. Instead of the conventional parabolic trough technology (line focusing), the power tower concept (point focusing) is, in this context, much more favorable.

The solar receiver or solar absorber is a key component of a CSP plant and must be optimized to keep thermal losses as low as possible, in order to maximize its thermal efficiency. This work will focus on volumetric absorbers (see Ref. [8] for a detailed review of volumetric receivers) for central receiver or power tower plants. In particular, it will focus on the optimization of ceramic foam absorbers working with air as heat transfer fluid at atmospheric pressure.

In the following, a detailed literature review will be given on the topic of ceramic foam absorbers, as well as on the application of multilayer absorber configurations with gradual property variations in direction of air flow, a likely enhancement method of the absorber's thermal performance.

Ceramic foam was already proposed as promising solar absorber material in the early 1990s by Chavez & Chaza [9] (alumina foam) and later on again by Fend et al. [10,11] (silicon carbide foam). However, although ceramic foam material has advantages regarding heat transfer, solar flux penetration, as well as radial heat transport, this volumetric absorber technology has not been applied in recent non-pressurized solar receiver demonstration projects, where ceramic honeycombs were