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Effect of baffle length on mixed convection coupled to surface radiation in a rectangle cavity

M.A. Belmiloud, N. Sad_Chemloul and M.B. Guemmour

Department of Mechanical Engineering
University Ibn Khaldoun, B.P. 78, 14000 Tiaret, Algeria

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Abstract - *In this study, we numerically investigated the influence of the baffle length and emissivity of the walls on heat transfer in a mixed convection coupled with radiation in a rectangular cavity ventilated. The baffle length is used* L_b *= 0.3, 0.5 and 0.7. The cavity walls are gray. The left vertical wall is heated to a constant heat flux and the other walls are adiabatic. The finite volume method is used to solve the governing equations. At the input of the cavity whose opening height w* =H/10*, the fluid velocity is determined from the number of Reynolds Re between 50 and 500. The Grashof number Gris fixed to* 10^4 *and the Prandtl number Pr is kept constant 0.71. The results obtained in this study show that the heat transfer enhancement depends on the increase in emissivity; However, the increase in total Nusselt number* Nu_T *depends on the increase of the baffle length.*

Résumé - *Dans cette étude, nous avons étudié numériquement l'influence de la longueur de baffles et de l'émissivité des parois sur le transfert thermique en convection mixte couplé à un rayonnement dans une cavité rectangulaire ventilée. Les longueurs de baffle utilisées sont Lb = 0.3; 0.5; et 0.7. Les parois de la cavité sont de couleur grise. La paroi gauche est chauffée à une densité de flux thermique constante et les autres parois sont adiabatiques. La méthode des volumes finis est utilisée pour résoudre les équations qui régissent le phénomène. A l'entrée de la cavité, dont la hauteur d'ouverture est égale à w=H/10, la vitesse du fluide est déterminée des valeurs du nombre de Reynolds compris entre 50 et 500. Le nombre de Grashof est fixé à* 10^4 *et le nombre de Prandtl est maintenu constant à 0.71. Les résultats obtenus dans cette étude montrent que l'amélioration du transfert thermique dépend de l'augmentation de l'emissivité, cependant, une augmentation du nombre de Nusselt dépend de l'augmentation de la taille de baffle.*

Keywords: Numerical study - Mixed convection - Baffle - Wall emissivity.

1. INTRODUCTION

The heat transfer by mixed convection coupled to the radiation in a rectangular cavity 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, electronic enclosures, industrial furnaces. The basic nature the problem interaction between the forced external air stream and the buoyancy-driven flow by the heat source.

The effect of height partition on combined mixed convection and surface radiation in a vented rectangular cavity is study by Bahlaoui *et al.*, [1].

They concluded that the relative height of the partition, contributes to increase/decrease the radiative/convective heat transfer component at the level of the heated wall. The radiation effect leads to a reduction of the convective Nusselt number component but the Reynolds number, supports both radiative and convective heat transfer modes. Effect of baffle number on mixed convection within a ventilated cavity, this study presented by Belmiloud *et al.*, [2].

Their results indicated the two provisions of the outlet of the cavity (BB and BT configurations) have a significant influence on the number of average Nusselt whose maximum value is obtained for the BB configuration due to streamlines that are clamped to the hot wall. Gravity has no influence on heat transfer and whatever the number of baffles used. The influence of gravity is observed only for the BT configuration and particularly in the case of a single baffle.

Chang *et al.,* [3] studied the effect of the baffle position on the heat transfer by mixed convection of pulsed flow in a vertical channel. They concluded that for the channel with both flow pulsation and a baffle will generate a improved heat transfer. A non-pulsating flow with a baffle has a best heat transfer. The heat transfer enhancement increases with Re.

Cheng *et al.,* [4] study the heat transfer enhancements of back- ward-facing step flow in a two-dimensional channel through the installation of solid and slotted baffles on to the channel wall. They pointed out that the pressure drop for the situation with slotted baffle is substantially smaller than that with solid baffle. With a slotted baffle installed on to the channel wall, the higher value of average Nusselt number obtained in the cases the solid baffle compared slotted baffle.

How *et al.,* [5] transient laminar mixed convection in a to-dimensional cavity partitioned by a baffle. The results show that the higher values of Re or lower the values of $\mathrm{Gr}/\mathrm{Re}^2$ delay the attainment of steady time. Increasing both the values of Re and $\mathrm{Gr}/\mathrm{Re}^2$ give rise to an increase of the heat transfer coefficient. The numerical study of laminar mixed convection in a rectangular cavity is study by Hsu *et al.*, [6].

