NUMERICAL STUDY OF MIXED CONVECTION COUPLED WITH RADIATION IN A VENTED PARTITIONED ENCLOSURE

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The present work reports numerical results of mixed convection and surface radiation within a horizontal ventilated cavity, with an aspect ratio $A = L'/H' = 2$, heated from below and provided with an adiabatic partition, of a fine thickness, on the heated surface. Air, a radiatively transparent medium, is considered to be the cooling fluid. The effect of the governing parameters, which are the Reynolds number, $200 \le Re \le 5000$, the partition position from the inlet, $0.25 \le L_b \le 1.75$, and the emissivity of the walls, $0 \le \varepsilon \le 0.85$, on the fluid flow and heat transfer characteristics is studied in detail. The relative height of the partition, $H_b = H'_b/H'$, and the relative height of the openings, $B = h'/H'$, are kept constant at 1/2 and 1/4 respectively.

Keywords: mixed convection, surface radiation, ventilated cavity, adiabatic partition

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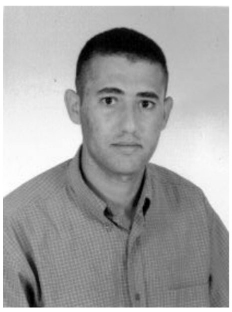

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Nomenclature

A – aspect ratio of the cavity $(= L'/H')$
B – relative height of the openings $(= h'/H')$
cv – convection
F_{ij} – view factor from S_i surface to S_j one
g – acceleration due to gravity, m/s^2
h' – height of the openings, m
H' – height of the cavity, m
H_b – relative height of the partition $(= H'_b / H')$
I_i – dimensionless irradiation $(= I'_i/\sigma T_C'^4)$
J_i – dimensionless radiosity $(= J'_i/\sigma T_C'^4)$
L' – length of the cavity, m
L_b – dimensionless x-direction distance of the partition from the inlet $(= L'_b / H')$
Nr – convection-radiation interaction parameter $(= \sigma T_C'^4 / q')$
Nu – average Nusselt number
Pr – Prandtl number $(= \nu/\alpha)$
q' – imposed wall heat flux, W/m^2
Q_r – dimensionless radiative heat flux $(= Q'_r /\sigma T_C'^4)$
Ra – Rayleigh number $(= g\,\beta\, q' H'^4/\alpha\nu\lambda)$
rd – radiation
Re – Reynolds number $(= u'_0 H'/\nu)$
t – dimensionless time $(= t' u'_0 / H')$
T – dimensionless fluid temperature $(= \lambda(T' - T'_C)/q' H')$
T' – dimensional fluid temperature, K
$\overline{T}$ – dimensionless mean temperature
$T_{\max}$ – dimensionless maximum temperature
T'_C – common temperature of the left vertical cold wall and imposed flow, K
T_0 – dimensionless reference temperature $(= \lambda T'_C / q' H')$
u'_0 – velocity of the imposed flow, m/s
(u, v) – dimensionless horizontal and vertical velocities $(= (u', v')/u'_0)$
(x, y) – dimensionless coordinates $(= (x', y')/H')$

Greek symbols

α – thermal diffusivity of fluid, m^2/s
β – thermal expansion coefficient of fluid, 1/K
Δt – dimensionless time step
ε – walls emissivity
λ – thermal conductivity of fluid, W/(K·m)
ν – kinematic viscosity of fluid, m^2/s
Ω – dimensionless vorticity $(= \Omega' H'/u_0)$
Ψ – dimensionless stream function $(= \Psi'/u'_0 H')$
σ – Stéfan-Boltzman constant (= $5.67\cdot10^{-8}$ W/m^2·K^4)

Subscripts and Superscripts

$_C$ – cold temperature
$_H$ – heated wall
$_{\max}$ – maximum value
$_{\min}$ – minimum value
$'$ – dimensional variable

