International Journal of Mechanics and Energy (IJME)
Vol. 8, Issue 1, 2021, ISSN: 2286-5845

Influence of Thermal Radiation on Steady Mixed Convection inside a Multiple Vented Cavity subjected to Sucked or Injected External Flow

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Abstract: A numerical study is carried out to investigate the interaction between mixed convection and thermal radiation in a horizontal ventilated enclosure. Air, a radiatively transparent medium, is admitted into the cavity by injection or suction, by means of two openings placed on the lower part of both right and left vertical sides. Another opening is located on the middle of the top wall to ensure the ventilation. The parameters governing the problem are the Reynolds number, Re, and the emissivity of the walls, ε. The effect of these parameters on flow and thermal fields as well as on the heat transfer rate within the enclosure is examined for the two ventilation modes (injection and suction). Variations, versus the main controlling parameters, of maximum and mean temperatures are also presented. The results indicate, for the two modes of imposed flow, that the thermal radiation affects the flow and thermal structures. Also, it is found that the radiation enhances the global heat transfer so that its contribution is more considerable even for the high values of Re. However, the suction mode is found to be more favorable to the heat transfer in comparison with the injection one.

Key words: Surface radiation, Mixed convection, numerical study, vented cavity, injection, suction

1. Introduction

Many engineering systems during their operation generate heat flux. If this generated heat is not dissipated rapidly to its surrounding atmosphere, this may cause rise in temperature of the system components and leads to system failure. In order to maintain the system at its recommended temperature for its efficient working and avoid serious overheating problems, it is necessary to employ an external ventilation to achieve higher thermal performance and dissipate the generated heat within the system. Since natural or forced convection is not able to provide cooling effectiveness in many thermal processes, the mixed convection cooling mechanism has been recommended in such systems. However, the performance of any cooling strategy depends largely on the method of air distribution and mode of ventilation. Understanding the hydrodynamic and heat transfer processes under mixed convection taking into consideration the impact of ventilation mode is very important for effective design of the cooling system. Therefore, the mixed convection heat transfer in ventilated cavities has received a growing interest during the last decades. An exhaustive review of the literature shows that the problem of mixed convection in vented enclosure was examined in the past by several authors [1-3] in the absence of radiation.

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However, the effect of the latter can be neglected in the case of configurations involving low temperature differences and 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. The interaction between natural convection and surface thermal radiation has been investigated numerically by Alvarado et al. [4] in a tilted slender cavity. The bottom and top surfaces of the cavity were heated and cooled at constant temperatures, while its sidewalls were considered thermally insulated. The steady-state results indicated that surface radiation coupled with natural convection notably modifies the flow patterns and the average heat transfer in the slender cavity. Moreover, the total heat transfer increases by increasing the inclination angle, except when the flow structure changes from the multi-cell to the unit-cell pattern. Also, it is observed that the increase of the aspect ratio of the cavity has a negative impact on the total heat transfer; the latter decreases by increasing the aspect ratio. Colomer et al. [5] analyzed the natural convection phenomenon coupled with radiant exchange in a three dimensional differentially heated cavity. The problem was solved for a transparent medium showing that radiation significantly increases the heat exchange. More importantly, for a given Plank number and a constant reference temperature ratio, the contribution of radiation remains almost impervious in a designated range of Ra. Transient natural convection and surface radiation in a closed cavity with heat conducting solid walls of finite thickness and internal heat source have been studied by Martyushev and Sheremet [6]. It has been found that regardless of the considered solid-fluid interface the average convective Nusselt number increases with the Rayleigh number and thermal conductivity ratio, and decreases with the surface emissivity and ratio of solid wall thickness to cavity spacing. The effects of surface radiation and inclination angle on heat transfer and flow structures in an inclined rectangular cavity with a centred inner body were studied numerically by Bouali et al. [7]. It was revealed that the increase of the inclination angle reduces considerably the total heat transfer in the cavity but the inner body reduces the heat transfer in the cavity especially in the presence of radiation exchange. Interaction between surface radiation and natural convection in an air filled cavity with a heated plate placed at its center has been studied numerically by Saravanan and Sivaraj [8]. It was found that the contribution of the convective mechanism to the overall heat transfer increases with emissivity when the plate is horizontally placed whereas decreases when it is vertically placed.