Their results indicated that, the average Nusselt number is increased as Re is increased at fixed $\mathrm{Gr}/\mathrm{Re}^2$. Similarly, a gradual increase in heat transfer rate is found with increasing $\mathrm{Gr}/\mathrm{Re}^2$ at constant value of Re. The mixed convection with conduction and surface radiation from a vertical channel with discrete heating study by Londhe *et al.,* [7]. Their results show a good emitting surface is observed to decrease the local left wall temperature and increase the local right wall temperature.

Singh *et al.,* [8] studied the Conjugate laminar free convection with surface radiation in a two-dimensional open top cavity. The results indicate that the surface radiation changes the basic flow physics and enhances the radiative heat transfer as a result of which heat transfer by convection decreases. Thermal conductance and volumetric heat generation decrease the non-dimensional maximum temperature. Volumetric heat generation rate enhances the convective and radiative heat transfer.

Tsay *et al.,* [9] showed that the variation of the baffle position on the top wall, relative to the entrance, leads to an increase of the average Nusselt number. The results show that the higher value of average Nusselt number obtained in the cases with baffle compared without baffle. The effects of baffle width on heat transfer are in significant.

2. MATHEMATICAL MODEL

The numerical simulation depends on the variation of the emissivity walls ε, the length of the baffle L_b and the Reynolds number Re between 50 and 500. The walls of the geometry under consideration are gray, the left vertical wall of the cavity is heated to a constant heat flux, the other walls are adiabatic. The input of the cavity is at the bottom of the left vertical wall and the outlet at the upper part of the opposite wall. Three baffle lengths are used ($L_b = 0.3$, 0.5 , 0.7), the aspect of the cavity report $A = 2$ and the fluid temperature is $T_{in} = 288\ \mathrm{K}$.

The general equations of conservation of mass, momentum and energy along the axes x and y, are :

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{1}$$

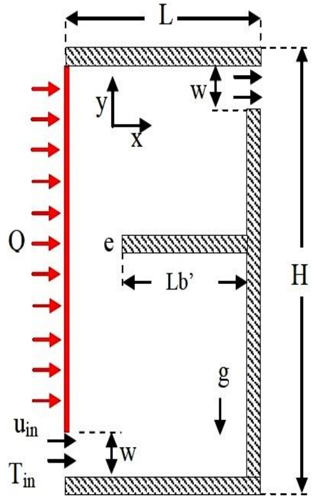

Fig. 1: Schematic of the physical problem

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{\partial P}{\partial x} + \upsilon\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) + g.\beta\,(T - T_{in}) \tag{2}$$

$$u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} = -\frac{\partial P}{\partial y} + \upsilon\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right) \tag{3}$$

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \alpha\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \tag{4}$$

The general equation of the radiosity is given by,

$$J_i = \varepsilon_i.\sigma.T_i^4 + (1-\varepsilon_i)\sum_{j=1}^{N} F_{ij}.J_j \tag{5}$$

$$F_{ij} = \frac{1}{A_i}.\int_{A_i}\int_{A_j}\frac{\cos\theta_i\ .\cos\theta_j}{\pi\, r^2}\,\delta_{ij}\, dA_i\, dA_j \tag{6}$$

The general equation of the net flux is given by,

$$Q_{rd} = \varepsilon_i.\left(\sigma.T_i^4 - \sum_{j=1}^{N} F_{ij}.J_j\right) \tag{7}$$

In the above equations, the dimensionless variables are defined by,

$$T_{in}^{+} = \frac{\lambda.T_{in}}{Q.H}\,; \qquad T^{*} = \frac{\lambda.(T - T_{in})}{Q.H} \tag{8}$$

The Grashof number Gr, the Reynolds number Re, the Richardson number Ri and the Prandtl number Pr are given by:

$$Gr = \frac{g.\beta.Q.H^4}{\lambda.\nu^2}; \quad Ri = \frac{Gr}{Re^2}; \quad Re = \frac{u_{in}.H}{\nu}; \quad Pr = \frac{\nu}{\alpha} \tag{9}$$

