Numerical Investigation of the Flow and Heat Transfer Generated by Natural Convection and Surface Radiation in an Open Enclosure

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Abstract: This work presents a numerical investigation of surface radiation (SR) coupled to natural convection (NC) within an air-filled open cavity having the same emissivity on its three walls. The control volume method combined with the algorithm SIMPLE is used to resolve the conservation equations. The radiosity method is adopted to determine the radiative component of the heat exchange between all the cavity surfaces. This paper aims to analyze the effects of two parameters controlling the flow and heat transfer, namely the emissivity and the Rayleigh number. The simulations carried out show that SR makes the streamlines and isotherms very sensitive to the Rayleigh number. In addition, the convective, radiative, and total heat transfer in the cavity increase by increasing this parameter. The findings also reveal that the emissivity has almost no effect on the streamlines for large Rayleigh numbers, while it has a remarkable impact on the isotherms. Furthermore, increasing this parameter leads to an important increase (slight decrease) in the radiative (convective) flux.

1 INTRODUCTION

Natural convection flows in open cavities have attracted significant interest from scientific researchers and engineers during recent decades. This interest is dictated by the critical role played by this mode of heat transfer in many industrial applications. Examples include cooling of electronic components, aerospace engineering, heating and ventilation of buildings, heat exchangers, solar thermal receivers, fire spread in rooms, nuclear reactors, building insulation, etc.

In the literature, there are many studies, both numerical and experimental, which describe the flow and heat transfer caused by NC in open enclosures. These studies can be classified into two categories: (a) pure NC, (b) NC coupled with SR.

The first category has been extensively studied in the last decade. For example, (Bondareva, 2017) numerically investigated NC and entropy generation in an open triangular enclosure. They found that the natural flow and the resulting heat transfer intensify by increasing the Rayleigh number. On their side, (Hussein, 2017) investigated NC in a parallelogram-shaped enclosure, utterly open from the top and filled with a nanofluid. The obtained results revealed that increasing the volume fraction of nanoparticles leads to an intensification of the mean heat transfer. (Öztop, 2017) published a paper intending to study NC in a triangular open enclosure partially heated from below. Their results indicate that the heat source cools less as it gets closer to the opening.

Concerning the second category, the few publications available in the literature show that NC coupled to SR in open cavities is poorly documented and needs more investigative efforts to expand the fields of application. Among these studies, we can mention that of (Hinojosa, 2017). They numerically analyzed the impact of SR on entropy generation caused by NC in an open enclosure. The simulations show that SR increases the global entropy generation from 33% to 560%. On their side, (Shirvan, 2017) presented a numerical solution of NC combined with SR in an open cavity of a solar receiver. They deduced that the Rayleigh number and emissivity significantly affect the hydrodynamic and thermal fields.

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In the present paper, SR coupled to NC in a square cavity, open from the right and uniformly heated via its vertical wall is briefly studied. The effects of two parameters that control the flow and heat transfer in such a configuration will be discussed, namely the Rayleigh number and the emissivity of the cavity surfaces.

2 PROBLEM STATEMENT

The studied configuration is represented in Fig. 1. It is a bidimensional cavity, of height H and width L, open on the right side and filled with air (Pr=0.71). We designate by A=H/L the aspect ratio of the enclosure, which is taken equal to unity since the cavity is assumed to be square (H=L=5cm). The cavity is heated via its vertical wall, which is isothermal at temperature T_h, while the rest of the walls is kept adiabatic. Beyond the opening, the ambient air is isothermal at the temperature T_∞=293K. The flow within the cavity is considered laminar and two-dimensional. The fluid is Newtonian, incompressible, and transparent to SR. The internal surfaces of the cavity are considered grey, diffuse, and have the same emissivity ε. The aperture is assimilated to a black surface (ε=1) brought to the temperature T_∞.

