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Numerical Modelling of Rib Width and Surface Radiation Effect on Natural Convection in a Vertical Vented and Divided Channel

Nadia Dihmani[1], Samir Amraqui[1], Ahmed Mezrhab[1,2] and Najib Laraqi[3]

Abstract: Natural convection with surface radiation heat transfer is investigated numerically in a vented vertical channel heated asymmetrically. The numerical solution is obtained using a finite volume method based on the SIMPLER algorithm for the treatment of velocity-pressure coupling. Concerning the radiation exchange, in particular, the working fluid is assumed to be transparent, so that only the solid surfaces (assumed diffuse-grey) give a contribute to such exchange. The effect of Rayleigh numbers and rib width (for Pr=0.7 air fluid) on the heat transfer and flow structure in the channel is examined in detail. Results are presented in terms of isotherms, streamlines, and average Nusselt number.

Nomenclature

A	aspect ratio, $A = L/b$
b	channel width $[m]$
F_{i-j}	view factor between the surfaces S_i and S_j
g	gravity acceleration $[ms^{-2}]$
k	thermal conductivity $[Wm^{-1}K^{-1}]$
L	channel length $[m]$
L_i^*	dimensionless distance of the plate from the channel inlet, $L_i^* = L_i/b$
L_r^*	dimensionless length of the rib, $L_r^* = L_r/b$
L_t^*	dimensionless distance of the vent opening centre from the channel inlet, $L_t^* = L_t/b$
Nr	radiation number, $Nr=\sigma T_h^4/(k_f \Delta T/b)$
p	pressure $[Pa]$
P	dimensionless pressure, $P=(p+\rho_o gy)b^2/\rho_o \alpha^2$

[1] Université Mohammed Premier, Faculté des Sciences, Laboratoire de Mécanique & Energétique, 60000 Oujda, Maroc.
[2] Author correspondant: amezrhab@yahoo.fr
[3] Université Paris Ouest, LTIE, EA 4415, GTE, F-92410 Ville d'Avray, France

Pr	Prandtl number, $Pr=\nu/\alpha$
Ra	Rayleigh number, $Ra = g\beta(T_hT_a)b^3/\nu\alpha$
R_i	dimensionless radiosity of surface S_i
R_k	thermal conductivity ratio, $R_k = k_s/k_f$
t_0^*	dimensionless vent opening size, $t_0^* = t_0/b$
T_a	temperature of the ambient air $[K]$
T_h	temperature of the hot wall $[K]$
T_o	average temperature, $T_o = (T_h + T_a)/2$ $[K]$
U,V	dimensionless velocity components along x, y. $U=ub/\alpha$, $V=vb/\alpha$
w^*	dimensionless plate width, $w^* = w/b$
X,Y	dimensionless cartesian coordinates, $X = x/b, Y = y/b$

Greek symbols

δ_{ij}	kronecker symbol
α	thermal diffusivity $[m^2s^{-1}]$
β	volumetric expansion coefficient $[K^{-1}]$
ΔT	temperature difference, $\Delta T=(T_h\text{-}T_a)$ $[K]$
ν	kinematic viscosity of the fluid $[m^2s^{-1}]$
ρ_o	fluid density at T_o $[Kg\ m^{-3}]$
θ	dimensionless temperature, $\theta=(T\text{-}T_a)/\Delta T$
Θ	dimensionless temperature, $\Theta=T/T_h$
σ	Stefan-Boltzmann constant $[Wm^{-2}K^{-4}]$
ε_i	surface emissivity S_i
λ	dimensionless viscosity ratio, $\lambda = \mu_\sigma/\mu_f$
μ	dynamic viscosity, $[kg\ m^{-1}s^{-1}]$

Subscripts

f	fluid
s	solid

1 Introduction

Enhancement of the heat transfer by natural convection in vertical channels has been extensively studied numerically and experimentally owing to its important applications in electronic cooling, building services engineering, solar energy collectors and other industrial processes [Akbari, Borgers (1979), Incropera (1988), Hegazy (1996)]. The first experiments were carried out by [Elenbaas (1942)] which served as a reference for many numerical and experimental succeeding works. He conducted a study on vertical channel consisting of isothermal parallel plates. He proposed a correlation between the average Nusselt number and the channel Rayleigh number. Indeed, by examining a simplified set of equations, and by adjusting constants to fit experimental data, he established an overall heat transfer correlation for isothermal channel over a wide range of thermal and geometric parameters.

