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Conjugate heat transfer modeling and asymmetric characteristic analysis of the heat collecting element for a parabolic trough collector

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ABSTRACT

We analyzed in this study the conjugate heat transfer process in the heat collecting element (HCE) for a parabolic trough concentrator. The effect of asymmetric characteristics, such as non-uniform heat flux, eccentric configuration, and incident angle, are especially considered in this paper. We constructed a two-dimensional model of surface radiative transfer coupled with convection using the finite volume method combined with the net radiation method. We presented a detailed numerical method in which we applied an additional source term method to the boundary conditions to capture the absorption and emission processes in the coating of the HCE. The results were validated against the experimental data. Thus, we were able to identify the limiting interactions between thermal radiation and convection in the annulus. We systematically analyzed the effect of the incident irradiation angle and the eccentricity of the configuration on heat loss through the annulus of the HCE under two limiting conditions: "air in annulus" and "vacuum in annulus". The results show that asymmetric characteristics, especially the non-uniform heat flux and eccentricity configuration, have a significant influence on local heat transfer processes. These insights may be used to further optimize the configuration of the HCE for improved thermal performance.

1. Introduction

The decrease in the thermal efficiency of solar parabolic trough collectors (PTCs) is the main reason for the efficiency decline of concentrated solar power plants. The getter can extend the lifetime of the vacuum in the annulus, but it cannot ensure that all the collector tubes work under that condition within their service lifetime under an existing manufacturing technique. Currently, the length of the unit row of the heat collecting element (HCE) array in the solar field is over 100 m, and the pipes are connected by welding. In consideration of the operation and maintenance costs, the pipes cannot be replaced immediately but continue working under the "air in annulus" status. Therefore, the following questions need to be answered: Is the HCE required to be replaced immediately after the vacuum becomes invalid in the annulus? What is the metric number for the replacement of the invalid HCE in one row? How do the heat transfer characteristics of HCE change after the vacuum becomes invalid? To answer these questions, an in-depth study should be conducted on the conjugate heat transfer characteristics of an annulus under the "air in annulus" or "vacuum in annulus."

Many studies have been conducted on natural convection in a horizontal annular cavity. Experimental research using a Mach–Zehnder interferometer was conducted by Kuehn and Goldstein [1–3]. Their results visually showed the temperature distributions and presented the local heat transfer coefficients for a concentric or eccentric annulus. They also investigated the effect of *Ra*, *Pr*, and diameter ratio using the numerical simulation with finite difference method. Kumar et al. [4] presented a similar numerical study on natural convection in a horizontal annuli. The boundary conditions were a constant heat flux for the inner wall and a constant temperature for the outer wall. In the studies of Kolesnikov and Bubnovich [5] and Lacroix and Joyeux [6], different boundary conditions were invoked, and external natural convection and internal heat conduction were coupled. Vafai and Ettefagh [7–9] examined the instability of natural convection in the end opening of the annular cavity and analyzed the effects of the axial fluid

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transportation on the radial circulation. The natural convection behavior in an eccentric annular cavity was numerically analyzed by Guj and Stella [10].

In some industrial applications, such as solar thermal power systems, large current bus bars, nuclear reactors, etc., thermal radiation is coupled with natural convection. The effect of thermal radiation cannot be ignored when it dominates the heat transfer process. Balaji et al. [11,12] considered the effect of surface radiation in a closed/open square cavity on natural convection. An annular cavity structure that has an inner cylinder heated by a heat source was numerically analyzed by Shaija and Narasimham [13]. They presented the correlations of the Nusselt number for both radiation and natural convection. Lari et al. [14] used finite volume method (FVM) combined with DOM to consider the effect of radiation.

Studies on the heat transfer process in the annulus of an absorber have also been conducted. Ratzel et al. [15] measured the effective thermal conductivity of a vacuum layer of the HCE under different pressures of the residual gas and experimentally determined the relationship curves of heat loss and the vacuum degree. Dudley et al. [16] tested the LS-2 PTC of a SEGS power plant. They considered three types of working conditions for the collector tube: a) "vacuum in annulus" (pressure less than 10^{-2} Pa), b) "air in annulus" (pressure equal to the atmospheric pressure), and c) the absorber exposed in an environment without coverage of the glass envelope. The effects of *DNI*, flow rate, and temperature of the heat transfer fluid were tested. Kassem et al. [17] gave a sinusoidal approximation of the heat flux boundary to consider the uneven heat flux and simulated the natural convection in the annulus of the HCE. Parametric analysis showed that the eccentric structure could improve the thermal efficiency. Kim and Choi [18] analyzed the effect of the thickness of a cavity on heat loss and considered the conjunct heat transfer with the HCE. In view of the vacuum invalid state in the annulus, Al-Ansary and Zeitoun [19] proposed a partly filled cavity structure to reduce heat loss caused by natural convection.