Introduction

Mixed convection heat transfer in ventilated systems continues to be a fertile area of research, due to the interest of the phenomenon in many technological processes, such as the design of solar collectors, thermal design of buildings, air conditioning and recently the cooling of electronic circuit boards. In the literature, numerous analytical, numerical and experimental studies dealing with mixed convection in ventilated geometries have been reported without radiation effect. The effect of the latter can be neglected in the case of configurations with non emissive or weakly emissive boundaries which is not the case in general since the contribution of radiation to the overall heat transfer could be significant. In the absence of radiation, mixed convection in a square enclosure provided with a partially dividing partition was studied numerically by Hsu et al. [1], How and Hsu [2] and Hsu and Wang [3]. Results of the simulations indicate that the heat transfer and flow structure are strongly dependent on the height, the conductivity ratio and the location of the conducting baffles. Laminar mixed convection in a two-dimensional enclosure with assisting and opposing flows was studied numerically by Raji and Hasnaoui in the case of a cavity uniformly heated from one or two side walls [4-6]. The obtained results show that the *Re-Ra* plane can be divided in regions corresponding to the dominance of the forced convection or to the mixed convection regime where the heat transfer is maximum. Recently, mixed convection from a flush-mounted uniform heat sources in a rectangular enclosure with openings was numerically investigated by Bhoite et al [7]. and Saha et al. [8]. In the case of ventilated cavities, the numerical study, conducted by Raji and Hasnaoui [9] on combined mixed convection and radiation, showed that the contribution of radiation could be important even though the cooling fluid is transparent to radiation. The neglected effect of thermal radiation is mainly justified by the fact that the heat transfer is especially ensured by mixed or forced convection. However, moderate temperature differences give rise to significant radiation effects and the fact of neglecting their contribution becomes non realistic.

The main objective of the present study consists of examining the effect of the Reynolds number, *Re,* the horizontal position of partition, L_b, and the emissivity of the walls, ε, on flow and thermal fields. Variations, versus the main controlling parameters, of maximum and mean temperatures are also explored.

Problem formulation

The configuration under study, together with the system of coordinates is depicted in Fig. 1. It consists of a ventilated rectangular cavity. The bottom wall is uniformly heated with a constant heat flux and provided with a vertical adiabatic baffle. The upper horizontal and right vertical walls are considered insulated, while the left side of the cavity is cooled with a constant temperature. The system is submitted to an imposed flow of ambient air through an opening located on the lower part of left vertical wall. The forced flow leaves the cavity through an outflow opening placed on the higher part of the right vertical wall. The inner surfaces, in contact with the fluid, are assumed to be gray, diffuse emitters and reflectors of radiation with identical emissivities. The fluid properties are evaluated at a mean temperature and the airflow is assumed to be, two-dimensional, laminar, incompressible and obeying the Boussinesq approximation. Under these assumptions, the dimensionless governing equations, written in terms of vorticity and stream function formulation, are as follows:

$$\frac{\partial \Omega}{\partial t}+u\frac{\partial \Omega}{\partial x}+v\frac{\partial \Omega}{\partial y}=\frac{1}{Re}\left[\frac{\partial^2\Omega}{\partial x^2}+\frac{\partial^2\Omega}{\partial y^2}\right]+\frac{Ra}{Re^2\,Pr}\;\frac{\partial T}{\partial x}, \quad (1)$$

$$\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\frac{1}{Re\,Pr}\left[\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right], \quad (2)$$

$$\frac{\partial^2\Psi}{\partial x^2}+\frac{\partial^2\Psi}{\partial y^2}=-\Omega\,. \quad (3)$$

The stream function and the vorticity are related to the velocity components by the following expressions:

$$u=\frac{\partial \Psi}{\partial y}\;,\;\; \mathrm{v}=-\frac{\partial \Psi}{\partial x}\;\text{ and }\;\Omega=\;\frac{\partial v}{\partial x}-\;\frac{\partial u}{\partial y}. \quad (4)$$