Conjugate heat transfer by natural convection and radiation provided in the case of side-vented open cavities has been studied experimentally by Ramesh and Merzkirch [9] and numerically by Singh and Venkateshan [10] where opening is localised on the top and one isothermal vertical wall forms the heat source. Typical results showed that, for cavities with low emissive walls, natural convection was found to be the dominant mode and with highly emissive walls, both natural convection and surface radiation were found to have a competing behaviour by contributing equally to the total heat transfer.

Unfortunately, the literature survey reveals that coupled effects of mixed convection and radiation in the case of vented enclosures are not well documented although it is frequently encountered in practice. Such configurations can be used effectively for the cooling purposes in many applications such as electronic equipment cooling, room cooling, air conditioning, etc. In this context, Raji and Hasnaoui [11] studied the interaction between thermal radiation and mixed convection in a ventilated rectangular cavity by considering different configurations depending on the inlet and outlet positions. The obtained results show that thermal radiation alters significantly the temperature distribution, the flow fields and the heat transfer across the active walls of the cavities. Recently, Bahlaoui et al. [12] investigated the combined effect of mixed convection and surface radiation within a horizontal ventilated cavity heated from below and provided with an adiabatic thin partition on the heated surface. The numerical simulations revealed that the radiation effect reduces the convective Nusselt number component and the latter is favored by the Reynolds number, Re, and the baffle position, L_b. 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 the emissivity ε.

Based on the literature review, it is clear that no work has been reported on combined mixed convection and surface radiation in a rectangular enclosure having multiple ports and ventilated by two different modes (injection and suction). Therefore, due to its modern applications, the subject needs further effort to improve knowledge in this field. Hence, the aim of the present study is to examine the effect of the Reynolds number, the emissivity of the walls and the mode of the imposed flow, on flow and temperature fields. Variations versus the main controlling parameters of maximum and mean temperatures are also reported.

2. Problem Definition and Mathematical Modeling

The configuration under study, together with the system of coordinates is depicted in Fig. 1. It consists of a ventilated horizontal rectangular cavity having an aspect ratio A = 2 and uniformly heated from below with a constant heat flux. The upper horizontal and right vertical walls are considered perfectly insulated, while the left side of the cavity is cooled with a uniform temperature. The physical system is submitted to an imposed ambient air stream which is admitted by injection (Fig. 1a) or suction (Fig. 1b) through two openings located on the lower part of both right and left vertical walls. The forced flow leaves or enters the cavity through a third opening localised on the middle of the upper wall. The inner surfaces, in contact with the fluid, are assumed to be gray, diffuse emitters and reflectors of radiation with identical emissivities. The third dimension of the cavity (direction perpendicular to a plane of the diagram) is assumed to be large enough so that the fluid motion can be considered two-dimensional. The flow is conceived to be laminar and incompressible. The fluid properties are constant except the fluid density in the buoyancy term, which obeys the Boussinesq approximation. Therefore, under these assumptions and using the following dimensionless variables:

$$A = L'/H' \ , \ B = h'/H' \ , \ x = x'/H' \ , \ y = y'/H' \ ,$$

$$u = u'/u'_o \ , \ v = v'/u'_o \ , \ t = t'u'_o/H' \ ,$$

$$T = (T' - T'_C)/(T'_H - T'_C) , \ \Psi = \Psi'/u'_o H' , \ \Omega = \Omega' H'/u'_o$$

$$\mathrm{Pr} = \nu/\alpha , \ Ra = \mathrm{g}\,\beta\,\mathrm{q}'\mathrm{H}'^4/\alpha\nu\lambda \quad \mathrm{Re} = u'_o H'/\nu$$

The corresponding dimensionless differential equations that governing the convection can be written, using the vorticity-stream function (Ω-Ψ) formulation, as follows:

$$\frac{\partial \Omega}{\partial t} + u\frac{\partial \Omega}{\partial x} + v\frac{\partial \Omega}{\partial y} = \frac{1}{\mathrm{Re}}\left[\frac{\partial^2 \Omega}{\partial x^2} + \frac{\partial^2 \Omega}{\partial y^2}\right] + \frac{\mathrm{Ra}}{\mathrm{Re}^2\ \mathrm{Pr}}\ \frac{\partial T}{\partial x} \tag{1}$$

$$\frac{\partial T}{\partial t} + u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \frac{1}{\mathrm{Re\ Pr}}\left[\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right] \tag{2}$$

$$\frac{\partial^2 \Psi}{\partial x^2} + \frac{\partial^2 \Psi}{\partial y^2} = -\Omega \tag{3}$$