The thermophysical air properties are considered constant excluding the density in the thrust term, which is assumed to follow the Boussinesq approximation. The dimensionless equations representing the conservation of mass, momentum,

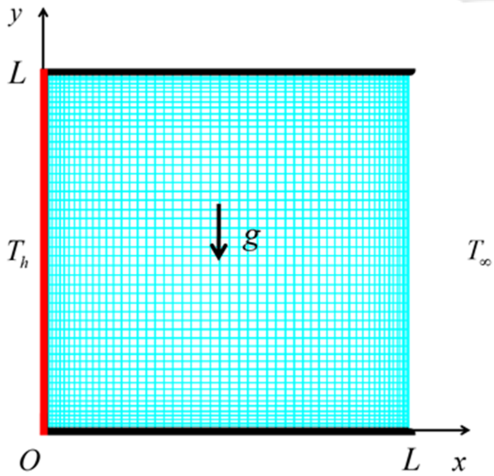

Figure 1: Studied configuration.

and energy in the fluid medium are expressed as:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \tag{1}$$

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

$$\frac{\partial V}{\partial \tau}+U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\Pr\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)+\mathrm{Ra}\Pr\theta \tag{3}$$

$$\frac{\partial \theta}{\partial \tau}+U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}=\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2} \tag{4}$$

SR is determined by the radiosity method. It consists of solving the following two systems of equations. They give respectively the radiosity and the net radiative flux lost by the i^{th} surface element:

$$J_k(r_k)=\varepsilon_k\left(1+\frac{\theta_k(r_k)}{\theta_{ref}}\right)^4+(1-\varepsilon_k)\sum_{j=1}^{4}\int_{S_j} J_j(r_j)\,dF_{ds_k-ds_j} \tag{5}$$

$$Q_{r,k}(r_k)=J_k(r_k)-\varepsilon_k\sum_{j=1}^{4}\int_{S_j} J_j(r_j)\,dF_{ds_k-ds_j} \tag{6}$$

Where $T_r=T_\infty/T_h$ and F_{ij} is the form factor between the i^{th} and j^{th} surface elements.

3 NUMERICAL PROCEDURE AND VALIDATION

The conservation equations (1-4) have been solved numerically by the control volume method and the algorithm SIMPLE (Semi-Implicit Method for Pressure Linked Equations), using the power-law scheme. Several tests have been done to study the influence of the mesh, time step, and convergence criterion on the accuracy of results. These tests led to the choice of a non-uniform mesh of 60×60, a dimensionless time step of 10^{-6}, and a convergence criterion of 10^{-4}. The validity of the calculation code was tested by confronting its results with several studies well-known in the literature. Table 1 shows an excellent agreement between the mean Nusselt numbers (convective and radiative) provided by the numerical code and those found by (Singh, 2016), who studied the interaction of NC and SR in an open enclosure.

4 RESULTS AND DISCUSSION

4.1 Effect of the Rayleigh Number

Fig. 2 gives the streamlines (a) and isotherms (b) obtained for four values of the Rayleigh number ($10^3 \leq Ra \leq 10^6$). The emissivity of the internal surfaces of the cavity is kept equal to unity (black surfaces). This figure shows that the cold fluid penetrates the cavity from the lower part of the aperture by moving along the bottom wall. The latter heats up through SR and transfers heat by convection to the cold fluid. This one continues to move toward the vertical wall, and when it becomes sufficiently heated, it rises upwards under Archimedes' thrust. Thus, as the warm fluid approaches the top adiabatic wall, it changes its direction back to the aperture, transferring some of its energy to this wall by convection before leaving the cavity.

For low Rayleigh numbers ($Ra=10^3$), the buoyancy force, which is responsible for NC, is still low. Thus, the flow structure keeps an almost symmetrical aspect to the horizontal median of the cavity. This symmetrical aspect is destroyed as the Rayleigh number increases, and the streamlines become denser near the active and upper adiabatic wall. Consequently, the hydrodynamic boundary layers developing against these walls become very thin. This can be explained by the viscous nature of the working fluid and its fast displacement due to the increased thrust force.

Fig. 2 also shows that the isotherms exist in the entire fluid domain for $Ra=10^3$, indicating that the conduction mode preponderate the heat exchange. For higher Rayleigh numbers ($Ra \geq 10^3$), the isotherms become very tight against the active and upper adiabatic wall. Thus, thermal gradient in the central

Table 1. Validation in terms of Nusselt number for a temperature ratio $T_r=0.8$ and $\varepsilon=1$.

Ra	Nu_c		Nu_r	
	Present work	(Singh, 2016)	Present work	(Singh, 2016)
10^3	2.10	2.19	67.95	67.95
10^4	3.42	3.54	68.42	68.45
10^5	6.46	6.53	69.10	69.09
10^6	12.21	12.41	70.05	69.98

part of the enclosure decreases considerably. In other words, the predominant mode of heat exchange, in this case, is NC. The isotherms also show that a thermal boundary layer develops against the active wall, where a very significant temperature gradient occurs. Such a gradient becomes stronger by increasing Ra. This is due to the reduction in the thickness of the thermal boundary layer. Therefore, the fluid extracts more heat from the active wall.