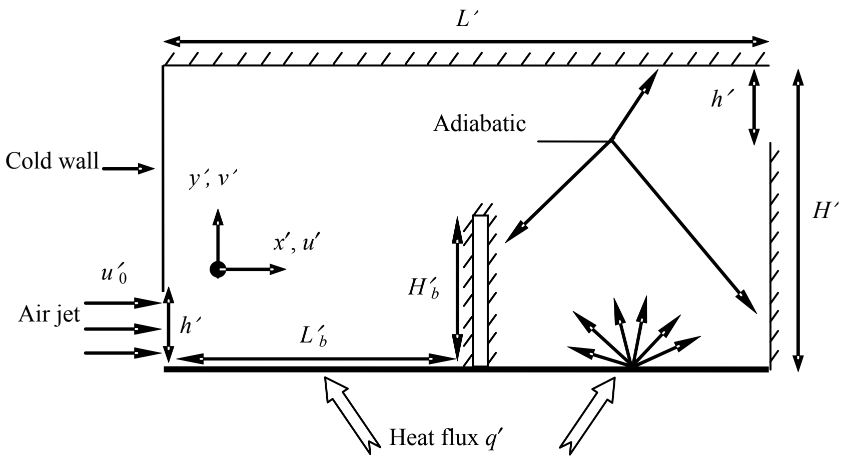


Fig. 1. *Schematic of the studied configuration*

Boundary conditions

The boundary conditions, associated to the problem are as follows: $u = v = 0$ on the rigid walls; $T = v = \Omega = 0$, $u = 1$ and $\Psi = y$ at the inlet of the cavity; $T = 0$ on the left vertical cold wall; $-\frac{\partial T}{\partial \mathrm{y}}+N_rQ_r=1$ on the lower horizontal heated wall; $-\frac{\partial T}{\partial n}+N_rQ_r=0$ on the adiabatic walls; "n" being the normal direction to the considered adiabatic wall.

For this problem, the boundary conditions are unknown at the outflow opening. Values of u, v, T, Ψ and Ω are obtained at each time step by mean of an extrapolation technique [9-11].

Radiation equations

The calculation of the radiative heat exchange between the cavity and its surrounding (through the inlet and the exit) is based on the radiosity method. In addition, the radiative heat transfer between the system surfaces is expressed by the following set of equations in non-dimensional form:

$$J_i=\varepsilon_i\left(\frac{T_i}{T_0}+1\right)^4+(1-\varepsilon_i)\sum_{j=1}^{N}F_{ij}\,J_j \quad (5)$$

The dimensionless net radiative heat flux leaving an element of surface S_i is evaluated by:

$$Q_r=J_i-I_i=\varepsilon_i\left[(\frac{T_i}{T_0}+1)^4-\sum_{j=1}^{N}F_{ij}\mathrm{J_j}\right] \quad (6)$$

Heat transfer

The average Nusselt numbers, characterizing the contributions of mixed convection and thermal radiation through the heated wall, are respectively defined as:

$$Nu_H(cv)=-\frac{1}{A}\int_0^A-\frac{1}{T}\left(\frac{\partial T}{\partial y}\right)\Bigg|_{y=0}dx\;;$$

$$Nu_H(rd)=\frac{1}{A}\int_0^A\frac{1}{T}\left(N_rQ_r\right)\Bigg|_{y=0}dx\,. \quad (7)$$

The total Nusselt number, Nu, is evaluated as being the sum of the corresponding convective and radiative Nusselt numbers, i.e. $Nu = Nu(cv) + Nu(rd)$.

