MODELING OF THE THERMAL TRANSFERS IN AN ENCLOSURE OF THE TROMBE WALL TYPE

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The heat transfer by natural convection and thermal radiation between the two compartments of an enclosure filled of air and divided by a vertical partition wall are studied numerically. The air is aspired in the cavity from the bottom, then it is heated and pushed into the ventilation openings at the top. This two-dimensional configuration is a simplified representation of the solar system with ventilated wall Trombe. We show that the radiative exchanges reduce the temperature differences between the surfaces and increase the average Nusselt number as well as the air flow through the openings. The increase in the dimensionless width of the opening promotes the heat transfer within the cavity.

Keywords: solar buildings, natural convection, thermal radiation, partition, average Nusselt number

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Nomenclature

d – thickness of the partition wall, (m)
D – dimensionless thickness of the partition wall ($D = d/L$)
g – gravity acceleration, (m·s^{-2})
k – thermal conductivity, (W·m^{-1}·K^{-1})
L – width of the cavity, (m)
l – distance between the low opening and the bottom adiabatic wall, (m)
l_0 – width of the opening, (m)
L_0 – dimensionless width of the opening, l_0/L
l_p – length of the partition wall, (m)
s – distance between the hot wall and the partition wall, (m)
S – dimensionless distance between the hot wall and the partition wall, $S = s/L$
Nr – radiation number, $\sigma T_c^4/(k_a \Delta T/L)$
Nu – Nusselt number
p – pressure, (Pa)
P – dimensionless pressure, $(p + \rho_o g y)L^2/\rho_o \alpha^2$
Pr – Prandtl number, ν/α
Q – dimensionless air flow crossing the openings
q_r – net radiative flux density, (W·m^{-2})

Qr – dimensionless net radiative flux density, $q_r/\sigma T_h^4$
Ra – Rayleigh number, $g\beta(T_h - T_c)L^3/\nu\alpha$
R_k – thermal conductivity ratio, k_w/k_a
T – temperature, (K)
T_h – temperature of the hot wall
T_c – temperature of the cold wall
T_0 – average temperature, $(T_h + T_c)/2$, (K)
u, v – velocity components, (m·s^{-1})
U, V – dimensionless velocity components $U = uL/\alpha$, $V = vL/\alpha$
x, y – coordinates, (m)
X, Y – dimensionless coordinates ($X = x/L$, $Y = y/L$)

Greek letters

ΔT – maximal temperature difference, $T_h - T_c$, (K)
α – thermal diffusivity of fluid, (m^2·s^{-1})
β – volumetric expansion coefficient, (K^{-1})
λ – dynamic viscosity ratio, μ_w/μ_a
μ – dynamic viscosity, (Kg·m^{-1}·s^{-1})
ν – kinematic viscosity of fluid, (m^2·s^{-1})
ρ_0 – density of the fluid with T_0, (kg·m^{-3})
θ – dimensionless temperature, $(T - T_0)/(T_h - T_c)$
σ – Stefan–Boltzmann constant, (W·m^{-2}·K^{-4})
ε – emissivity of a radiative surface

Indices and exponents

a – air
h – hot
c – cold
w – wall
o – opening

Introduction

Several numerical and experimental studies were carried out on the heat transfer by natural convection and thermal radiation in partitioned rectangular cavities [1-2]. This interest is due to the various industrial applications that these geometries reflect in several problems of engineering. Among this work, one quotes in particular that of Mezrhab et al. [1] who studied the interaction natural convection - radiation in a vertical enclosure blocked by a solid block. They showed that the thermal radiation has a great influence on the pace of the isotherms and streamlines, then increases the average Nusselt number considerably. They have also found that the effect of the thermal conductivity of the solid block is more pronounced in combined mode than in pure natural convection. Mezrhab et al. [2] analyzed the effect of the thermal radiation on the heat transfer and the air flow within an enclosure containing a solid block generating heat. They found that the thermal radiation reduces the maximum temperature in the enclosure, because of radiative fluxes lost by the side bordering solid block. An experimental and numerical study concerning the effect of a partition placed at the medium of a rectangular enclosure was carried out by Nakamura and Asko [3]. They concluded that emissivities of the walls opposite influence considerably the heat transfer by convection.