Many analyses of natural convection in divided vertical parallel plates channels are available in the literature. [Desrayaud and Fichera (2002)] studied numerically the natural convection in vertical isothermal channel with two rectangular ribs, symmetrically located on each wall. They found that the best position of the ribs for heat extraction depends on the magnitude of the Rayleigh number, and that the increase of the rib length has only a limited influence on the heat transfer while the increase of its width decreases dramatically the mass flow rate and the heat transfer especially when the region obstructed was greater than the half of the opening. It appears that research conducted on heat transfer in divided channel have been intensified in recent years, but the radiation exchange was neglected in the majority of this work. [Cheng and Müller (1998)] investigated numerically the effect of heated wall temperature, the wall emissivity and the channel geometry on the turbulent natural convection coupled with thermal radiation in a vertical, rectangular channel with asymmetric heating. They concluded that even for low temperatures of the heated wall, the high emissitivies wall affect significantly the total heat transfer in channel. Few studies have focused on heat transfer and flow in the vented channel. Among these works are cited in particular [Moutsoglou, Rhee and Won (1992)], who studied numerically the heat transfer by convection and radiation in a vertical vented channel. The channel is asymmetrically heated with uniform heat flux on the heated wall, while the other is adiabatic. They concluded that, in general, the vent opening degrades the total cooling. [Muresan , Bennacer , Menezo (2006)] conduced a numerical investigation of turbulent convective flows transporting humidity in a vertical heated channel. They found that the wet wall induces humidity effect and modifies the obtained chimney effect. The water evaporation modifies significantly the temperature field and the apparent heat transfer. In another study [Muresan, Menezo, Bennacer (2006)], they carried out Numerical simulation of a vertical solar collector integrated in a building frame to determine the best integration strategy. A parametric study was performed as a function of the channel width, wall heat flux, and dimensionless turbulent intensity in the case of a vertical channel heated asymmetrically. The authors proposed a preliminary application of the coupled radiation-the natural convection heat transfer problem corresponding to a photovoltaic-thermal collector configuration. [Azizi, Benhamou, Galanis, El-Ganaoui (2007)] investigated numerically the effects of thermal and buoyancy forces on both upward flow (UF) and downward flow (DF) of air in a vertical parallel-plates channel. Also, they studied the cases with evaporation and condensation for both UF and DF. They concluded that buoyancy forces have an important effect on heat and mass transfers. They found that the heat transfer associated with these phase changes (i.e. latent heat transfer) depends on the temperature and humidity conditions and may be more or less important compared with sensible heat transfer. More recently, [Mohamad, El-Ganaoui, Bennacer (2009)] simulated the natural convection in an open ended cavity using Lattice Boltzmann Method LBM. They discuss the physics of flow and heat transfer in open end cavities and close end slots. The flow is induced into the cavity by buoyancy force due to a heated vertical wall. The results are similar to conventional CFD method finite volume method, FVM predictions. They found that the rate of heat transfer deceases asymptotically as the aspect ratio increases and may reach conduction limit for large aspect ratio. The flow evaluation in the cavity starts with recirculation inside the cavity, as the time proceeds the flow inside the cavity communicates with the ambient.

The objective of the present investigation is to study numerically the surface radiation effect on the heat transfer and the air flow in a vertical channel heated asymmetrically. In comparison to the previous works achieved on vertical channel, the originality of our contribution consists in the taking into account of the surface radiation in this kind of geometry.

2 Mathematical model

Details of the geometry are shown in Fig.1. The flow is assumed to be incompressible, laminar and two-dimensional in a divided vented vertical parallel-plates channel with length L and distance between the walls b. A rectangular rib of length L_r and width w is placed on the left wall which is kept at the temperature $T_h > T_a$ inside a vertical channel. The rectangular rib is located at a distance $L/3$ downstream of the channel inlet. Whereas the other plate is insulated and contains a vent opening of size t_0 located at the distance L_t from the channel inlet.

Using non-dimensional variables defined in the nomenclature, the flow and temperature distributions are governed by continuity, Navier-Stokes and energy equation.

Continuity:

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

X-momentum:

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

Y-momentum:

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

Energy:

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

Where λ and R_k are set equal to 1 in the fluid region, $\lambda=\infty$ and $R_k=k_s/k_f$ in the solid region.