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

$$u = \frac{\partial \Psi}{\partial y} \ ; \ v = -\frac{\partial \Psi}{\partial x} \quad \text{and} \quad \Omega = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} \tag{4}$$

The radiosity method is used for the calculation of the radiative heat exchange between elementary surfaces of the cavity. The non-dimensional radiosity equation for the ith element of the enclosure may be written as:

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

The dimensionless net radiative heat flux leaving the elementary surface S_i is given by:

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

Where F and T_o are the view factors and the dimensionless reference temperature, respectively

2.1 Boundary Conditions

The common boundary conditions, in the dimensionless form, applied to both ventilation modes can be defined as follows:

$u = v = 0$ on the rigid walls

$T = 0$ on the left vertical cold wall

$\Psi = 0$ on the lower horizontal heated wall

$-\frac{\partial T}{\partial y} + N_r Q_r = 1$ on the lower horizontal heated wall

$-\frac{\partial T}{\partial n} + N_r Q_r = 0$ on the adiabatic walls

″n″ indicates the outward direction normal to the considered adiabatic wall.

The appropriate dimensionless boundary conditions related to the injection or suction cases can be written as:

Injection case:

$T = v = \Omega = 0$, $u = 1$ and $\Psi = y$ at the left inlet of the cavity

$T = v = \Omega = 0$, $u = -1$ and $\Psi = -y$ at the right inlet of the cavity

$\Psi = B$ between the left inlet port and the upper outlet port

$\Psi = -B$ between the right inlet port and the upper outlet port

For this injection mode, the boundary conditions are unknown at the upper outlet opening. Values of u, v, T, Ψ and Ω are extrapolated at each time step by considering zero second derivatives of these variables at the exit of the cavity.

Suction case:

$T = 0$ at the upper inlet port

$u = -1$, $v = 0$, $\Psi = -y$, $\Omega = 0$ at the left outlet port

$u = 1$, $v = 0$, $\Psi = y$, $\Omega = 0$ at the right outlet port

$\Psi = -B$ between the left outlet port and the upper inlet port

$\Psi = B$ between the right outlet port and the upper inlet port

For this suction mode, the boundary conditions for u, v, Ψ and Ω are unknown at the upper inlet opening whereas the temperature T is unknown at both left and right outlet openings. Similarly to the previous case, values of these variables are obtained at each time step by considering zero second derivatives of these variables at these openings.

The absence of physical boundary conditions for vorticity on rigid walls is surmounted by using the approximation proposed by Woods [13]:

$$\Omega_\omega = -\frac{1}{2}\Omega_{\omega+1} - \frac{3}{\Delta\eta^2}\left(\Psi_{\omega+1} - \Psi_\omega\right) \tag{7}$$

In the precedent equation, ω stands for the wall and Δη is the space step in the outward direction normal to the wall.

2.2 Heat Transfer

The mean Nusselt numbers, characterizing the contributions of mixed convection and thermal radiation through the heated wall, are respectively determined by:

$$Nu_H(\mathrm{cv}) = -\frac{1}{\mathrm{A}}\int_0^A \frac{1}{T}\left(\frac{\partial T}{\partial y}\right)\Bigg|_{y=0} dx \ ;$$
$$\mathrm{Nu}_H(\mathrm{rd}) = \frac{1}{\mathrm{A}}\int_0^A \frac{1}{T}(\mathrm{N}_r Q_r)\Big|_{y=0} dx \qquad (8)$$

The overall Nusselt number, Nu, is evaluated as being the sum of the corresponding convective and radiative Nusselt numbers; i.e. $\mathrm{Nu_H} = \mathrm{Nu_H(cv)} + \mathrm{Nu_H(rd)}$.