Method of solution

The non linear partial differential governing equations, Eq. (1)-(3), were discretized using a finite difference technique. The first and second derivatives of the diffusive terms were approached by central differences while a second order upwind scheme was used for the convective terms to avoid possible instabilities frequently encountered in mixed convection problems. The integration of equations (1) and (2) was ensured by the Alternating Direction Implicit method (ADI). At each time step, the Poisson equation, Eq. (3), was treated by using the Point Successive Over-Relaxation method (PSOR) with an optimum over-relaxation coefficient equal to 1.88 for the grid (81×41) adopted in the present study. The set of Eq. (5), representing the radiative heat transfer between the different elementary surfaces of the cavity, was solved by using the Gauss-Seidel method. The numerical code was validated against the results of Akiyama and Chong [12] obtained in the case of a square cavity differentially heated. Comparisons, made in terms of convective Nusselt numbers, evaluated at the heated wall, showed a fairly good agreement with relative maximum deviations limited to 1.07 %/(1.36 %) for $\varepsilon = 0$ / (1) for *Ra* varying in the range $10^3 \le Ra \le 10^6$ (Table 1).

Table 1

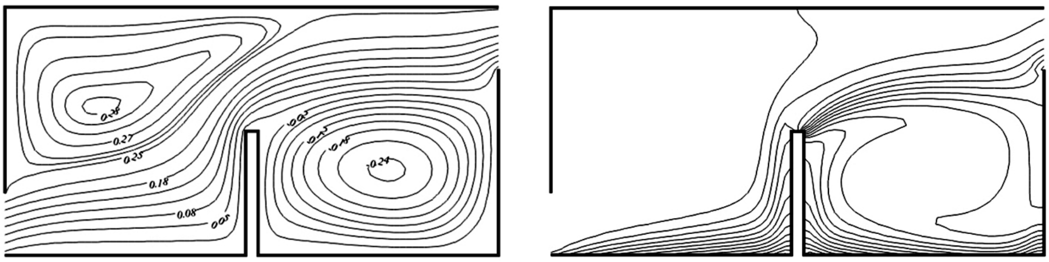
Effect of *Ra* and ε on the mean convective Nusselt number, Nu_{cv}, evaluated on the heating wall of a square cavity for T'_H = 298.5 K and T'_C = 288.5 K

	ε = 0				ε = 1			
Ra	10^3	10^4	10^5	10^6	10^3	10^4	10^5	10^6
Present work	1.118	2.257	4.627	9.475	1.250	2.242	4.192	8.100
Akiyama and Chong [12]	1.125	2.250	4.625	9.375	1.250	2.250	4.250	8.125

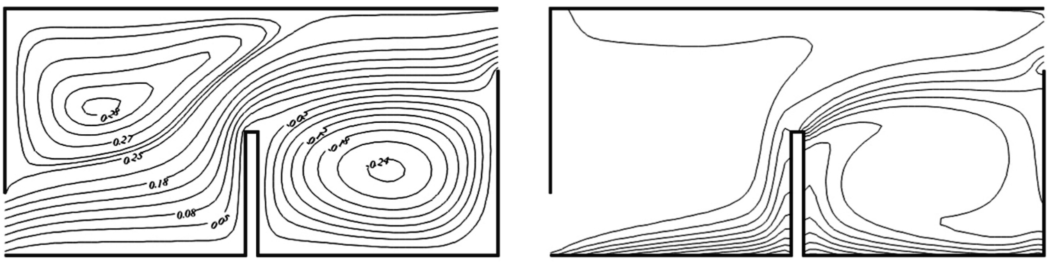
Results and discussion

In the present study, the value of Rayleigh number, *Ra,* is fixed at $5\cdot10^6$.