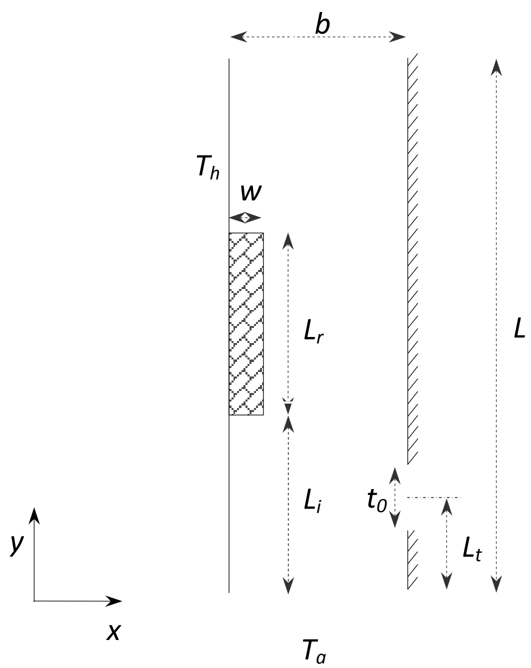


Figure 1: Geometry of the physical system.

The boundary conditions corresponding to the considered problem are as follows:
$X=0$ and $0\leq Y\leq A$:

$$U=V=0,\quad \theta=1,\quad \frac{\partial P}{\partial X}=0 \tag{5}$$

$X = 1$ and $(0 \leq Y \leq L_t^* - t_0^*/2$ or $L_t^* + t_0^*/2 \leq Y \leq A)$:

$$U = V = 0, \quad \frac{\partial P}{\partial X} = 0, \quad \frac{\partial \theta}{\partial X} - Nr\phi_r = 0 \tag{6}$$

X= 1 and $L_t^* - t_0^*/2 < Y < L_t^* + t_0^*/2$:

$$\frac{\partial U}{\partial X} = 0, \quad V = 0, \quad \theta = 0, \quad P(Y) = -\frac{U(Y)^2}{2} \tag{7}$$

$0 \leq X \leq 1$ and $Y = A$:

$$U = 0, \quad \frac{\partial V}{\partial Y} = 0, \quad \frac{\partial \theta}{\partial Y} = 0, \quad P = 0 \tag{8}$$

$0 \leq X \leq 1$ and $Y = 0$:

$$U = 0, \quad \frac{\partial V}{\partial Y} = 0, \quad \theta = 0, \quad P = -\frac{Q_{in}^2}{2} \tag{9}$$

3 Numerical procedure and code validation

The governing equations reported above are discretized on a staggered, non-uniform Cartesian grid using a finite-volume procedure with a central differencing scheme (CDS) for the convective terms. The SIMPLER (Semi-Implicit Method for Pressure Linked Equations Revised) algorithm is employed for the velocity–pressure coupling. The solid radiative surfaces forming the channel and the rib are divided into a number of surfaces A_i, (i=1, N). N is the number of total radiative surfaces forming the channel and the plate; which are equal to the total control volume interfaces solid-air. In fact, the control volume faces were also arranged so that a control volume face coincided with an interface solid-fluid. Therefore, the dimensionless net radiative flux density along a diffuse grey and opaque surface A_i is expressed as:

$$\phi_{r,i} = R_i - \sum_{j=1}^{N} R_j F_{i-j} \tag{10}$$

Where the dimensionless radiosity is defined as:

$$\sum_{j=1}^{N} (\delta_{ij} - (1 - \varepsilon_i) F_{i-j}) R_j = \varepsilon_i \Theta_i^4 \tag{11}$$

The average hot wall Nusselt number, based on the channel width, along the heated wall is determined by the integration of the expression (11) along the Y-axis:

$$Nuw = \frac{1}{A}\int_0^A \left(-\left.\frac{\partial \theta}{\partial X}\right|_{X=0,Y} + Nr\phi_r(X=0,Y) \right) dY \quad (12)$$

The non uniform grid sizes were evaluated in order to ensure the grid-independence solutions. The grid 30×140 is retained for reasons of coast calculations and it will be used for all our calculations reported here.

In order to validate the computer program, computations were first performed for the natural convection heat transfer in a divided vertical channel. Comparisons of local and average wall Nusselt numbers, mass flow rates, and average channel Nusselt number were made with those given numerically and experimentally by Naylor and Tarasuk [Naylor, Tarasuk (1993)]. In all cases, the present calculations gave identical results. For example of this validation, we present in tab.1, the average wall Nusselt number Nu_w and the rib Nusselt number Nup obtained by our code and those of Naylor and Tarasuk [Tarasuk (1993b)] shown in tab.1 (page 391).