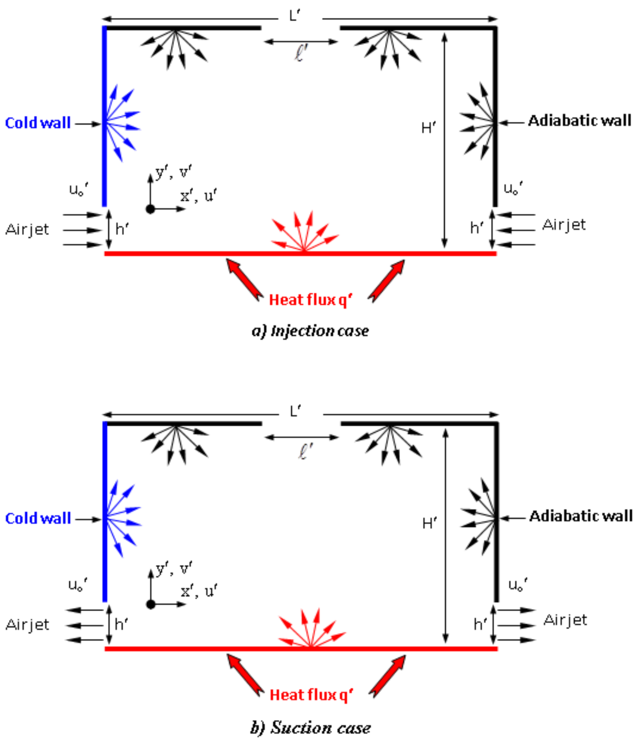


Fig. 1: Schematic diagram of the studied configuration: a) Injection case and b) Suction case.

3. Numerical Approach

The non-linear partial differential governing equations, Eqs. (1)-(3), were discretized using a finite difference method. 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 Eqs. (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 adopted in the present study (201×101). 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.

4. Code validation and grid independence study

The present code was validated against the numerical results obtained by Wang et al. [14] for natural convection coupled with radiation in a square cavity differentially heated. The comparative results are exemplified in Fig. 2 in terms of total mean Nusselt number evaluated on the heated wall for Ra ranging from 10^4 to 10^6 and ε varying from 0 to 0.85. The figure shows that the maximum difference does not exceed 1.62 %. In addition, an overall energy balance for the system was systematically checked for all the computations. Thus, the energy released by the heating wall to the fluid is evaluated and compared with that leaving the cavity through the cold wall and the openings. In our situation, the energy balance is satisfied and the maximum difference remains very weak (lower than 2 %).

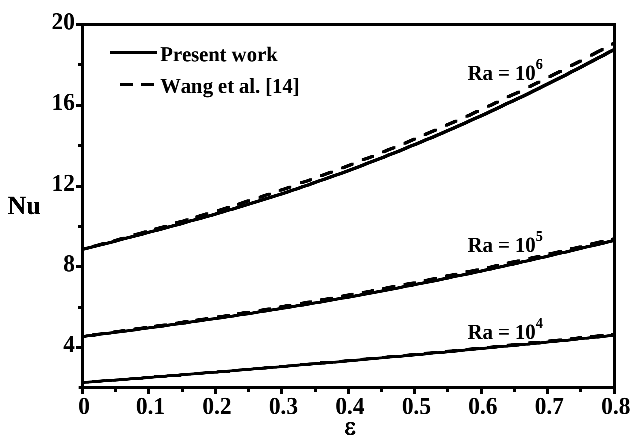
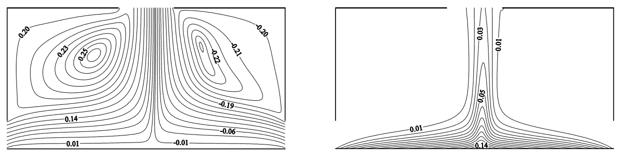


Fig.2: Effect of Ra and ε on the mean total Nusselt number, Nu, evaluated on the heated wall of a differentially heated square cavity for T'_H = 298.5 K and T'_C = 288.5 K

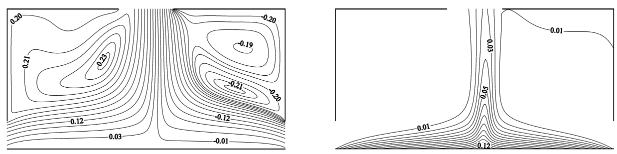
5. Results and Discussion

In the present study, the value of the temperature of the ambient imposed flow was considered constant at $T'_C = 298.15\ \mathrm{K}$. The fixed value of Rayleigh number ($Ra = 5\times10^6$), retained in this work, induces automatic values of the parameters N_r and T_o equal to 2.02 and 0.704, respectively. In the following, effects of Reynolds number, $300 \le Re \le 5000$ and walls emissivity, $0 \le \varepsilon \le 0.9$ and the imposed flow mode through the cavity, on fluid flow and heat transfer characteristics are explored. The investigations were performed with air as the working fluid (Pr = 0.72). The aspect ratio, A, the relative height of the vertical openings, h, and the relative length of the upper horizontal opening, ℓ, are maintained constants at 2, 1/5 and 2/5 respectively.