To solve the equations (2) -(4) considering the conditions on these dimensionless limits:

At input of the cavity: $T_{in} = 288\ K;\ u_{in} \neq 0;\ v = 0$ (10)

Adiabatic walls: $u = v = 0;\ Q = 0\ W/m^2$ (11)

In hot vertical wall: $u = v = 0;\ Q = 100\ W/m^2$ (12)

To determine the characteristics of heat transfer in the steady state, we must take into consideration the contribution of convection and radiation. In this study, the total average Nusselt number is defined as:

$$Nu_T = Nu_{cv} + Nu_{rd} = \frac{(Q_{cv} + Q_{rd}).H}{\lambda(T_H - T_{in})} \tag{13}$$

3. NUMERICAL ANALYSIS

The mass, momentum, and energy equations have been solved by a finite difference algorithm called the semi-implicit method for pressure linked equations (SIMPLE). Details of this method are described by Patankar [10].

The differential equations are discretized over a control volume. The power law difference scheme (PLDS) has been employed for the calculation of scalar variables and the quadratic upstream-weighted interpolation for convective kinematics (QUICK) scheme for Hayase *et al.*, [11] vector variables. The relative tolerance for the error criteria is considered to be:

$$\frac{|\varphi_n - \varphi_{n-1}|_{max}}{|\varphi_n|} < 10^{-9} \tag{14}$$

Table 1 show the dimensionless values of T^*_{av} and Nu_{av} determined in the case length $L_b = 0.5$ and wall emissivity $\varepsilon = 0.15$. Note that the relative difference between the obtained meshes (30 × 80) and (40 × 101) is 1.36 %, for Nu_{av} the relative difference of 1.48 %. For meshes (34 × 93) and (40 × 101) the relative difference is 1.22 % and for Nu_{av} is 1.47 %. The mesh used in all subsequent calculations is (40 × 101).

Table 1: Grid effect on the obtained results for $L_b = 0.5$ and $\varepsilon = 0.15$

	Grid			% change In abs		
	30×80	34×93	40×101	30×80	34×93	40×101
T^*_{av}	0.149	0.1488	0.147	1.36	1.22	-
Nu_{av}	6.719	6.720	6.820	1.48	1.47	-

3.1 Validation of the code

Numerical simulation performed using the commercial code ANSYS (Fluent). Version 6.3.26. The numerical validation of computer code, we made two comparisons of the results obtained in this study. For the first comparison, use the results obtained by Saha *et al.* [12], for the second comparison, use the results of Bahlaoui *et al.* [13].

The table shows the values of the average Nusselt number Nu_{av} and determined the maximum temperature T^*_{max} at the hot wall to a Reynolds number $Re = 100$ and the

Richardson number of $R_i = 10$. Note the values of Nu_{av} and T^*_{max} obtained in this study and those obtained by Saha *et al.* [12] are almost the same. The relative value is 1.80 % for Nu_{av} and to 0.48 % for T^*_{max}.

Table 2: Average Nusselt number and the maximal temperature evaluated in the heating wall

	Present work	Saha *et al.* [12]	% change in abs
Nu_{av}	2.235	2.276	1.80
T^*_{av}	0.206	0.207	0.48

Table 3 shows the values of number average Nusselt Nu_{av} determined at the hot wall for the two values of the Reynolds number Re and for the emissivity $\varepsilon = 0.15$. Note the values are almost equal to those Bahlaoui *et al.* [13].

Table 3: Average Nusselt number and the maximal evaluated in the heating wall for different Re and V

	Present work	Bahlaoui *et al.* [13]	% change in abs
Re = 200	2.235	2.276	1.80
Re = 300	0.206	0.207	0.48

4. RESULTS AND DISCUSSION

4.1 Heat transfer

Figure 2 shows the influence of the emissivity ε on the average Nusselt number convection Nu_{cv}, radiation Nu_{rd} and total Nu_T at the heated surface of the cavity as a function of Reynolds numbers and for $L_b = 0.5$. The Nusselt number Nu_{cv}, Nu_{rd} and Nu_T, increase with the increase in emissivity ε.

Figure 2(**a**), shows that the Nu_{cv} is increases with the Reynolds Re regardless of the value of the emissivity ε, the valid result for the other length $L_b = 0.3$ and $L_b = 0.7$. The difference between the values obtained between $\varepsilon = 0$ and $\varepsilon = 0.15$ is of 21.05 %. Knowing that the radiation effects reduced the number of convective Nusselt Nu_{cv}, the result does not agree with those found by Bahlaoui *et al.*, [1].