Table 1: Comparison of the average Nusselt numbers at the wall and the rib for Ra/A=1561.6, Pr=0.733, L_i^*=0, w^*=0.2, A=6.84, $L_r = L/3$

	Experiment results [13]	Numerical results [12]	Our results
Wall Nusselt number, Nu_w	1.44	1.60	1.57
Rib Nusselt number, Nup	3.13	3.48	3.41

When the radiation exchange is taken into account, the code was also validated for vertical channels divided by a partition as shown in Ref. [Bouali and Mezrhab (2006)]. It was concluded that the largest discrepancies between our and published results can be estimated to be less than 1%. For this reason and for the sake of brevity it is not repeated here.

Based on the above studies, it was concluded that the code could be reliably applied to the problem under consideration.

4 Results and discussions

In this study, a series of calculations was done for Pr=0.71, A=5, R_k=5627, L_r^*=1, and for Rayleigh and rib width number ranged respectively between $1\times10^3 \le Ra \le 1\times10^5$ and $0.1 \le w^* \le 0.5$. ε is set equal to 0 in pure natural convection and 1 in presence of the radiation exchange. The average temperature T_0 is chosen equal to 300K and the terminal temperature difference ΔT is kept equal to 30K.

4.1 Flow patterns and isotherms

The effects caused by the variation of the rib width, on the isotherms and streamlines in absence (fig.2) and in presence (fig.3) of the surface radiation, are presented for two Rayleigh numbers $Ra=10^4$ and 10^5.

In the pure natural convection (fig.2), in one side, it is noted that when the rib width increases, the air starts to follow the rib contours more closely and the flow circulation decreases in the entire computational domain. This can be explained by the fact that as w^* increases, less cold air enters the channel so the air velocity in the channel decreases causing the increase of the air temperature inside the channel. This is due to the presence of the plate which obstructs the flow. In the other side, we note the increase of the air circulation in the entire channel owing to the increase of the buoyancy forces. In this case, the isotherms are concentrated close to the hot wall indicating a more important heat transfer from the hot isothermal wall towards the remainder of the channel.

In the natural convection combined to the surface radiation (fig.3), in one hand, when increasing the width rib, the same observations are noted and the preceding explanations remain valid. In the other hand, when the Rayleigh number increase, the isotherms structures near the adiabatic wall are affected by thermal radiation and for this reason the isotherms are inclined whereas they are perpendicular in the pure natural convection mode. This is explained by the fact that the radiation number Nr is proportional to Ra. Thereby, the temperature becomes homogenous due to the radiation exchange. Furthermore, the radiation exchange increases the air velocity in the channel, particularly at high Ra. The streamlines show that the surface radiation causes a considerably increase of the air circulation in the channel particularly at high Ra.

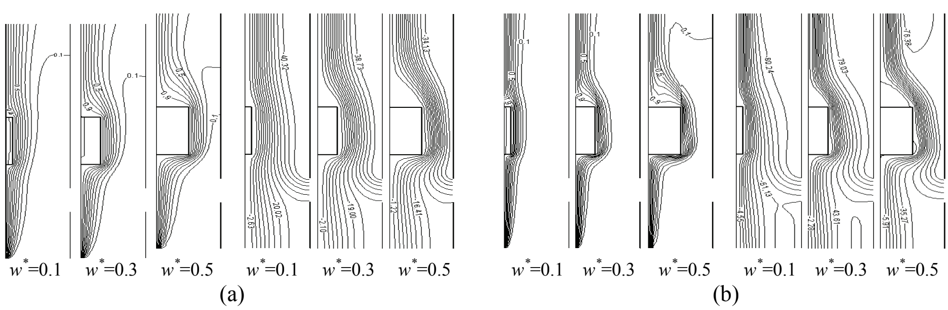


Figure 2: Effect of the rib width on the isotherms and the streamlines for $\varepsilon=0$: (a) $Ra=10^4$ and (b) $Ra=10^5$.

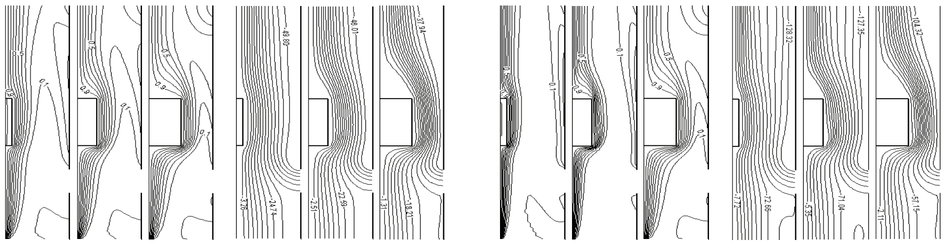


Figure 3: Effect of the rib width on the isotherms and the streamlines for ε=1: (a) Ra=10^4 and (b) Ra=10^5.