Among technologies, search and studies carried out in simple enclosure of geometry, several works was interested in the study of the thermal behavior of the solar systems of the type wall Trombe and its alternatives. Tadrari et al. [4] studied numerically the coupled heat transfer by conduction, natural convection and radiation in a vertical rectangular enclosure delimited by a glass and a massive wall absorbing a solar flux. They noted that for ΔT=10K, there is a critique value (Rac) of Ra about 1.7×10^8 for which the flow change direction. They have also shown that more than 75% of the total heat transfer in the enclosure is through radiation. Buzzoni et al. [5] presented a numerical solution to the problem of natural convection in the case of heating buildings by a passive solar system, which is a variant of the system Trombe-Michel. Gan [6] studied the Trombe walls for use in the cooling of buildings in summer conditions. He pointed to a study which showed that the computer code developed by CFD (Computational Fluid Dynamics) can be used for predicting the movement of the air flow and buoyant in the fences with the geometry of Trombe wall. Borgers and Akbari [7] studied numerically turbulent convective flow free in a Trombe wall. They then developed correlations to estimate the performance of Trombe walls. Ong [8] proposed a simple mathematical model of a solar chimney, the physical model is similar to the Trombe wall. Awbi and Gan [9] used a CFD programs to simulate air flow and heat transfer in a Trombe wall.

In our work, the results are obtained by using the real data of the solar systems with passive wall. We study the effect of two openings located meadows of the hot wall, one in bottom and the other in top of the enclosure (Fig. 1) on the heat transfer within an enclosure differentially heated. One holds account of the coupling between thermal conduction through the walls, the exchanges by natural convection in and between the compartments and the exchanges by radiation between radiative surfaces constituting the enclosure.

The principal objective of this work is to analyse the influence of the radiative exchanges on the average Nusselt number and the air flow through the openings.

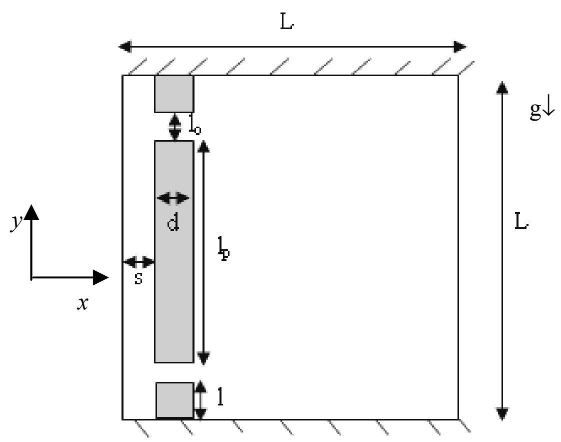

Fig. 1. *Studied configuration*

Mathematical formulation and numerical procedure

We assume that the geometry is two-dimensional, the flow is laminar, the radiative surfaces are diffuse gray and the physical properties of the air, except its density, are constants at the average temperature T_o. Only solid surfaces take part in the radiation exchange. The horizontal walls of the enclosure are adiabatic, while the vertical walls left and right-hand side are respectively hot and cold.

The dimensionless equations controlling the heat transfer and the flow in the enclosure are written in the dimensionless form:

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \qquad (1)$$

$$U\frac{\partial U}{\partial X} + V\frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \lambda Pr\left(\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2}\right) \qquad (2)$$

$$U\frac{\partial V}{\partial X} + V\frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + \lambda Pr\left(\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2}\right) + RaPr\theta \qquad (3)$$

$$U\frac{\partial \theta}{\partial X} + V\frac{\partial \theta}{\partial Y} = R_k\left(\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2}\right) \qquad (4)$$

with: $\lambda = 1$, $R_k = 1$ in the fluid area and $\lambda = \infty$, $R_k = k_w/k_a$ in the solid; k_a, k_w are thermal conductivities of the air and the wall, respectively.

The boundary conditions are: on the walls of the enclosure: $U = V = 0$; for $X = 0$, $0 \le Y \le 1$: $\theta = 0.5$ and for $X = 1$, $0 \le Y \le 1$: $\theta = -0.5$.

Along the partition wall: $R_k\frac{\partial \theta_w}{\partial n} = \frac{\partial \theta_a}{\partial n} - NrQr$; on the adiabatic walls: $0 = \frac{\partial \theta_a}{\partial Y} - NrQr$, n: normal direction on the radiative surface considered.

The equations (1-4) are discretized using the method of finite volumes. The coupling pressure-velocity is treated using the algorithm SIMPLER. The view factors, with screening effects, were determined by using the methods of the boundary element and Monte Carlo [2]. The equations, of the algebraic system, obtained were solved by the method of the conjugate gradients.

After study, we found that the grid 60×60, irregular and fine near the enclosure walls and the faces of the partition wall, gives a good compromise between the precision of the results and the computing time.