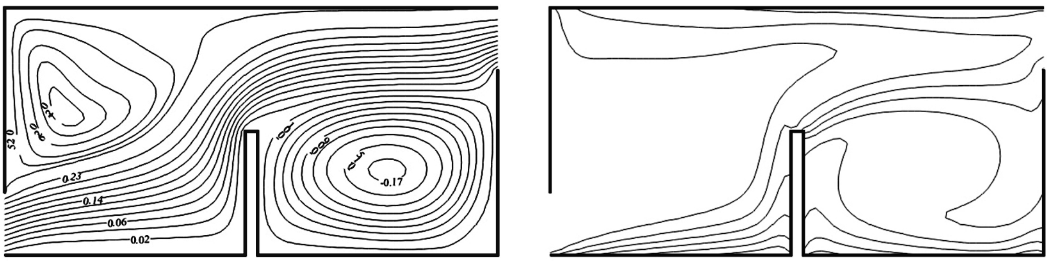
The effect of radiation on the flow structure and temperature distribution inside the cavity is illustrated in Fig. 2 for *Re* = 250 and various values of ε.



a (Ψmin = -0.25, Ψmax = 0.28)



b) (Ψmin = -0.22, Ψmax = 0.27)



c) (Ψ_{min} = -0.17, Ψ_{max} = 0.27)

Fig. 2. *Streamlines and isotherms obtained for Re = 250, L_b = 1 and different values of ε: a) ε = 0, b) ε = 0.15 and c) ε = 0.85*

The analysis of the streamlines in Fig. 2, *a*, obtained for ε = 0, reveals the existence of open lines surmounted by a trigonometric cell whose formation is rather due to the shearing effect and a clockwise natural convection cell (its direction of rotation is imposed by the forced flow) under the open lines, located on the right of the baffle. The corresponding isotherms are tightened at the level of the heating wall indicating a good convective heat exchange between this wall and the open lines/(closed cell) on the left/(right) side of the baffle. An important heat exchange can also be seen between the lower cell and the open lines while this interaction is quasi absent between the forced flow and the upper cell. Consequently, the cold zone occupies a non negligible part of the available space on the left part of the fin (in the entrance region of the cavity), visibly reduced by increasing the emissivity. The reduction of the cold zone space, following the increase of ε, indicates that the wall's radiation plays an important role in the homogenization of the fluid temperature inside the cavity. In addition, the increase of the emissivity reduces the importance of the upper cell (size and intensity) in favor of the open lines (Fig. 2, *b, c*) and leads to an increasing spacing of the isotherms in regions where the thermal interaction is important in the absence of radiation. Moreover, when the inner surfaces are radiatively participating, a heating of the adiabatic upper wall is observed and presents increasingly significant thermal gradients as the emissivity increases.

Variations, versus *Re*, of the average Nusselt numbers, resulting from contributions of convection and radiation and the total Nusselt number, evaluated along the heated wall, are presented in Fig. 3, *a-c* for various values of ε. As expected, Fig. 3, *a* shows a monotonous increase of $Nu_H(cv)$ with *Re* either with or without radiation effect. The rate of this increase becomes more important from *Re* ~ 1000 (an increase in the slope of the curves is observed from this threshold). This tendency is justified by the flow intensification with the inertia effect, promoted by the increase of *Re*. For a fixed value of this parameter, the increase of the emissivity of the walls leads to a noticeable decrease of the convection effect. The negative role of radiation on the natural convection is well known and it is confirmed here in the case of mixed convection. The effect of the emissivity of the walls on the radiative heat transfer component is presented in Fig. 3, *b* in terms of $Nu_H(rd)$ variations with *Re* for ε = 0.15, 0.5 and 0.85. Globally, it can be deduced that, for a given value of *Re*, the effect of radiation is important for ε = 0.5 and 0.85 and it is characterized by an important increase of $Nu_H(rd)$ with ε. Also, the effect of *Re* on $Nu_H(rd)$ is limited in the case of ε = 0.15 but becomes increasingly positive by increasing ε. The variations of the total Nusselt number with *Re*, presented in Fig. 3, *c*, show increasing tendencies of Nu_H with *Re* and ε which means that the positive impact of radiation on the radiative Nusselt number is more important than its negative effect on the convective Nusselt number.

