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# Radiative and variable thermophysical properties effects on turbulent convective flows in cavities with thermal passive configuration

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ABSTRACT

The influence of the radiative heat transfer and the effects of air variable properties on the natural convection flows established in square cavities with thermal passive device geometry are numerically investigated. Two-dimensional, laminar, transitional and turbulent simulations are obtained, considering both uniform heat flux and uniform wall temperature heating conditions. The average Nusselt number and the dimensionless mass-flow rate have been obtained for a wide range of the Rayleigh number varying from $10^3$ to $10^{16}$. The results obtained for different heating intensities are analyzed and compared. In addition, the influence of considering surface radiative effects on the differences reached for the Nusselt number and the mass flow rate obtained with several heating intensities is studied. The obtained results show that the effects of thermal radiation on the appearance of the *burnout* phenomenon are particularly relevant under given circumstances. The influence of the wall-to-wall spacing of the vertical channel inside the cavity is also analyzed. The changes produced in the flow patterns into the cavity when the radiative heat transfer and the effects of variation of properties are relevant, are also shown.


## 1. Introduction

### *1.1. Some topics in convective flows in cavities and thermal passive devices*

Buoyancy-driven airflows in enclosures and cavities can be found in different applications in Engineering, such as electronic cooling devices, nuclear energy cooling systems, drying of agricultural products, fire and smoke spread in rooms and atriums, and especially in thermal passive devices in buildings (*Trombe walls*, *thermosyphons*, *solar chimneys*). Different aspects of the problem were considered by several authors (Ostrach [1], Chan and Tien [2], Bejan [3], Manca et al. [4]). Recent works including numerical studies focused on square cavities with different morphologies, are those conducted by Bilgen and Oztop [5], Bilgen and Balkaya [6], and Muftuoglu and Bilgen [7], among others.

Regarding the *thermal (solar) passive systems*, they constitute the basic elements of *bioclimatic design*; the absence of mechanical or electrical devices is their main feature. The Trombe wall is the primary example of the *indirect gain technique*, whose typical configuration is usually formed by a thick, darkened, masonry wall along with a glazed wall. In *ventilation applications*, passive systems such as thermosyphons, heat shyphons or solar chimneys, can produce natural airflows due to the induced temperature differences through solar heating. Usually, the thermosyphoning is caused by a glazed solar chimney, which can be also considered as a particular configuration of a Trombe wall. A generalized approach to the problem is that carried out by Zhang et al. [8]. They treated the *air layer involved envelopes* as a unified research subject. These authors pointed out that the air layer basically functions as an extra insulation layer or as a ventilation channel. Several air layer applications in building envelopes can be cited, such as multi-layer door/windows, air flow windows, double-skin façades, solar chimneys, Trombe walls, and others. A considerable body of research over Trombe walls and solar chimneys has been carried out by several investigators, such as Smolec and Thomas [9], la Pica et al. [10], Warrington and Ameel [11], Gan and Riffat [12], Burek and Habeb [13], or Radhakrishnan et al. [14], among others.

The study of the natural convection flow established within the channels (or cavities or enclosures) formed by the walls can be considered as a main ingredient of the problem. In fact, there are still several areas of study and some topics deserve to be mentioned (Desrayaud and Lauriat [15]). The assessment of adequate boundary conditions for numerical simulation of airflow in cavities and enclosures was studied for laminar regime by Khanafer and Vafai [16], and Anil Lal and Reji [17], among others. Because of the large scale of passive ventilation and heating systems, the

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**Nomenclature**

| | |
|---|---|
| $b$ | width of the vents, m (Fig. 1) |
| $b_c$ | wall-to-wall spacing of the vertical channel, m (Fig. 1c and d) |
| $b_w$ | width of the inside solid wall, m (Fig. 1c and d) |
| $c_p$ | specific heat at constant pressure, J kg$^{-1}$ K$^{-1}$ |
| $D_s$ | radiative *stopping distance*, m |
| $g$ | gravitational acceleration, m s$^{-2}$ |
| $Gr_H$ | Grashof number for heat flux cases, $g\beta qH^4/\nu_\infty^2\kappa_\infty$ |
| $Gr_H$ | Grashof number for isothermal cases, $g\beta(T_w - T_\infty)H^3/\nu_\infty^2$ |
| $H$ | height of the cavity (and the heated wall) (Fig. 1), m |
| $h_y$ | local heat transfer coefficient, $-\kappa(\partial T/\partial n)_w/(T_w - T_\infty)$, W m$^{-2}$ K$^{-1}$ |
| $I$ | turbulence intensity |
| $J$ | radiosity (W m$^{-2}$) |
| $k$ | turbulent kinetic energy, Eq. (17), m$^2$ s$^{-2}$ |
| $k_s$ | coefficient of radiative scattering, m$^{-1}$ |
| $L$ | length of the cavity (Fig. 1), m |
| $l$ | typical length, m |
| $M$ | dimensionless mass flow rate, $m/\rho_\infty\alpha_\infty$ |
| $m$ | mass flow rate, kg s$^{-1}$ m$^{-1}$ (two-dimensional) |
| $n$ | coordinate perpendicular to wall, m |
| $Nu_H$ | average Nusselt number based on $H$, heat flux cases, Eq. (9) |
| $Nu_H$ | average Nusselt number based on $H$, isothermal cases, Eq. (10) |
| $Nu_r$ | radiative average Nusselt number based on $H$, Eq. (11) |
| $Nu_y$ | local Nusselt number, $h_yH/\kappa$ |
| $P$ | average reduced pressure, N m$^{-2}$ |
| $P_T$ | total-average reduced pressure, N m$^{-2}$ |
| $p$ | pressure, N m$^{-2}$ |
| $Pr$ | Prandtl number, $\mu c_p/\kappa$ |
| $q$ | wall heat flux (convective), W m$^{-2}$ |
| $q_r$ | boundary heat flux (radiative), W m$^{-2}$ |
| $R$ | constant of air, $R = 287$ J kg$^{-1}$ K$^{-1}$ |
| $Ra_H$ | Rayleigh number based on $H$, $(Gr_H)(Pr)$ |
| $S_{ij}$ | mean-strain tensor, s$^{-1}$ |
| $T, T'$ | average and turbulent temperatures, respectively, K |
| $-\overline{T'u_j}$ | average turbulent heat flux, K m s$^{-1}$ |
| $U_j, u_j$ | average and turbulent components of velocity, respectively, m s$^{-1}$ |
| $-\overline{u_iu_j}$ | turbulent stress, m$^2$ s$^{-2}$ |
| $u_\tau$ | friction velocity, $u_\tau = (\tau_w/\rho)^{1/2}$, m s$^{-1}$ |
| $x, y$ | cartesian coordinates (Fig. 1), m |
| $y_1$ | distance between the wall and the first grid point, m |
| $y^+$ | $\rho y_1 u_\tau/\mu$ |

