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ScienceDirect

Solar Energy 85 (2011) 126–138

SOLAR ENERGY

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Thermal performance simulation of a solar cavity receiver under windy conditions

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Received 10 June 2010; received in revised form 16 October 2010; accepted 19 October 2010
Available online 12 November 2010

Communicated by: Associate Editor L. Vant-Hull

Abstract

Solar cavity receiver plays a dominant role in the light–heat conversion. Its performance can directly affect the efficiency of the whole power generation system. A combined calculation method for evaluating the thermal performance of the solar cavity receiver is raised in this paper. This method couples the Monte-Carlo method, the correlations of the flow boiling heat transfer, and the calculation of air flow field. And this method can ultimately figure out the surface heat flux inside the cavity, the wall temperature of the boiling tubes, and the heat loss of the solar receiver with an iterative solution. With this method, the thermal performance of a solar cavity receiver, a saturated steam receiver, is simulated under different wind environments. The highest wall temperature of the boiling tubes is about 150 °C higher than the water saturation temperature. And it appears in the upper middle parts of the absorbing panels. Changing the wind angle or velocity can obviously affect the air velocity inside the receiver. The air velocity reaches the maximum value when the wind comes from the side of the receiver (flow angle $\alpha = 90°$). The heat loss of the solar cavity receiver also reaches a maximum for the side-on wind.

Keywords: Solar cavity receiver; Monte-Carlo method; Flow boiling heat transfer; Heat loss

1. Introduction

The technology of tower-type solar power, which is one of the three primary solar power technologies, had been concerning and studying more and more all over the world, as it has many obvious advantages including clean energy source, large-scaled power generation, and low average cost. One possible configuration is to utilize a solar cavity receiver, which transforms light into heat in the tower-type solar power system. Its performance directly relates to the efficiency of the whole power generation system. So far, most of the studies on the thermal performance of tower-type solar cavity receiver are still focused on the heat loss of the receiver. Clausing (1981) presented an analytical model for the estimation of convective heat loss of a large cubical cavity receiver. Subsequently, the model was refined by including the aperture area and validated with experimental data (Clausing, 1983). Behnia et al. (1990) have studied the combined radiation and natural convection in a rectangular cavity filled with a non-participating fluid. They found that external convection weakens the internal circulation, while radiation strengthens the flow. Balaji and Venkateshan (1993) have numerically investigated the natural convection combined with surface radiation for a rectangular enclosure. A general method was employed in their paper since different emissivities had been considered for the side walls and the top and bottom walls. But the evaluation of view factors is highly tedious. Ramesh and Venkateshan (1999) have led an experimental study of heat transfer by natural convection and surface

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doi:10.1016/j.solener.2010.10.013

Nomenclature

a, b, m	constants in Eq. (25)
Bo	boiling number $= q/(\dot{m} i_{lg})$
C_p	specific heat, J/kg K
D	hydraulic diameter, m
f	friction factor
G	mass flow, kg/m^2 s
h	heat transfer coefficient, W/m^2 K
i	enthalpy, J/kg
k	thermal conductivity, W/m K
N	number (of the light rays)
Nu	Nusselt number
Pr	Prandtl number
Q	heat rate, W
q	heat flux, W/m^2
R	random number
Re	Reynolds number
RD	radiative heat transfer factor
S	surface unit, m^2
T	temperature, K
V	volume unit, m^3
x	equilibrium quality; x coordinate, m
y	y coordinate, m
z	z coordinate, m
ΔT_{sat}	wall superheat $= T_w - T_{sat}$, K
ΔT_{sub}	liquid subcooling $= T_{sat} - T_l$, K

Greek symbols

α	absorptivity
ε	emissivity
θ	angle, degree; zenith angle, degree
κ	absorption coefficient
λ	heat conductivity, W/m K
μ	viscosity, N s/m^2
υ	specific volume, m^3/kg
σ	Stefan–Boltzmann constant, W/m^2 K^4; surface tension, N/m
φ	circle angle, degree

Subscripts

A–E	corresponding to A–E in Fig. 2
f	fluid
g	vapor
i	incidence
l	liquid
lo	all flow as liquid
lg	latent
r	reflection
sat	saturated state
sub	subcooling state
TP	two-phase (flow)
w	wall

Abbreviations

CBD	convective boiling dominant
FDB	fully developed boiling
NBD	nucleate boiling dominant
ONB	onset of nucleate boiling
PB	partial boiling
SB	saturated boiling

radiation in an air-filled cubical enclosure using a differential interferometer. They demonstrated that natural convection is suppressed in the presence of surface radiation. Taumoefolau et al. (2004) conducted some experimental studies on the natural convection heat loss of cavity receiver, using electric heating as the heat source, and found the relationship between the natural convection heat loss and the inclination of the cavity. Reynolds et al. (2004) used the commercial software *FLUENT* to calculate the heat loss of a trapezoidal cavity absorber, comparing it to the experimental data, and found some heat loss characteristics. Paitoonsurikarn and Lovegrove (2006) put forward the natural convection Nusselt number correlation based on the numerical simulation results of three different cavity geometries. Sendhil Kumar and Reddy (2007) developed a 2-D natural convection heat loss model for the modified cavity receiver. It was estimated that the least natural convection heat loss presents at 90° of receiver inclination. Muftuoglu and Bilgen (2008) studied heat transfer in inclined rectangular cavities and found that the Nusselt number is an increasing function of the Rayleigh number, the cavity aspect ratio, and the cavity inclination angle. Based on the computed data, a correlation was derived in the form of $\mathrm{Nu} = f(\mathrm{Ra}, A, \varphi)$. A 3-D model was presented by Reddy and Sendhil Kumar (2009) to investigate the natural convection heat loss from an actual geometry of the modified cavity receiver without insulation at the bottom of the aperture plane. Wu et al. (2010) gave a detail review on the research investigations and activities for cavity receiver convection heat loss. However, these studies were based on the assumption that the wall temperature of the boiling tubes or the surface heat flux of the cavity receiver was given. But the surface heat flux is a key performance parameter of the cavity receiver, which should be calculated according to the sunlight condition in the aperture of the receiver. The solar cavity receiver usually has only one aperture. The concentrated sunlight from the heliostat field passes across the aperture and projects on the surfaces inside the cavity receiver. Besides, the solar cavity receiver is usually designed in an irregular shape, as shown in