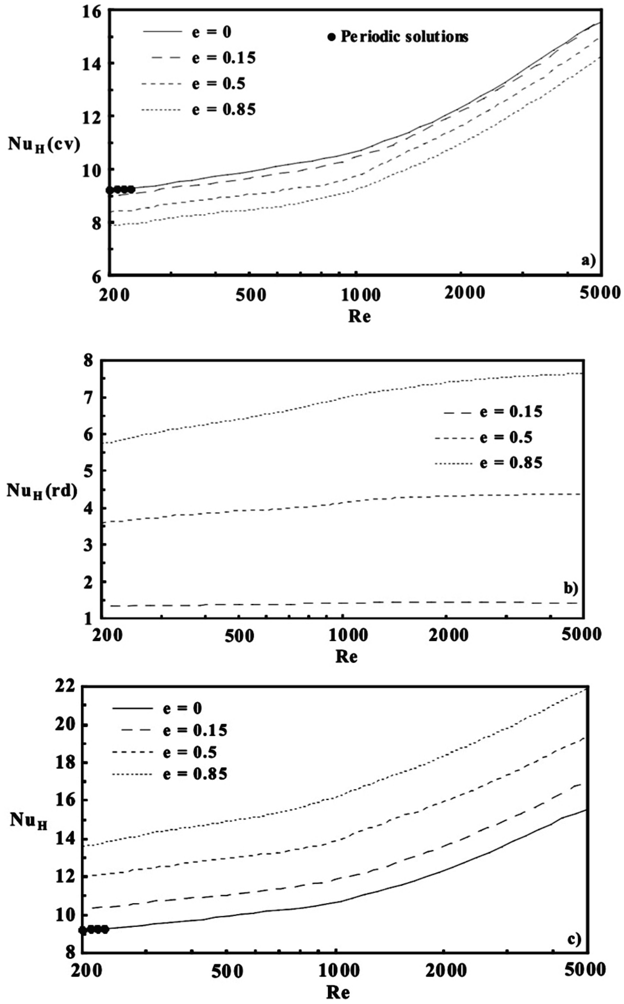
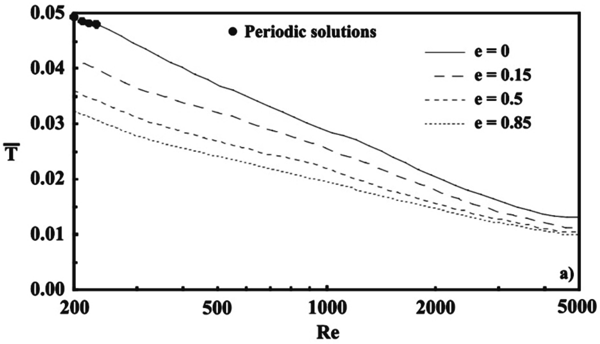


Fig. 3. *Variations, with Re, of the average Nusselt numbers on the heated wall for L_b = 1 and various values of ε : a) $Nu_H(cv)$, b) $Nu_H(rd)$ and c) Nu_H*

For practical applications, it is of great importance to know the impact of the governing parameters on mean and maximum temperatures of the fluid inside the cavity. Thus, the variations of these quantities with *Re* are presented in Figs. 4, *a*, *b* for different values of ε. For all considered values of ε, Fig. 4, *a* shows that the evolution of $\overline{T}$ is characterized by a continuous decrease by increasing *Re*. In addition, for a given *Re*, the increase of ε is accompanied by a decrease of the average temperature since the part of energy provided by the hot wall and leaving directly the cavity through the openings, without being transported by the fluid, increases with ε. The evolution of the maximum temperature, presented in Fig. 4, *b* and generally located on the heated wall, is marked also by a monotonous decrease with *Re* and ε which means that the overheating phenomenon can be avoided by increasing the emissivity of the walls and the velocity of the external imposed flow. Quantitatively, for *Re* = 200, the increase of the walls emissivity from zero to 0.85 generates a decrease in mean and maximum dimensional temperatures by about 7.25 °C and 60.83 °C, respectively but this decrease drops to about 1.37 °C (case of ε = 0) and 35.28 °C (case of ε = 0.85) for Re = 5000.