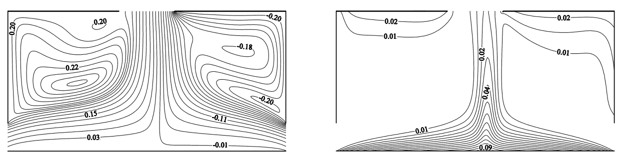
Typical streamlines and isotherms illustrating radiation effect on the flow structure and temperature patterns, in the injection case, are presented in Figs. 3a-3d for Re = 300 and various values of ε. In the absence of radiation effect (ε = 0), Fig. 3a shows that the forced flow enters horizontally through the two vertical inlet openings and then ascends vertically at the middle of the cavity before leaving the latter through the upper outlet opening. Also, we note the existence of two closed cells surmounting the open lines of the forced flow. The formation of these cells, located in the corners of the upper part of the cavity, is due to the shear effect. The corresponding isotherms are tightened at the level of the heated bottom wall indicating a good convective heat exchange. From the temperature distribution, a thin horizontal thermal boundary layer is seen at the level of the heated wall. The heat provided by the heated horizontal wall is carried out vertically through the vertical corridor leading to the outlet of the cavity. Consecutively, a great part of the space offered in the cavity is at a uniform cold temperature. In the case of radiatively participating inner surfaces (ε = 0.15), it can be seen from Fig. 3b that the flow structure is affected. In fact, the right clockwise cell is divided into two cells separated by the open streamlines while the left cell becomes more distorted. A progressive increase of the emissivity up to 0.5 and 0.9 (moderately and highly emissive walls), as shown in Fig. 3c and 3d respectively, leads to a complete disappearance of the closed cells located in the right part of the cavity in favor of the open streamlines of the forced flow. This results from the aspiration of the injected flow under the effect of the buoyancy force which develops at the level of the vertical right wall heated under the effect of radiation. The cold zone space is seen to be reduced indicating the important role of radiation in the homogenization of the fluid temperature inside the cavity. This result is attributed to the heating of the adiabatic upper and right walls which present increasingly significant thermal gradients as the emissivity increases.



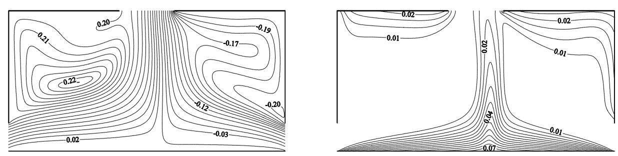
a) ε = 0



b) ε = 0.15



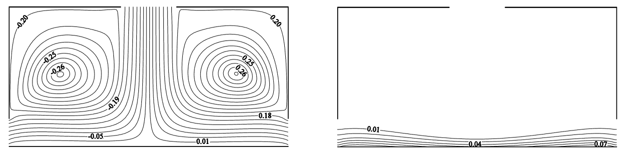
c) ε = 0.5



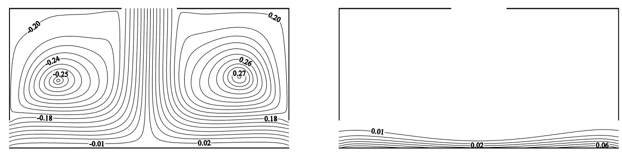
d) ε = 0.9

Fig. 3: Streamlines and isotherms obtained, in the injection mode, for Re = 300 and various values of ε: a) ε = 0; b) ε = 0.15; c) ε = 0.5 and d) ε = 0.9.

