



A unified heat transfer model in a pressurized volumetric solar receivers



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ABSTRACT

In the present work, we developed an overall mathematical model adequately describing the main heat transfer processes in a pressurized volumetric receiver. The key components, a windowed cavity, incorporating with the irradiated surface of the absorber, were theoretically modeled as a closed diffuse-gray surfaces system. Accordingly, a boundary condition for the absorber concerning its porous structure surface was developed using net radiation method (NRM) under local thermal non-equilibrium (LTNE) condition. The same method is also applied to the back cavity. Then a modified P1 approximation with collimated irradiation was introduced to incorporate the radiation transfer penetrating in the absorber. The major characteristic of the heat transfer behavior combining radiation, thermal conduction, and convection in the windowed cavity, absorber and the back cavity, are detailedly presented. Also, the key design parameters, such as those relating to pore structure (ϕ and d_p), the volumetric heat transfer coefficient h_v , the emissivity ϵ for window and absorber, and their thickness L_a and L_g were systematically analyzed. Optimization design can be carried out for both of the solar thermal system and the receiver itself in the future work based on our model.

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1. Introduction

A central receiver system (CRS) is characterized by its high temperature and efficiency in the concentrated solar plant (CSP) system. Compared with the parabolic trough system [1] or open-loop CRS [2], the main advantage of the closed-loop system is the significant potential for cost reduction when integrated with combined cycles. High-temperature solar power, fed into the Brayton cycle of a combined cycle plant, can be converted into electricity with efficiencies of up to 30% [3].

Kribus [4] points out the inherent limitation of the volumetric solar receiver using a temperature-dependent model, and suggests a pressurized receiver to achieve a uniformity of the flow and temperature along the absorber. Several types of receivers have been tested during the implementation of R&D projects. System configuration, component efficiencies, and operation experiences are reported in detail [5–7]. The potential of introducing solar

energy into the gas turbine of Combined Cycle systems (CC) and also the biomass pyrolysis/gasification technology [8,9] have been proposed, the overall optimization can be achieved by coupling the thermodynamic equilibrium model of each part of the system. The absorber is the core medium where the concentrated radiative energy is transferred to the thermal energy of the air. The material of the absorber, which is made of metal (e.g., knitted wires, layered grids) or ceramic (e.g., reticulated foams, matrix structures), usually has a highly porous structure. Heat transfer analyses for these types of porous media are carried out. Xu et al. numerically investigates the distribution of the fluid and solid temperatures, based on the local thermal non-equilibrium (LTNE) model. The local thermal equilibrium (LTE) model also has been applied by Bai [10] in his thermal performance analysis. The temperature difference between the solid and fluid phases is very large, especially in the front of the receiver. Wu et al. [11] presented a 2D model, based on the P-1 method, in which the radiation and convection were coupled. A similar model also was applied to an analysis of the thermal performance of the solar dish system [12]. Wang et al. [2] developed a detailed dimensionless LTNE model using a Rosseland approximation for an air receiver. We then analyzed the transportation of

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Nomenclature

c_p	Specific heat of fluid at constant pressure [$\text{J kg}^{-1} \text{K}^{-1}$]
d_p	Pore diameter [m]
E_b	Blackbody emissive power
F	Inertial coefficient
G	Incident radiation
h_v	Volumetric heat transfer coefficient [$\text{W m}^{-2} \text{K}$]
J	Radiosity
K	Permeability [m^2]
L_a	Thickness of a absorber [m]
$\dot{m}$	Mass flow rate [kg/s/m^2]
P	Pressure [Pa]
Pr	Prandtl number
q_0	Initial heat flux [W/m^{-2}]
q	Heat flux
$\hat{s}$	Unit vector in the direction of fluid flow
T	Temperature [K]
u	Velocity[m/s]
$\mathbf{V}$	Velocity vector [m s^{-1}]

Greek symbols:

α_{sf}	Specific surface area of the porous medium [m^{-1}]
ε	Emissivity
ϕ	Porosity
λ	Thermal conductivity [$\text{W m}^{-1} \text{K}^{-1}$]
μ	Dynamic viscosity [$\text{kg m}^{-1} \text{s}^{-1}$]

β	Extinction coefficient [m^{-1}]
σ	Stefan-Boltzmann constant
σ_s	Scattering coefficient
θ	Dimensionless temperature
ζ	Ratio of solid to fluid thermal conductivities
ρ	Density/transmissivity
τ	Optic thickness
ω	Single scattering albedo
Ψ	Dimensionless heat flux

Subscripts

a	Average/absorber
b	Back
c	Collimated/cavity
d	Diffuse
e	Effective/environment
f	Fluid phase
g	Glass
i	Inner/inlet
l	heat loss
o	outer
op	Optic
r	Radiative
s	Solid phase
t	Total
v	Void
w	Wall

radiative energy under collimated irradiation perpendicular to a packed bed [13], by which the essential characteristic of the volumetric effect in this process is revealed. Roldán et al. [14] numerically studied the heat transfer characteristic of an cavity solar furnace, the results of which were in appreciable agreement with the experimental data. Capuano et al. [15] gave an overview regarding the numerical approaches for the characterization of thermo-fluid dynamic behavior. Some researchers proposed a multi-cavity type of receiver [16,17] that consisted of layers of a porous medium due to its compactness and durable to low mass, but their studies were all based on an open receiver system. Other studies, which focused on the optical performances and the heat-flux profiles using Monte Carlo ray tracing (MCRT) methods, also have been reported.

Little attention has been devoted to a pressurized volumetric receiver, and most of the present works have focused on the absorber itself or the test of a prototype. An overall heat transfer process coupled with the absorber medium has not been reported. Hence, in the present work, we propose a uniform model covered the main heat transfer behavior in the flow path of the heat transfer fluid (HTF), air. Boundary conditions taking into consideration the radiation between the window and the absorber, based on the latter's porous structure, are derived. The detailed energy transport in the porous medium, based on the LTNE model under collimated irradiation, is analyzed. The effect of inherent parameters of the porous medium, such as the porosity ϕ , the pore diameter d_p , and the heat transfer parameters, such as the ratio of solid to fluid thermal conductivities ζ and the interphase heat transfer coefficient h_v , on the energy transport (incorporating thermal conduction, convection, and radiative heat transfer) will be examined. The key design parameters, such as thickness and emissivity for the absorber and the window, will be analyzed systematically with respect to their effects on the heat transfer to and from the

absorber.

2. Model

A fundamental configuration composed of a porous slab paralleling with the window in a cavity is considered as can be seen in Fig. 1. The area of the absorber and window is assumed to be sufficient to their thickness, so that a one-dimensional approximation in the direction of the incoming irradiation can be invoked. Besides, we also assume that the solid matrix is homogeneous and isotropic.

2.1. LTNE model [13]

Continuum equation:

$$\frac{d\langle \rho_f u \rangle}{dx} = 0 \quad (1)$$

Momentum equation:

$$\frac{\rho_f \langle u \rangle}{\phi} \frac{d\langle u \rangle}{dx} = \frac{\mu_f}{\phi} \frac{d^2 \langle u \rangle}{dx^2} - \frac{d\langle P \rangle^f}{dx} - \frac{\mu_f}{K} \langle u \rangle - \frac{\rho_f F \phi}{\sqrt{K}} \langle u \rangle^2 \quad (2)$$

where the permeability K , empirical function F which depends primarily on the microstructure of the porous medium can be represented as follows:

$$K = \frac{\phi^3 d_p^2}{150(1 - \phi)^2} \quad (3)$$

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