When one takes into account the thermal radiation, the average temperature T_0 is fixed at 290 K. In order to respect the validity of the Boussinesq approximation, the maximal temperature difference ΔT is chosen lower or equal to 15 K. The Prandtl number was fixed at $Pr = 0.71$. R_k is fixed to 50 in all calculations, whereas the Rayleigh number Ra and the dimensionless width L_0 of the opening were varied from 10^3 to 10^7 and 0 to 0.2, respectively. The ends high and low of the central partition wall are located at the same distance of adiabatic walls and its height is fixed at $0.6L$. Consequently, the increase in widths of the two openings causes the reduction in widths L of the walls (top and bottom), respectively attached to the adiabatic walls. The distance between the hot wall and the partition wall, and the thickness of the partition wall were fixed respectively to $S = 0.05$ and $D = 0.15$.

Our objective in this work, is to study the effects of the thermal radiation, the Rayleigh number and the opening width on the average Nusselt number and the air flow through the openings.

$$Nu = \int_0^1\left(-\left.\frac{\partial \theta}{\partial X}\right|_{X=0,Y} + NrQr(X = 0, Y)\right)dY$$

and the dimensionless air flow through the openings:

$$Q = -\int_{l/L}^{0.2} U(X = S, Y)dY = \int_{0.8}^{(0.8+L_0)} U(X = S, Y)dY.$$

The comparison between the results obtained in pure natural convection and natural convection combined with the thermal radiation makes it possible to highlight the influence of the thermal radiation.

Results and discussion

Fig. 2 presents the isotherms and streamlines obtained for $Ra = 10^6$, $0 \le L_0 \le 0.2$ and $\varepsilon = 0$. For the comparison, the case of the empty enclosure (without partition) is also presented in this study. It is noted that the fluid circulates in the clockwise direction and is mono-cellular owing to the position of the isothermal walls (hot and cold). Indeed, the fluid goes up along the hot wall and goes down along the cold wall. At the center of the enclosure, one attends a thermal stratification of the temperature under the effect of the buoyancy forces. Also, the isotherms are dense in the vicinity of the hot wall and the vicinity of the left wall of the partition, like those of the cold wall of the right face of the partition wall.

The isotherms and streamlines corresponding to $Ra = 10^6$ and $0 \le L_0 \le 0.2$ are presented in Fig. 3 for $\varepsilon = 1$. We note that the radiative exchanges bring closer the temperatures the hot wall and the left face of the partition wall, like those of the cold wall and the right face of the partition wall. The slope of the isotherms close to the adiabatic walls is due to the importance of radiative fluxes. Along the cold wall, the variations in temperature are clearly less important in the upper part than in the low part of the enclosure. Indeed, the difference between the average temperatures of the air in the left and right parts of the enclosure rises under the effect of the radiation.

The average convective Nusselt number is more important when the radiation exchange is taken into account. Furthermore, we observe also the increase of the air flow through the openings.