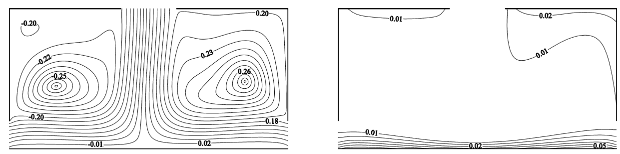
In the case of the suction mode, typical results in terms of streamlines and isotherms, illustrating the radiation effect on the flow structure and temperature distribution within the cavity, are presented in Figs. 4a-4d, for Re = 300 and various values of ε. Hence, Fig. 4a, obtained for ε = 0, shows that the flow descends vertically from the upper horizontal inlet and then leaves the cavity horizontally through the vertical two outlets. The structure is characterized by the presence of two big closed cells surmounting the open streamlines. Such cells are straight and large in size and intensity compared to those presented in the injection mode (see Fig. 3a). These two cells, having opposite directions of rotation, reflect the considerable shearing effect. Moreover, it can be noticed that the dynamic field is symmetrical with respect to the median of the cavity. The corresponding isotherms take the form of parallel and dense streamlines at the vicinity of the hot wall; which forms a very limited thermal boundary layer. Consequently, the heat released by the bottom hot wall is quickly and directly transferred to the exit through the thermal boundary layer without ascending under the buoyancy forces. In addition, the cold zone is larger in comparison with the other mode. It results from this behavior that the suction mode is thermally strong. By increasing progressively ε from 0.15 up to 0.9 (Figs. 4b-4d), the left closed cell is reduced in size and intensity in favor of a small convective cell generated at the right upper corner of the cavity and whose formation is due essentially to the growing natural convection effect created by the heating of the upper wall under the radiation effect. Also, the right closed cell remains unchanged due to the equilibrium thermal effect of the neighboring walls. The isotherms show the appearance of thermal gradients at the level of insulated walls.



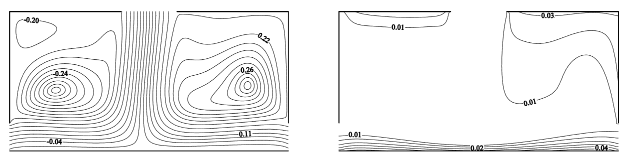
a) ε = 0



b) ε = 0.15



c) 0.5



d) 0.9

Fig. 4: Streamlines and isotherms obtained, in the suction mode, for Re = 300 and various values of ε: a) ε = 0; b) ε = 0.15; c) ε = 0.5 and d) ε = 0.9

In order to illustrate the thermal performance of both injection and suction modes, 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 Figs. 5a-5c for various values of ε. As expected, Fig. 5a shows that globally, an increase of $Nu_H(cv)$ with Re is observed for both injection and suction modes either with or without radiation effect. The rate of this increase becomes more important for 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. However, in the injection case, this increase is very limited for Re < 1000. Such behavior is explained by the fact that natural convection dominates for Re < 1000 and any increase in Re within this range has a negligible effect on convective heat transfer. For a fixed value of this parameter, the convection effect is relatively reduced when the emissivity of the walls is increased whatever the imposed flow mode. The negative impact of radiation on the mixed convection is well known and it is confirmed here in the case of vented cavities. It should be noted that, in comparison with the injection mode, the suction mode enhances the convective heat transfer more, and consequently permits better cooling within the cavity for all Re values ranging from 300 to 5000 with or without radiation.

The effect of walls emissivity on radiative heat transfer is presented in Fig. 5b in terms of Nu_H(rd) variations with Re for $\varepsilon = 0.15$, 0.5 and 0.85 in both injection and suction cases. Generally, it can be noted that, for a fixed value of Re, the radiation effect is more important for $\varepsilon = 0.5$ and 0.85 and the increase of the emissivity leads to an important increase of the radiative heat transfer component. Alternatively, the effect of Re on Nu_H(rd) remains very limited for weak or moderate value of ε. But for the higher value of the latter ($\varepsilon = 0.85$), Re contributes to enhance Nu_H(rd) for predominant forced flow (Re > 1000).

Also, it should be underlined that the radiation heat exchange is almost independent to the ventilation mode.

