International Journal of Heat and Mass Transfer 107 (2017) 510–519

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

# Optical thickness effect on natural convection in a vertical channel containing a gray gas

I. Zavala-Guillén [a,*], J. Xamán [a], C. Salinas [b], K.A.R. Ismail [b], I. Hernández-Pérez [c], I. Hernández-López [d]

[a] Centro Nacional de Investigación y Desarrollo Tecnológico CENIDET-TecNM-SEP, Prol. Av. Palmira S/N. Col. Palmira, Cuernavaca, Morelos CP 62490, Mexico
[b] Faculty of Mechanical Engineering, University of Campinas-UNICAMP, CP 6066, 13083-970 Campinas, SP, Brazil
[c] Universidad Juárez Autónoma de Tabasco, División Académica de Ingeniería y Arquitectura (DAIA-UJAT), Carretera Cunduacán-Jalpa de Méndez km. 1, Cunduacán, Tabasco CP 86690, Mexico
[d] Cátedras Conacyt, Universidad de Quintana Roo, CONACYT-UQRoo, Boulevard Bahía S/N, Col. Del Bosque, Chetumal, Quintana Roo CP 77019, Mexico

ARTICLE INFO

*Article history:*
Received 20 June 2016
Received in revised form 17 October 2016
Accepted 21 November 2016

*Keywords:*
Natural convection
DOM
Participating media
Vertical channel

ABSTRACT

The effect of radiation on natural convection heat transfer in a vertical parallel-plate channel with asymmetric heating, considering the radiation effects for both walls and participating air is presented. The channel is formed by one vertical wall heated by a uniform heat flux and by a vertical adiabatic plate. The governing equations of laminar natural convection and radiative transfer are solved by the finite volume method (FVM) and by the discrete ordinates method (DOM), respectively. The code was validated and verified with data reported in the literature. The effect of optical thickness ($\tau$), channel width ($b$) and wall emissivity ($\varepsilon_h$) on the heat transfer and mass flow are investigated. The mass flow of the channel for $\tau = 0.1$ is up to 42% greater than that obtained for a transparent medium ($\tau = 0.0$). When $\tau = 0.1$, the average temperature difference between the air at the inlet and air at the outlet of the channel decreases up to 75% due to the increase of $b$ from 0.02 to 0.10 m. Varying $\varepsilon_h$ from 0.1 to 0.9 increases the radiative heat flux at the heated wall up to 72% and the mass flow rate increases up to 29%. A set of correlations were obtained for the mass flow, average convective Nusselt number and average radiative Nusselt number.



## 1. Introduction

The combined natural convection and radiation in a vertical channel is representative of several applications such as solar chimneys, solar panels, double skin facades or Trombe wall [1]. The natural convection flow is induced between vertical parallel-plates due to the buoyancy generated when the fluid is heated. Thus, it occurs when at least one of the two walls is heated. The resulting regime flow can be laminar or turbulent depending on the channel geometry, the fluid properties and the temperature of boundary conditions. Natural convection between vertical parallel-plates symmetrically heated and non-symmetrically heated has been extensively studied in the last years, both experimentally and numerically [2–9]. In most of these applications represented by a vertical parallel-plate channel the radiative heat transfer is significant, nevertheless in numerical studies this mechanism is often neglected. The problem of combined radiation and natural convection has been studied for closed cavities with surface radiation, and with both surface and gas radiation [10–14]. However, there are few studies that consider natural convection combined with surface and gas radiation in a vertical parallel-plate channel. One of the first experimental and analytical studies was reported by Yamada [15]. He reported the combined heat transfer of convection and radiation in a vertical channel asymmetrically heated with an absorbing and emitter medium. The author considered the natural convection with laminar flow regime in two dimensions, and he used an exponential wide-band model and gray gas model in one dimension. He concluded that even at intervals of low temperature between ambient temperature and 150 °C, the surface radiation is important in the studied system. He also concluded that an increase of the emissivity of the surfaces from 0.18 to 0.82 increases the heat transfer of the heated surface by 30%. Webb and Hill [16] reported the effect of surface radiation on the heat transfer in a vertical parallel-plate channel, one wall heated with uniform heat flux and the other thermally insulated. Local temperatures along both walls were measured for a range of Rayleigh number regime within $503 \leqslant Ra \leqslant 1.75 \times 10^7$. The temperatures were used to determine

* Corresponding author.
*E-mail addresses:* zavala@cenidet.edu.mx (I. Zavala-Guillén), jxaman@cenidet.edu.mx (J. Xamán), csalinas@unicamp.fem.br (C. Salinas), kamal@unicamp.fem.br (K.A.R. Ismail), ivan.hernandezp@ujat.mx (I. Hernández-Pérez), irving.hernandez@uqroo.edu.mx (I. Hernández-López).