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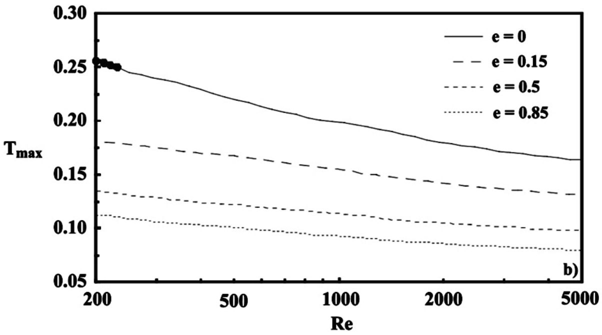
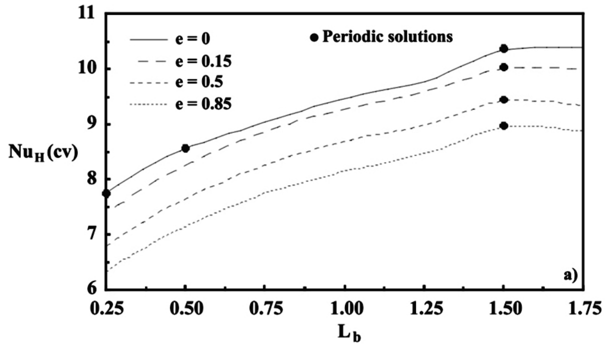


b

Fig. 4. *Variations, with Re, of the temperature for L_b = 1 and various values of ε: a) mean temperature $\overline{T}$, b) maximum temperature T_{max}.*

Variations of the Nusselt numbers, with the position L_b of the baffle, are presented in Fig. 5 for Re = 300 and different values of ε. Fig. 5, *a* shows that the convective component of heat transfer increases by moving away the partition from the cold wall but the rate of this increase becomes limited beyond L_b = 1.5; critical value from which the reduction of the space between the partition and the adiabatic vertical wall leads to a limited interaction between the closed cell and the open lines. Fig. 5, *b* shows that the increase of L_b is characterized by a limited decrease of $Nu_H(rd)$ what means that the latter is almost independent of L_b. However, the figure shows clearly an increasing positive effect of radiation when ε is increased leading to an enhancement of the total heat transfer (Fig. 5, *c*) and the latter is favored also by the increase of L_b. Improvements in terms of the total heat transfer of about 28.2 % and 18.1 % are obtained respectively for ε = 0 and 0.85, when the parameter L_b is varied from 0.25 to 1.75. It should be mentioned that the solution is unsteady for L_b = 1.5. The corresponding results, presented by full circles in Figs. 5, *a-c*, were obtained as averaged values during flow cycles.



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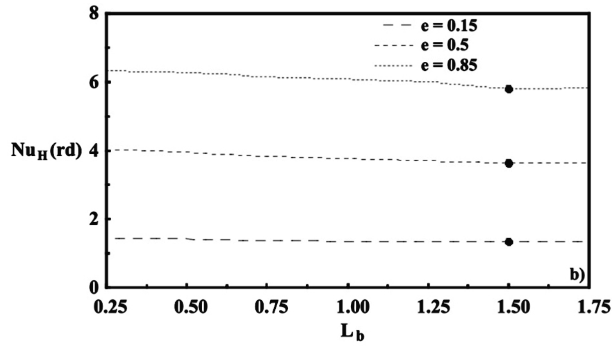