Most of the previous works on the process of convection coupled with radiation in the annulus only consider the first and second kind of boundary conditions. In the models of these works, constant wall temperature, heat flux, and internal heat source are usually applied. Research on the heat transfer process under multiple boundary conditions for the convection coupled with the radiation heat transfer cannot be found. Most of the studies on its application in the HCE have focused on the acquisition of macroscopic quantities (efficiency and total heat loss), and not much attention has been devoted to its internal heat transfer mechanism. The conjugate heat transfer coupled with thermal radiation in the annulus under an asymmetric configuration and boundary conditions has not been considered thus far.

Therefore, the current study presents a detailed two-dimensional conjugate heat transfer model of the HCE in its radial section. The effect of the incident angle, eccentricity, and ratio of the radius to the thermal radiation coupled with the convection in the annulus is determined. With a non-uniform circumferential heat flux, two typical working conditions are analyzed: "air in annulus" (natural convection in the annulus with pressure equal to one atmospheric pressure) and "vacuum in annulus" (free molecular flow in the annulus with pressure of less than 1 Pa).

2. Physical model and assumptions

For convenience, the subscript number for each of the surfaces is sequenced from 2 to 5, the 1 and 6 refer to heat transfer fluid (HTFs) and ambient, respectively (see the heat balance model in Appendix B). The inner and outer diameters of the absorber are $D_{t,i}$ and $D_{t,o}$, respectively, and the inner and outer diameters of the glass tube are $D_{g,i}$ and $D_{g,o}$, respectively (Table 1). The concentrated heat flux (per unit aperture length) $Q_{3,incid}$ is imposed onto the outer wall of the absorber; the conductive heat flux of the absorber wall is $Q_{32,cond}$; the convection heat flux between the HTF and the absorber is $Q_{21,conv}$; the heat exchange between the inner and outer walls of the annulus by convection and radiation are $Q_{34,conv}$ and $Q_{34,rad}$, respectively; the convective and radiative heat loss between the glass envelope and the ambient are represented by $Q_{56,conv}$ and $Q_{56,rad}$, respectively.

The heat transfer process in the radial section is analyzed on the basis of the following assumptions: (1) the glass tube is treated as transparent for the incident irradiation but opaque for the inner thermal radiation [20]; (2) the absorbing and emitting of the coating is considered a surface phenomenon; and (3) the inner and outer surfaces of the annulus in the model are diffuse-gray surfaces.

3. Two-dimensional model analysis

To further investigate the heat transfer process under a non-uniform heat flux, we proposed a two-dimensional model, in which the coupled process for convection–radiation was analyzed. We chose the PTCs of the DIrect Solar Steam (DISS) project [21] as the physical model, as shown in Fig. 1 [22]. The detailed parameters are presented in Table 1. Note that the numerical analysis was conducted on the basis of the polar coordinate system for the concentric annulus. For the eccentric annulus in the rectangular coordinate system, the detailed numerical treatment can be found in Appendix A.

3.1. Governing equations

The governing equations include the conduction, convection, and radiative transfer between the inner and the outer surfaces of the annulus. Continuity, momentum, and energy are expressed as Eqs. 1–35 in the r-θ coordinate system [2]:

Conservation of mass:

$$\frac{1}{r}\frac{\partial (ru_r)}{\partial r}+\frac{1}{r}\frac{\partial u_\theta}{\partial \theta}=0 \tag{1}$$

Momentum equation:

$$u_r\frac{\partial u_r}{\partial r}+\frac{u_\theta}{r}\frac{\partial u_r}{\partial \theta}-\frac{u_\theta^2}{r}=g_r\beta\left(T-T_{ref}\right)-\frac{1}{\rho_f}\frac{\partial p}{\partial r}+\nu\left(\frac{\partial^2 u_r}{\partial r^2}+\frac{1}{r}\frac{\partial u_r}{\partial r}+\frac{1}{r^2}\frac{\partial^2 u_r}{\partial \theta^2}-\frac{u_r}{r^2}-\frac{2}{r^2}\frac{\partial u_\theta}{\partial \theta}\right) \tag{2}$$

Table 1
Parameters of LS-2 and LS-3 PTC.

	Parameters	LS-2	LS-3
Operation	p/MPa	–	10
	T_{in}/K	–	600
	Q_m/kg/s	–	0.8
Parabolic trough reflector	Wa'/L'/m	5/2	5.76/2
	f'/m	1.40	1.71
	θ'/°	70	80
Absorber tube	$D_{t,i}/D_{t,o}$/m	0.066/0.07	0.054/0.07
	$D_{g,i}/D_{g,o}$/m	0.109/0.115	0.109/0.115
	ρ'	0.94	0.94
	α'	0.94	0.96
	ε'	0.24	0.15
	τ'	0.95	0.96
Glass envelope [20]	ε''	0.86	0.86
	λ_g/W/mK	1.04	1.04