The variations of the total Nusselt number with Re, presented in Fig. 5c, show an increasing tendencies of Nu_H with Re and ε for the two ventilation types 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. These tendencies follow the tendencies already described of both components of Nu_H with ε and Re. Quantitatively, in the injection case, for $\varepsilon = 0.85$, improvements in terms of total heat transfer due to surface radiation is about 36% and 29% for Re = 300 and 5000 respectively in comparison with the reference case corresponding to $\varepsilon = 0$. Moreover, with respect to the injection mode, the enhancement of heat transfer achieved by the suction mode is very important. For instance, for $\varepsilon = 0.85$ and Re = 5000, passing from the injection to the suction mode, Nu_H increases from 35.9 to 66.4 which corresponds to an enhancement of the heat transfer by about 84.9 %. This revealed result proves the originality of our work since the majority of previous works considered the injection mode to study the thermal performances of such ventilated systems. Furthermore, most of them had neglected the effect of radiation in spite of its significant contribution to the overall heat transfer. It is to reminder that, in the injection mode, unsteady periodic solutions are obtained in a range of Re, which depends on ε. These solutions are indicated by solid circles in Figs. 5a-5c. The mean values of the Nusselt number are averaged in time over several flow cycles. The physical origin of these instabilities is attributed to the competition between the mixed and forced convection flows.

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. 6a-6b for different values of ε. The mean temperature was obtained from the arithmetic mean temperatures of the inner nodes of the domain. Hence, for all considered values of ε, Fig. 6a shows, for injection case, that $\overline{T}$ increases slightly, in a hyperbolic way, by increasing Re up toward a maximum value situated between 900 and 1200, which depends strongly on ε. This reheating of the cavity can be justified by the limited increase of total heat transfer in this range of Re (see Fig. 5c). Such behavior is due to the complexity of interaction between natural and forced convections resulting from their assisting effects. Then, its trend of evolution changes in favor of a continuous and monotonous decrease accompanying the increase of Re in the remaining range. This tendency is expected since the increase of Re strengthens the forced convection predominance and thereby its well-known cooling effect. In addition, for a given Re, the increase of ε leads to the reheating of the cavity in the case of a predominant mixed convection because of the additional radiative heat flux transferred to the fluid by the surrounding walls interacting with the active one. In the other hand, the increase of ε, when the forced convection is a predominated regime, 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 upper opening, without being transported by the fluid, increases with ε. For suction case, the evolution of $\overline{T}$, is characterized by a noticeable decrease / (increase) by increasing Re / (the walls emissivity) especially for weak or moderate values of Re. Also, it appears obvious from such a figure that the suction mode leads to a better cooling of the cavity, since the corresponding values of $\overline{T}$ are lower in comparison with the injection case. To be clear, for ε = 0.85 and Re = 1500, a reduction of about 74.6 % of $\overline{T}$ occurs when passing from the injection mode to the suction one.

The evolution of the maximum temperature, presented in Fig. 6b and obviously located on the heated wall, is characterized generally by a monotonous decrease with ε and Re in both injection and suction modes. This mentioned behavior means that the overheat phenomenon of the lower active wall can be avoided by increasing the emissivity of the walls or the velocity of the external flow. It is to underline that in the injection case, T_{max} is insensitive to Re as long as this letter is lower than 1000 because of the insignificant heat transfer in this range. Also, it should be noted that the suction mode is more favorable to the cooling efficiency of the heated wall in comparison with the injection one. Such remark is justified by the lower values of T_{max} in the ventilation by suction.