For low Rayleigh numbers ($10^3 \leq Ra \leq 10^4$), Fig. 3 shows that the convective heat exchange is relatively low. This result is expected because, as already mentioned, conduction and SR dominate the global heat exchange in the enclosure. In addition, the convective Nusselt number increases rapidly for $Ra \geq 10^4$ because of the intensification of NC with the difference between the environmental air and the active wall in terms of temperature, and subsequently with the Rayleigh number based on this difference. The thermal radiation between the active wall and the cavity surfaces also increases with the Rayleigh number. But less rapidly than the convective heat

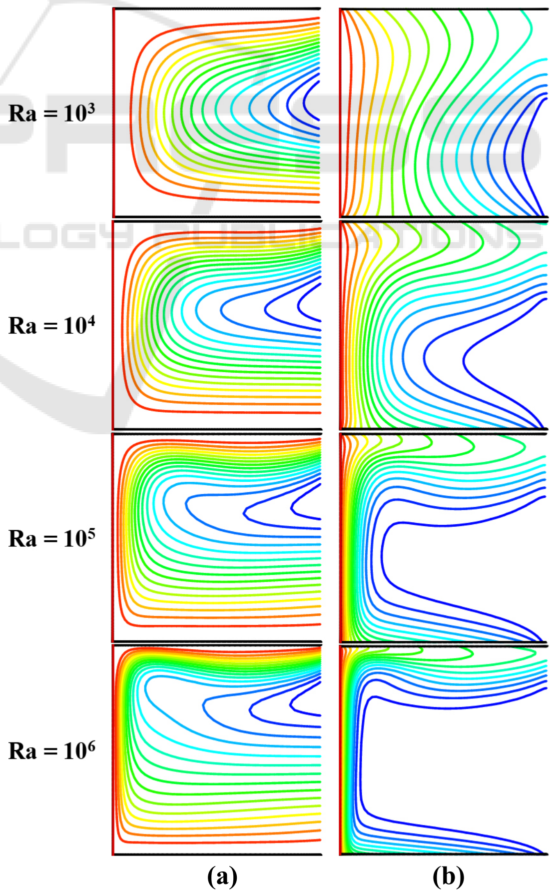

Figure 2: Streamlines (a) and isotherms (b) for different Ra and $\varepsilon=1$.

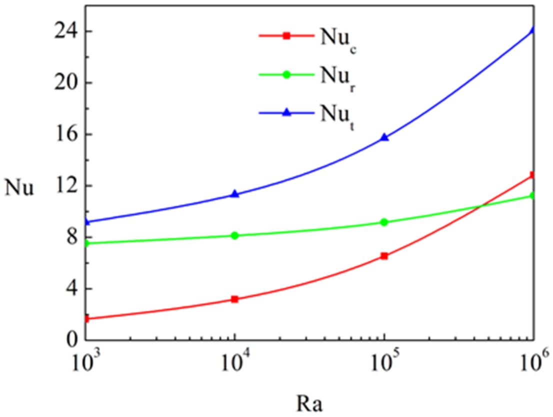

Figure 3: Variations of the convective, radiative, and global Nusselt numbers on the heated wall against Ra for ε=1.

heat transfer. As an indication, varying Ra from 10^5 to 10^6, the radiative Nusselt number increases by 22.5%. Fig. 3 also shows that for ($10^3 \leq Ra \leq 10^4$), the convective component is much less than the radiative one. This indicates that a conduction~radiation equilibrium occurs in the enclosure for low Rayleigh numbers. When $Ra \geq 10^5$, the radiative and convective Nusselt numbers are of the same order. In other words, a convection~radiation equilibrium is achieved. As for the total Nusselt number, it seems that it also increases with Ra. Therefore, the higher this parameter is, the more the active wall is cooled.

4.2 Effect of the Emissivity

Fig. 4 shows the streamlines (a) and isotherms (b) obtained for different emissivities (0≤ε≤1), keeping the Rayleigh number at 10^6. The streamlines are very dense near the active and upper adiabatic walls, indicating a very strong velocity gradient at this location. Such a density remains almost unchanged by increasing the emissivity of the internal surfaces of the cavity. On the other hand, these lines are relatively scattered in the centre of the cavity and near the lower adiabatic wall. They slowly approach the latter by increasing the emissivity. Based on these findings, it can be concluded that for high Rayleigh numbers the flow structure in most of the cavity is insensitive to SR.