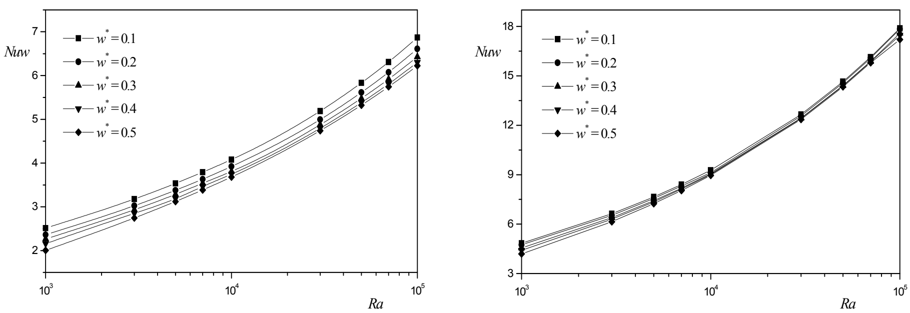


Figure 4: Evolution of the Nusselt number as a function of Ra for different w^*: (a)ε =0 and (b)ε =1.

4.2 Heat transfer and dimensionless temperature

Fig. 4 shows the variation of the average Nusselt number Nu_w according to the Rayleigh number in presence and in absence of the radiation exchange for five different rib widths w^*.

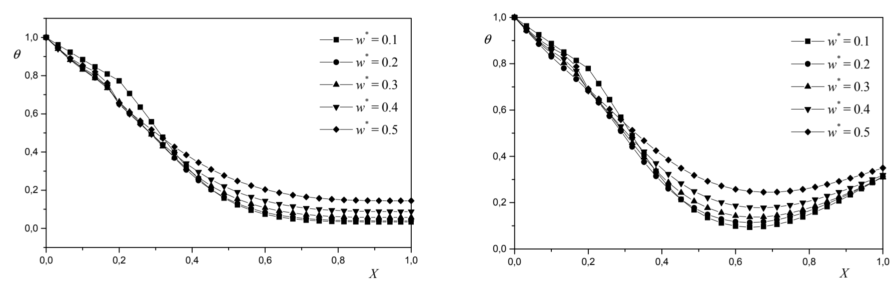
In absence of the radiation exchange, it is noted that the average Nusselt number Nu_w increases with the Rayleigh number Ra, because of the effects of the natural convection and the surface radiation which are more significant for larges Ra values. While, it is noted that the average Nu_w decrease with increasing the rib width.

In presence of the radiation exchange, the radiative surface of the rib becomes large, so the radiative flux exchanged between the rib and the channel plates increases greatly, which explains the increase of Nu_w.

Fig. 5 depicts the variation of the temperature distribution at Y=0.65A respectively in presence and in absence of the radiation exchange, and for rib of width w^*=0.2 and length Lr^*=1.

In case of natural convection (ε=0), in general way, the dimensionless temperature θ decreases with increasing X. This can be explaining by the fact that near the plate the temperature of the fluid washing the interface is high. Furthermore, when X is ranged between 0.6 and 1, one noted that θ remains nearly constant; this is due to the cooler fluid entering the channel through the vent which lowers the temperature at the mid channel. In the other side, the temperature increases with increasing the rib with w^*, except for w^*=0.1, where it is noted that θ is greater than the other cases, for X ranged between 0 and 0.3.

In the second case of the natural convection combined to radiation (ε=1), the effect of the rib width obtained in the pure natural convection case remains valid in the combined mode for $X \leq 0.6$. Moreover, it is noted that the dimensionless temperature has increased owing to the radiative flux emitted by the hot wall. In more, θ increases for X ranged between 0.5 and 1 which is due to the radiative flux emitted by the right surface of the channel. Therefore, the insulated wall dimensionless temperature is increased.



(a) (b)

Figure 5: Evolution of the temperature as a function of $X$for different w^*: (a)ε =0 and (b)ε =1.

5 Conclusion

In this study we showed the effects of the rib width and the Rayleigh number in a vertical vented divided channel, in presence and in absence of the radiation heat transfer. The principal conclusions obtained lead to:

- The surface radiation causes a considerable increase of the heat transfer and it allows a significant homogenization of the temperature in the channel. These effects are increasingly important as *Ra* increases.

- The increase of the rib width decreases the average hot wall Nusselt number Nu_w.

- The radiation exchange increases the dimensionless temperature. Thus, the insulated wall dimensionless temperature is increased.

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