Figure 2(**b**) show that the number of radiative Nusselt Nu_{rd} decreases with increasing Reynolds number Re; However, increasing the emissivity ε improved the number of radiative Nusselt Nu_{rd}. The result is in good agreement with those found by Bahlaoui *et al.*, [1] this result holds for any length of baffle

The result in figure 2 (**c**) shows that the value of the Nusselt number total Nu_T increased when the emissivity ε is increased. Heat transfer increases as the Reynolds number Re.

Figure 3 show the influence of the baffle length L_b on the total Nusselt number Nu_T as a function of the different values of the Reynolds number Re for $\varepsilon = 0.15$ and $Gr = 10^4$. the total number of Nusselt Nu_T decreases with decreasing. For $L_b = 0.3$, it is observed that the Nusselt number almost constant when the number of Reynolds exceeded $Re > 300$.

However, for $L_b = 0.5$ and $L_b = 0.7$ to increase the Nusselt number when the Reynolds number increase. The maximum different between the two curves corresponding to 39.26 % for $L_b = 0.3$ and $L_b = 0.5$, most difference compared to the value obtained between $L_b = 0.5$ and $L_b = 0.7$.

The baffle length L_b contributes to the increase in the transfer of radiative and convective heat in the heated wall. This result does not agree with that obtained by Bahlaoui *et al.* [1] (increase / (decrease) in heat transfer radiative / (convective) at the heated wall). Because the problem Bahlaoui *et al.* [1] is a horizontal cavity mounted by a vertical baffle installed on the lower horizontal hot wall, However, our problems presented on figure 1. This meant the difference obtained on the results of convective heat transfer.

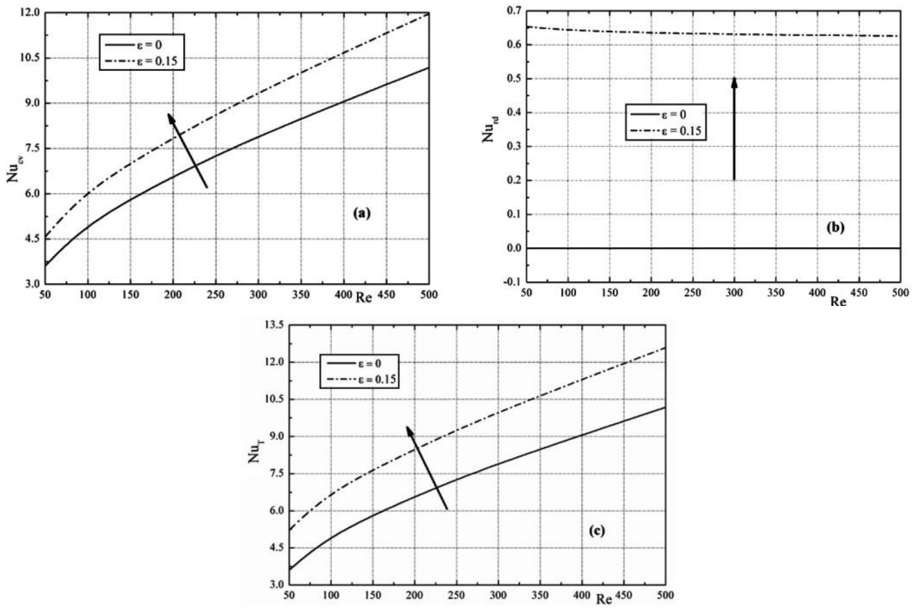

Fig. 2. Influence of emissivity number on the average Nusselt numberfor Lb = 0.5, $Gr = 10^4$ and different Reynolds number: a) Nu_{cv}, b) Nu_{rd} and c) Nu_T

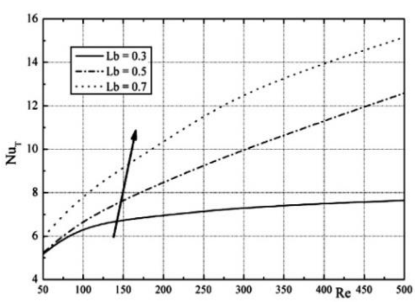

Fig. 3: Influence of length of baffle on the total Nusselt number for $\varepsilon = 0.15$, $Gr = 10^4$ and for different Reynolds number

4.2 Variation of dimensional temperature

The profile of the average temperature T_{av}^* for the various values of the Reynolds number Re is given by the figure 4.