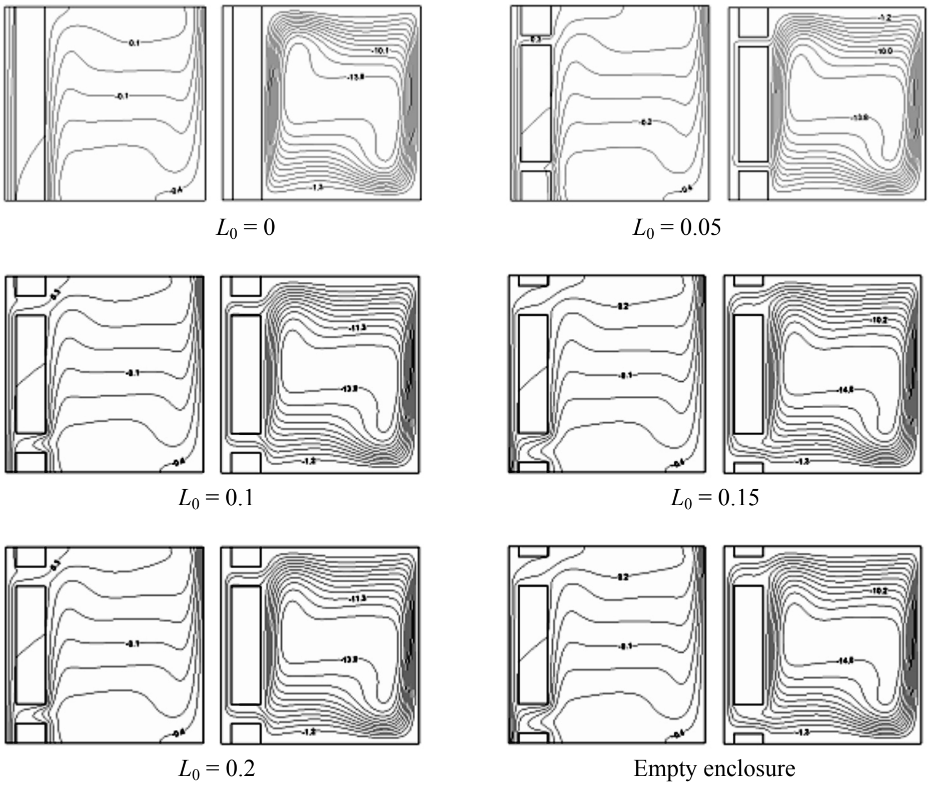

Fig. 2. *Isotherms and streamlines for Ra = 10^6, $\varepsilon = 0$ and $0 \le L_0 \le 0.2$*

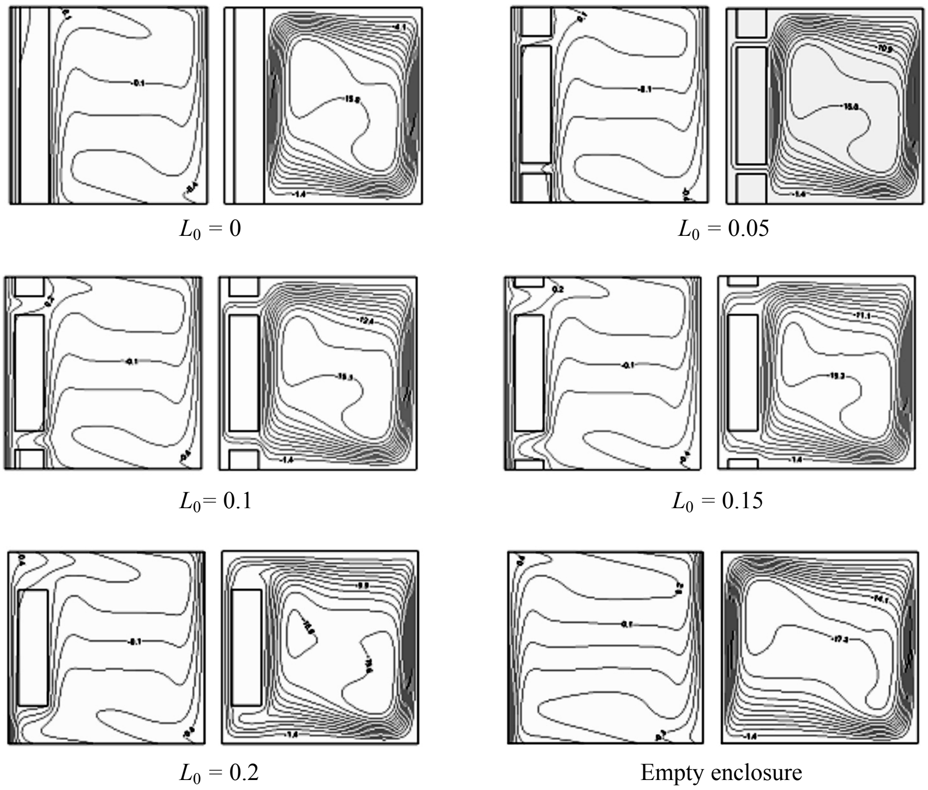

Fig. 3. *Isotherms and streamlines for Ra = 10^6, $\varepsilon = 1$ and $0 \le L_0 \le 0.2$*

Average Nusselt number and air flow crossing the openings

The average Nusselt number and the air flow increase with the opening width, in presence ($\varepsilon = 1$) or in absence ($\varepsilon = 0$) of the radiation, as it is indicated in Fig. 4 and 5, respectively.

