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Numerical and experimental study of a solar micro concentrating collector

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Abstract

Natural convection in fluid-filled enclosures driven solely by a temperature difference represents an important phenomenon due to its numerous engineering applications. Applications span diverse fields such as passive solar heating, solar collectors and the energy efficient design of buildings. In this paper, we study convection inside a rooftop concentrating collector designed to operate at temperatures up to 200 °C. The absorber is contained in a sealed enclosure to minimise convective losses. The main heat losses are due to natural convection inside the enclosure and radiation heat transfer from the absorber tube. A numerical and experimental analysis of the combined laminar natural convection and surface radiation heat transfer inside the collector receiver cavity are presented. A computational fluid dynamics model for the prototype collector has been developed using ANSYS-CFX. Radiation and convection heat loss has been investigated as a function of absorber temperatures, ranging from 70 °C to 200 °C. Measurements of overall heat loss and particle imaging velocimetry (PIV) were used to experimentally determine the heat and mass transport within the enclosure. Excellent qualitative and quantitative agreement between the CFD and experiments were achieved.

Keywords: Cavity receiver; Computational fluid dynamics (CFD); Heat loss; Particle image velocimetry (PIV); Solar energy

1. Introduction

Internal natural convection is a major source of heat loss in enclosed sealed solar thermal collectors (Duffie and Beckmann, 2006; Dey, 2004; Reynolds et al., 2004). A number of heat transfer studies (Balaji and Venkateshan, 1994; Ramesh and Venkateshan, 1999; Behnia et al., 1990) have been conducted in rectangular cavities with combined natural convection and radiation. However, there is limited literature available on the study of natural convection and radiation in a trapezoidal cavity enclosure, which is often used in Fresnel collectors.

Reynolds et al. used flow visualisation to determine convective heat flow patterns within a trapezoidal cavity to investigate the heat losses from a large-scale linear Fresnel solar absorber experimentally (Reynolds et al., 2004). The cavity was also modelled using computational fluid dynamics (CFD) with, only reasonable agreement found between computational and experimental heat transfer rate. Heat loss predicted by CFD model (623 W/m^2) was about 40% lower than that from the experimental results (1040 W/m^2) and the discrepancies could not be explained. Pye et al. studied a compact linear Fresnel reflector (CLFR) cavity receiver and developed computational fluid dynam-

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ics models for the heat loss, which are a combination of convective, radiative and conductive losses (Pye et al., 2003). They also developed an analytical model for a trapezoidal cavity and found that radiation makes up approximately 90% of the heat loss from a 330 °C sealed absorber. They also carried out CFD simulations for the cavity, simplifying the tubes by a plane surface. The results were presented through a correlation for natural convection (Nusselt versus Grashoff, based on the cavity depth dimension) and a correlation for radiation (view factor). These correlations assume that radiation convection interaction effects are negligible. Radiation modelling of the cavity showed that the effects of absorber tube geometry could not be neglected, leading to a radiative heat loss 25% higher than predicted by the cavity model with a plane absorber surface (Pye et al., 2003).

Facao and Oliveira studied heat loss in a trapezoidal cavity of a linear Fresnel receiver using computational fluid dynamics. In their study, natural convection and radiation were examined using two geometrical parameters: receiver depth and insulation thickness. They found a cavity 45 mm deep represents the lowest global heat loss coefficient and 35 mm of rock wool insulation provides a good compromise between insulation and shading (Facão and Oliveira, 2011).

Singh et al. developed overall heat loss coefficient correlations for a trapezoidal cavity absorber by comparing pipe geometry (rectangular vs. round) and pipe coating (ordinary black coating vs. selective surfaces) at various absorber temperatures (up to 175 °C). Selective surface coating on the absorbers reduced the overall heat loss coefficient by 20–30% relative to ordinary black paint (Singh and Sarviya, 2010).

The present research, combines many aspects of outlined literature for the concentrating collector and investigates the role of radiation and natural convection heat transfer inside a trapezoidal cavity that protects the mirrors and absorber tube in a rooftop linear Fresnel collector. The objective of this work is to determine, through numerical and experimental means, the buoyancy-driven airflow generated in the collector cavity and the heat loss taking into account radiation in order to determine methods to increase collector efficiency. A CFD model representing these phenomena is developed and the results obtained for the air flow are compared with experimental results using particle imaging velocimetry (PIV).

2. Micro concentrating collector (MCT) system

The geometry that we are focusing on in the work is a rooftop mounted system from Chromasun called an MCT which is shown in Fig. 1. The system module is 3.2 meters long by 1.2 meters wide and 0.3 m high. The MCT collector utilises linear Fresnel reflector optics that concentrate beam radiation to a stationary receiver. Each mirror array system consists of ten mirrors, which concentrate sunlight onto parallel absorber tubes. The mirrors are curved to generate a focus, with the inner six mirrors and outer four mirrors having a radius of 0.6 and 0.75 meters respectively. The receiver consists of two 16 mm diameter stainless steel absorber tubes. Each receiver has a secondary reflector that directs missed beam radiation to the absorber tube. The whole system is contained in a sealed glass envelop to minimise convective losses. The main heat losses are due to natural convection inside the enclosure and radiation heat transfer from the absorber tube. The design of the receiver is illustrated in Fig. 2. During operation, the hot absorber tube emits long-wavelength radiation into the cavity that is absorbed mainly by the glass and base surfaces which in turn heat up. These heated surfaces, along with the plume from the hot absorber tube, promote buoyancy-driven flows within the cavity resulting in convection losses to the environment and an associated reduction in thermal efficiency (Sultana et al., 2012).

The generalised thermal analysis of a concentrating collector is similar to that of a flat plate collector. For a concentrating collector at steady state, the useful energy output is given by Duffie and Beckmann (2006):

$$q_u = G\eta_o A_a - A_r U_L (T_r - T_a) \tag{1}$$

where G is the solar irradiation (W/m^2), η_o is the collector optical efficiency, A_a is aperture area (m^2), A_r is the receiver area (m^2), U_L is the solar collector heat transfer loss coefficient related to the absorber area (W/m^2 K), T_r is the temperature of the receiver (°C) and T_a is the ambient temperature (°C).

3. Laminar steady state flow

A laminar steady state buoyancy model was developed for the MCT collector. In purely natural convection problems, the Rayleigh Number (Ra) indicates the relative strength of the buoyancy induced flow, and is given by:

$$Ra = Gr\,Pr \tag{2}$$

where Gr is the Grashof number and Pr is the fluid Prandtl number.

The laminar flow regime is generally characterised by $Ra < 10^8$, while turbulent buoyant flow is characterised by $Ra > 10^{10}$ (Cengel and Ghajar, 2010; Zhou and Yang, 2009). The Rayleigh number for MCT receiver was calculated using Eq. (4);

$$Ra = \frac{g\beta(T_{abs} - T_{cover})L^3}{\vartheta^2} Pr \tag{3}$$

where T_{abs} is the absorber tube temperature, T_{cover} is the glass cover temperature and L is the characteristic length. The $Ra = 1.8 \times 10^4$ when the characteristic length, $L = 16$ mm used the diameter of the absorber tube and $Ra = 2.7 \times 10^7$ when $L = 300$ mm used the height of the MCT collector. Therefore, the air flow in the MCT enclosure was modelled as laminar flow. Using Eq. (4) the Rayleigh number was found to be in the range of 1.8×10^4–2.7×10^7 for air with absorber temperatures in the range from 50 to 200 °C.