Figure 4 (**a**) show that the average temperature profiles T_{av}^* have the same shape and are parallel and whatever the number Reynolds Re, the maximum value is obtained in the case of an emissivity $\varepsilon = 0$. The scientific significance of the latter, due to the exchange surface the strongest, hence the presence of baffle causes the fluid to the hot wall. Figure 4 (**b**) shows the variation the average temperature T_{av}^* as a function of the length L_b, it was observed that the average temperature decreases as the length increases, knowing that the T_{av}^* has calculated the hot wall. This one verifying the results obtained on the number of total Nusselt (figure 3). Is found that the length and the baffle position influence the heat transfer.

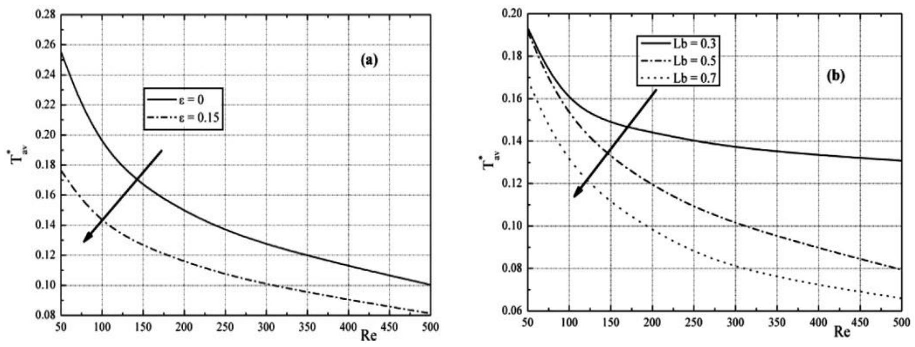

Fig. 4: Profile of T_{av}^* depending on Reynolds number Re obtained for **a**) $L_b = 0.5$ and **b**) $\varepsilon = 0.15$

5. CONCLUSION

The purpose of this study is to see the effect of emissivity ε, of the variation of the baffle length L_b on mixed convection coupled to a radiation in a rectangular cavity. According to the results, we concluded the heat transfer increases when the emissivity ε. Knowing that the Nusselt number convective Nu_{cv}, radiation Nu_{rd} and total Nu_T increases with ε, for low values of Reynolds Re, radiative transfer is good relative large values.

The total Nusselt number Nu_T increases as the length L_b of the baffle increases. Knowing that, low values of L_b, the almost stable thermal transfer when exceeded the number of Reynolds $Re > 300$.

The maximum value of the average temperature T_{av}^* is obtained in the case of the emissivity $\varepsilon = 0$ and for the length of the baffle $L_b = 0.3$. However the minimum values $\varepsilon = 0.15$ and $L_b = 0.7$.

One observation that, for improved heat transfer and cooled the hot plate must be used a gray body with a great emissivity and increased length.

NOMENCLATURE

A, aspect ratio of the cavity, $A = H/L$	e, Width of baffle dimensionless
F_{ij}, View factor from S_i surface to S_j	g, Acceleration of gravity
Gr, Grashof number	H, Height of the cavity
$J_{i,j}$, Radiosity	P, Pressure
N_r, Convection-radiation interaction parameter	L_b, Length of the baffle dimensionless
	$L_b = L_b / L$
L, Length of the cavity	Pr, Prandtl number
Nu, Average Nusselt number	α, Thermal diffusivity of fluid
β, Thermal expansion coef. of fluid	ε, Emissivity of the walls
λ, Thermal conductivity of fluid	ν, Kinematic viscosity
Q, Heat flux	Re, Reynolds number
R_i, Richardson number	T_{in}, Fluid temperature inlet
T, Fluid temperature	σ, Stefan-Boltzmann constant
u_{in}, The velocity inlet	x, y, Cartesian coordinates
w, Height opening inlet and outlet of the cavity	u, v, Horizontal and vertical velocities
	cv : convection
*, Dimensionless variables	rd, radiation ; T, total

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