http://dx.doi.org/10.1016/j.ijheatmasstransfer.2016.11.068
0017-9310/

**Nomenclature**

| | |
|---|---|
| $\dot{m}$ | mass flow rate, kg/m·s |
| $b$ | channel width, m |
| $C_p$ | specific heat, J/kg·K |
| $L$ | channel height, m |
| $q$ | heat flux, W/m$^2$ |
| $T$ | temperature, °C |
| $u, v$ | components of velocity, m/s |
| $w_k$ | weight of angular quadrature |
| $x, y$ | coordinates, m |
| *Greek* | |
| $\kappa$ | absorption coefficient m$^{-1}$ |
| $\lambda$ | thermal conductivity, W/m·K |
| $\mu$ | dynamic viscosity, kg/m·s |
| $\rho$ | density, kg/m$^3$ |
| $\sigma$ | Stefan–Boltzmann constant ($5.67 \times 10^{-8}$ W/m$^2$·K$^4$) |
| $\sigma_s$ | scattering coefficient m$^{-1}$ |
| $\tau$ | optical thickness, $\tau = \kappa b$ |
| $\varepsilon$ | emissivity |
| $\varepsilon_h$ | heated wall emissivity |
| $\xi_m, \mu_m$ | direction cosines |
| *Subscripts* | |
| $\infty$ | ambient |
| $c$ | convective |
| $g$ | glazing |
| $r$ | radiative |

the local radiative heat flux, by solving the radiation exchange for the channel with gray-diffuse walls considering an emissivity equal to 0.1. Manca et al. [17] studied experimentally the thermal radiation and natural convection in vertical parallel-plate channels. A correlation for Nusselt number in terms of Rayleigh number ($Ra$) was proposed for $Ra$ up to $10^6$. The measurements showed that the effect of surface radiation is more important for asymmetric heating than for symmetric heating. Experimental and semi-experimental research of laminar natural convection and surface radiation between three parallel vertical plates, a central, highly emissive ($\varepsilon = 0.85$) hot plate and two unheated polished plates ($\varepsilon = 0.05$) was investigated by Krishnan et al. [18]. The radiative heat transfer rate at the hot surface was computed by the radiosity-irradiation method and it was calculated from the power input to the heater. The convective heat transfer rates were obtained from the measured temperatures. Krishnan et al. [18] concluded that the radiation heat transfer rate is significant even at temperatures below 37 °C. Li et al. [19] also studied numerically the influence of surface radiation on the laminar air flow induced by natural convection in vertical asymmetrically-heated channels. They observed that the effect of surface radiation delayed the onset of recirculation at the top part of the channel due to the increased temperature of the adiabatic wall, even for a wall emissivity as small as 0.1.

The studies available in the literature show that radiation heat transfer has a significant effect in the system of vertical parallel-plates with asymmetric heating; hence the radiative effects cannot be neglected. However, most of the research only takes into account the radiative exchange between the walls of the channel (transparent medium), solved by the radiosity-irradiation method. Therefore, the aim of this work is to report the effect of radiation (optical thickness) on natural convection heat transfer in a vertical parallel-plate channel with asymmetric heating, considering the radiation effects for both walls and participating medium. In addition, the effect of emissivity of the heated wall and the channel width on the heat transfer and mass flow rate is analyzed.

## 2. Physical and mathematical model

The geometry of the vertical channel is shown in Fig. 1. The channel is formed by one vertical wall heated by a uniform heat flux, and by a vertical adiabatic plate. The height of the channel is $L = 1$ m and the width of the channel $b$ ranges from 0.02 to 0.10 m. The uniform heat flux at the wall is equal to 250 W/m$^2$. The physical properties of the fluid are assumed constant and they were evaluated at ambient temperature ($T_\infty = 24$ °C), with exception of the density, which was

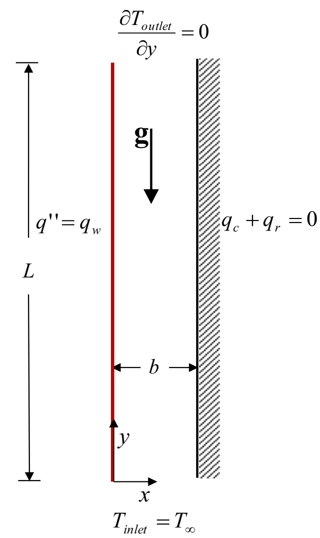


**Fig. 1.** Physical model of the vertical channel with asymmetric heating.

considered with the Boussinesq approximation. The fluid inside the channel is air ($Pr = 0.71$) and it is considered in steady laminar regime. The radiation heat transfer is assumed to be two-dimensional, the participating medium is considered as a gray gas with an optical thickness of $\tau = 0.1$. The vertical walls are considered as gray-diffusely reflecting surfaces, the emissivity of the adiabatic wall is 0.9 and the emissivity of the heated wall has different values ranging from 0.1 to 0.9. The inlet and outlet of the channel are assumed to be black body surfaces.

### 2.1. Mathematical model

The governing conservation equations for a two-dimensional incompressible flow in steady state are:

$$\frac{\partial \rho u}{\partial x} + \frac{\partial \rho v}{\partial y} = 0 \quad (1)$$

$$\frac{\partial(\rho uu)}{\partial x} + \frac{\partial(\rho uv)}{\partial y} = -\frac{\partial P}{\partial x} + \frac{\partial}{\partial x}\left[\mu \frac{\partial u}{\partial x}\right] + \frac{\partial}{\partial y}\left[\mu \frac{\partial u}{\partial y}\right] \quad (2)$$

$$\frac{\partial(\rho vu)}{\partial x} + \frac{\partial(\rho vv)}{\partial y} = -\frac{\partial P}{\partial y} + \frac{\partial}{\partial x}\left[\mu \frac{\partial v}{\partial x}\right] + \frac{\partial}{\partial y}\left[\mu \frac{\partial v}{\partial y}\right] - \rho g \beta (T - T_\infty) \quad (3)$$