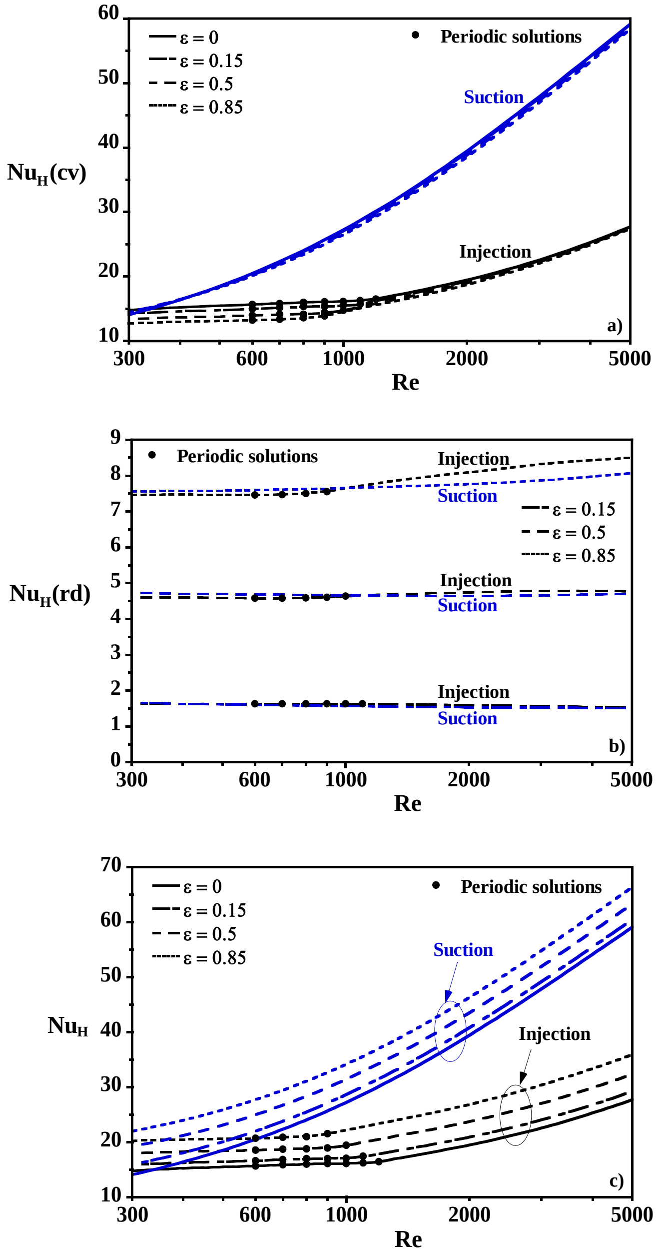


Fig. 5: Variations, with Re, of the average Nusselt numbers on the heated wall for various values of ε in both injection and suction modes: a) $Nu_H(cv)$; b) $Nu_H(rd)$ and c) Nu_H.

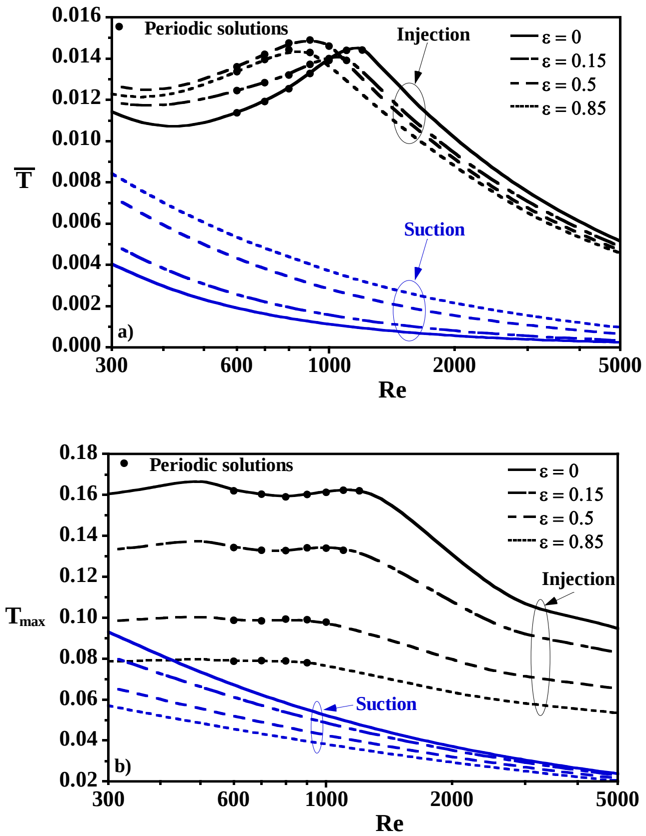


Fig.6: Variations, with Re, of the temperature for various values of ε in both injection and suction modes: a) mean temperature $\overline{T}$ and b) maximum temperature T_{max}.

6. Conclusions

In the present work, a numerical study was carried out to investigate laminar mixed convection coupled to thermal radiation in a multiple vented cavity totally heated from below. The study is conducted by considering two modes of imposed external flows (injection and suction). The obtained results reveal, for both two modes, that the surface radiation alters widely the flow and temperature patterns, contributes to the temperature homogenization within the cavity, reduces the convective component and enhances the overall heat transfer. Also, it is found that the injection mode favors the contribution of radiative component but the suction mode is more efficient than the injection mode by leading to more total heat transfer across the cavity even with or without radiation. Moreover, a better cooling of the cavity is reached with the suction mode since it involves lower values of the mean and maximum temperatures in comparison with the injection mode. Finally, it is to reminder that the thermal radiation contribution remains more important especially for high and moderate values of ε whatever the convection regime.

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