*Greek symbols*

| | |
|---|---|
| $\alpha$ | thermal diffusivity, $\kappa/\rho c_p$, m$^2$ s$^{-1}$ |
| $\alpha_r$ | coefficient of radiative absorption, m$^{-1}$ |
| $\beta$ | coefficient of thermal expansion, $1/T_\infty$, K$^{-1}$ |
| $\delta_{ij}$ | Krönecker delta |
| $\delta_T$ | thickness of the thermal boundary layer, m |
| $\epsilon$ | coefficient of surface radiation emissivity |
| $\phi$ | dependent variable |
| $\kappa$ | thermal conductivity, W m$^{-1}$ K$^{-1}$ |
| $\Lambda$ | heating intensity, Eqs. (1) and (5) for UWT and UHF heating |
| $\mu$ | viscosity, kg m$^{-1}$ s$^{-1}$ |
| $\nu$ | kinematic viscosity, $\mu/\rho$, m$^2$ s$^{-1}$ |
| $\theta$ | dimensionless temperature difference, $\theta = (T - T_\infty)/(\Lambda T_\infty)$ |
| $\rho$ | density, kg m$^{-3}$ |
| $\sigma$ | Stefan-Boltzmann constant, $\sigma = 5.6678 \times 10^{-8}$ W m$^{-2}$ - K$^{-1}$ |
| $\tau_w$ | wall shear stress, N m$^{-2}$ |
| $\omega$ | specific dissipation rate of $k$, s$^{-1}$ |

*Subscripts*

| | |
|---|---|
| $B$ | constant properties and Boussinesq approximation |
| $max$ | maximum value |
| $opt$ | optimum value |
| $r$ | radiative |
| $t$ | turbulent |
| $w$ | wall |
| $\infty$ | ambient or reference conditions |

*Superscripts*

| | |
|---|---|
| $-$ | averaged value |

*Abbreviations*

| | |
|---|---|
| RTE | Radiation Thermal Equation |
| UHF | Uniform Heat Flux |
| UWT | Uniform Wall Temperature |

convective flow may be laminar, transitional or even fully turbulent; thus, depending on the considered main application, the treatment should be consistent with this fact. The simulation of turbulent flow has received more limited attention, although some relevant works can be found in literature (Ben Yedder and Bilgen [18], Henkes and Hoogendorn [19], Xamán et al. [20]). Authors carried out a thermal and dynamic optimization of the convective flow in Trombe wall shaped channels (Zamora and Kaiser [21]), as well as the determination of the optimal wall-to-wall spacing in solar chimney shaped channels (Zamora and Kaiser [22]). The influence of the variable thermophysical properties of air in open square cavities was studied in Zamora and Kaiser [23]. Other effects, such as thermal radiation, can be relevant under certain circumstances. In this regard, authors have studied numerically the thermal radiative effects also on open square cavities (Zamora and Kaiser [24]). Now, present study deals with the same two topics aforementioned (the radiative effects, and the influence of the variable properties), but focusing the effort on the flows induced by buoyancy effects in cavities with thermal passive device geometry.

### 1.2. Effects of the variable thermophysical properties

Usually, the buoyancy forces are introduced in the numerical simulations of natural convection flows through the *Boussinesq approximation*. It is well known that this approximation assumes constant the thermophysical properties of the fluid (except the density variations produced by temperature differences in the buoyancy term of the momentum equation), and therefore it can be applied when temperature variations are low enough. In contrast, moderate and intense heating conditions can be found in applications such as passive heat dissipation in electronic systems. This fact can severely modify the properties of the fluid (air, in most cases), and therefore to change the predictions of the heat transfer coefficient and the mass flow rate (Gray and Giorgini [25]). The influence of property variations on convective flows in a square enclosure was analyzed by Zhong et al. [26], and Emery and Lee [27]. Chenoweth and Paolucci [28] explained that the Boussinesq approximation could produce important errors for temperature increases about 20% of the reference temperature