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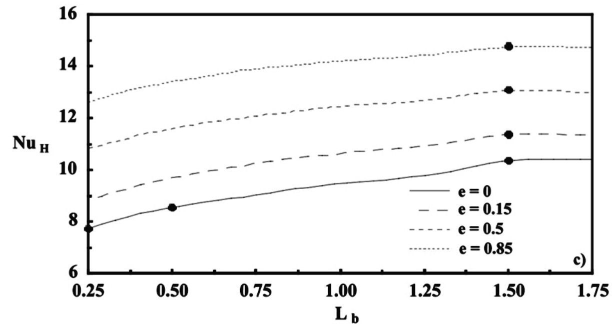
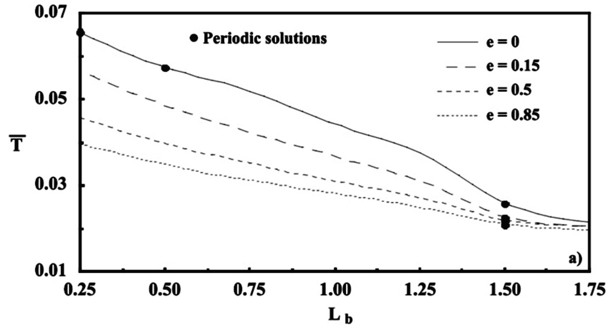


c

Fig. 5. *Variations, with L_b, of the average Nusselt numbers on the heated wall for Re = 300 and various values of ε : a) $Nu_H(cv)$, b) $Nu_H(rd)$ and c) Nu_H*

Variations, with L_b, of mean and maximum temperatures are presented in Figs. 6 for Re = 300 and different values of ε. It is seen from Fig. 6, *a* that $\overline{T}$ decreases by increasing ε and L_b. In fact, the increase of the overall heat transfer with both ε and L_b (Fig. 5, *c*) and the mixed convection heat transfer component with L_b (Fig. 5, *a*), contributes to a better cooling within the cavity. It should be noted that for L_b > 1.5, the effect of the emissivity ε on $\overline{T}$ becomes limited. The variations of the maximum temperature, T_{max}, reported in Fig. 6, *b*, show a decrease of this parameter by increasing ε or L_b with a drastic decrease from L_b = 1.25, due to the competition between natural and forced convections, towards a minimum reached at L_b = 1.5, position for which the maximum of the overall heat transfer is reached.



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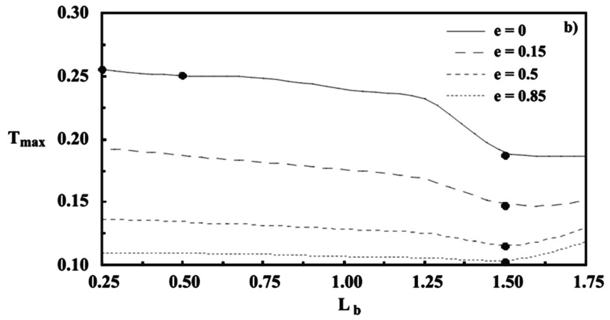

b

Fig. 6. *Variations, with L_b, of the temperature for Re = 300 and various values of ε: a) mean temperature $\overline{T}$, b) maximum temperature T_{max}*

Concluding remarks

The problem of combined mixed convection and radiation inside a partitioned ventilated cavity has been studied numerically. Results of the study show that the radiation effect leads to a better homogenization of the temperature inside the cavity by reducing the cold zone space in the entrance region. It is found that the radiation effect reduces the convective Nusselt number component and the latter is favored by the Reynolds number, *Re*, and the displacement of the partition away from the inlet, L_b. All the parameters *Re*, L_b and ε are found to have a positive effect on the total heat transfer. The better cooling of the cavity, expressed by the decrease of mean and maximum temperatures of the fluid, is obtained by the increase of the parameters *Re*, L_b and ε. Also, the contribution of radiation to the overall heat transfer is generally not negligible even for the weaker value (ε = = 0.15) considered for ε.

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