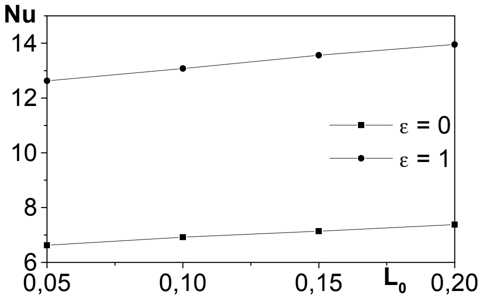

Fig. 4. *Average Nusselt number according to L_0 for $Ra=10^6$*

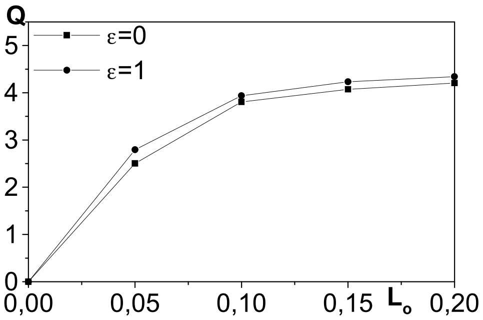

Fig. 5. *Air flow according to L_o for $Ra= 10^6$*

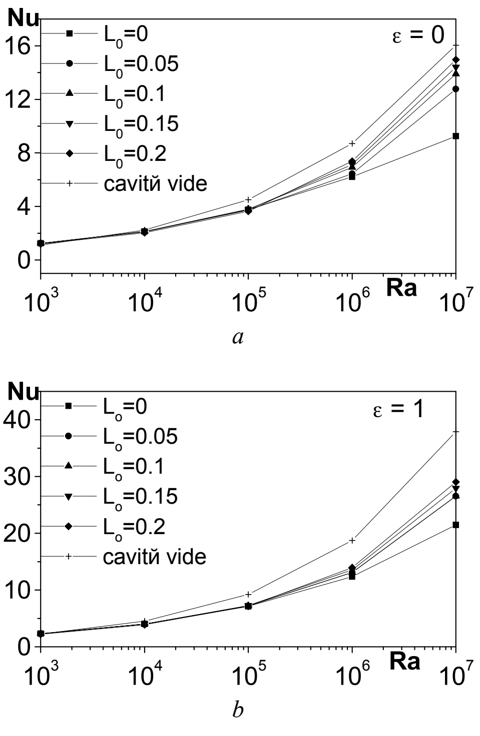

Fig. 6. *Variation of the average Nusselt number according to Ra: (a) $\varepsilon = 0$ and (b) $\varepsilon = 1$*

The influence of the opening on the average Nusselt number is negligible until $Ra = 10^5$ (Fig. 6 and 7) because, the air circulation being weak, the heat transfer is done mainly by conduction and radiation. We note an increase in *Nu* and *Q* with L_0, particularly for large Rayleigh number and in presence of the radiation exchange. The variation of the air flow according to L_0 is presented on figure 5 for $Ra = 10^6$. It is clear that the air flow increases with the increase in the width opening L_0 and also in presence of the thermal radiation. For $Ra \leq 10^5$ (Fig. 7) (mode of conduction and transition), the flow is low, even if it increases under the influence of the radiation.

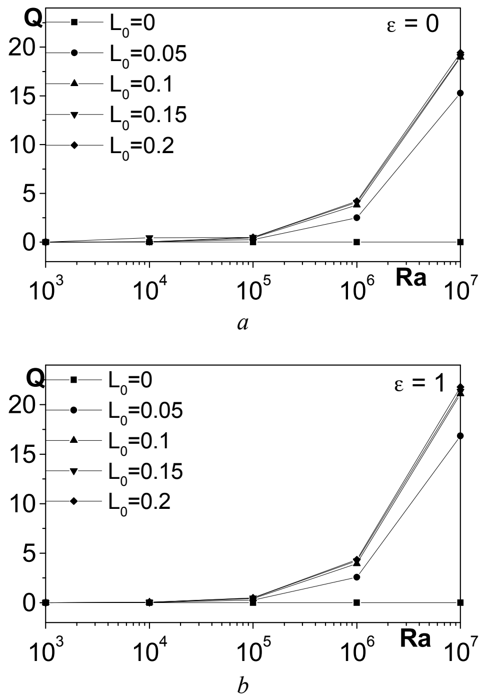

Fig. 7. *Variation of the air flow through the openings according to Ra: (a) $\varepsilon = 0$ and (b) $\varepsilon = 1$*

Conclusion

Numerical simulations carried out on the coupling between the natural convection and the thermal radiation in a partitioned enclosure lead to the following conclusions: (i) the streamlines and the isotherms are considerably affected by the presence of the partition wall, (ii) the thermal radiation standardizes the temperatures in the two parts of the enclosure and increases the difference between their average temperatures, (iii) the air flow through the openings increases under the thermal radiation effect, (iv) the thermal radiation contributes to an increase in the heat transfer, mainly for larger values *Ra*, (v) the heat transfer is minimum for $L_0 = 0$ and is maximum for an empty enclosure.

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