The isotherms given in Fig. 4 show that without radiative heat exchange (ε=0), the thermal boundary layer expected to be developed against the lower adiabatic wall is missing, and it only starts to take place from ε=0.2. This confirms the fact that the lower adiabatic wall heats up by SR. Furthermore, the

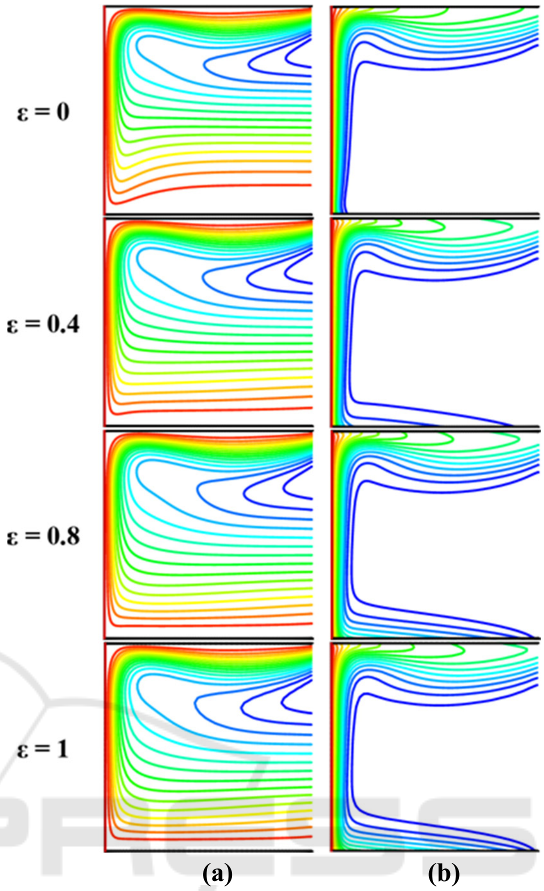

Figure 4: Streamlines (a) and isotherms (b) for different ε and Ra=10^6.

isotherms indicate that the thickness of this thermal boundary layer increases with emissivity. Therefore, the cold fluid volume in the cavity and the rate of convective heat exchange decrease.

Fig. 5 illustrates the variations of the mean convective, radiative, and global Nusselt numbers on the heated wall against the emissivity. It shows that the Nu_c decreases with emissivity. This decrease, which is almost linear for ε≥0.2, can go up to 10.9%. Such a result is predictable because, as mentioned above, the volume of cold air in the cavity decreases with emissivity. In contrast, the radiative heat transfer is an increasing function of this parameter. Such an increase is also almost linear for ε ≥ 0.2. As for the total heat transfer, Fig. 5 shows that it increases linearly with the emissivity although the attenuation of the convective heat transfer. In other words, increasing the emissivity promotes the cooling of the active wall.

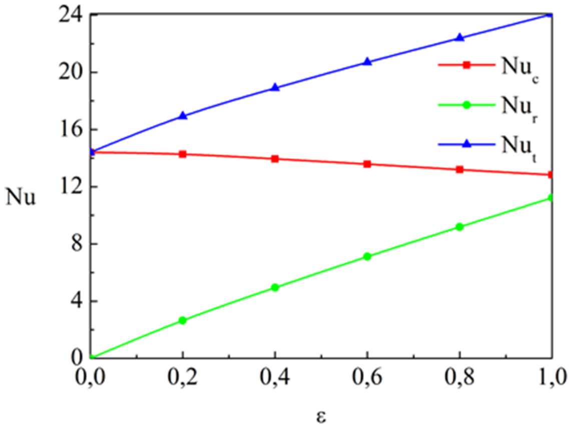

Figure 5: Variations of the convective, radiative, and global Nusselt numbers on the active wall against ε for Ra=10^6.

5 CONCLUSION

In the present paper, the authors studied the effects of the emissivity and the Rayleigh number on the streamlines, isotherms, and heat exchange in an open enclosure. The findings show that Ra significantly affects the velocity and temperature distributions. However, the emissivity has practically no impact on the flow structure for high Rayleigh numbers. But it remarkably affects the isotherms. The numerical simulations performed also show that increasing Ra results in an increase in the convective, radiative, and total heat exchange. While for the emissivity, its increase leads to an important increase (slight decrease) in the radiative (convective) flux, and consequently an increase in the total heat flux extracted from the active wall.

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