Natural Convection Heat Transfer in a Rib-Roughened Asymmetrically Heated Channel

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ABSTRACT

A two dimensional numerical simulation is carried out to investigate the effect of two adiabatic square ribs on laminar flow and heat transfer in an asymmetrically heated channel. The two ribs are symmetrically located on each wall, exactly above the heating zone. The adiabatic square ribs could be an effortless and cheap means that can be incorporated in natural ventilation systems (double skin facades, trombe walls…) to control the mass flow rate and heat transfer. The computational procedure is made by solving the unsteady bi-dimensional continuity, momentum and energy equations with the finite volume method. The investigations focused more specifically on the influence of the ribs sizes R_S (0, b/18, 2b/18, 3b/18, 4b/18, 5b/18 and 6b/18) on the flow structure and heat transfer enhancement. The numerical study is carried out for a fixed modified Rayleigh number $Ra^* = 2.86\times 10^6$ and for a fixed aspect ratio of the heated part $R_f = 5.2$. The modified Rayleigh number based on the heat flux density (φ) and the thickness of the channel (b) also takes into account the aspect ratio of the heated part R_f. Concerning the value of the heated part aspect ratios R_f, it is within the range of those found for horizontally divided double-skin facades of high-rise buildings. The results showed that the variation of the ribs sizes significantly alters the heat transfer and fluid flow distribution along the channel, especially in the vicinity of protrusions. Also, the results show that streamlines, isotherms, and the number, sizes and formation of vortex structures inside the channel strongly depend on the size of protrusions. The changes in heat transfer parameters have also been presented. The present numerical results are compared with experimental data and a good agreement was found.

INTRODUCTION

In the literature, a large number of numerical and experimental works have dealt with natural convection phenomenon in vertical channels. The main cause of interest in the study of free convection in channels is not just for the fundamental nature of the problem, but also mainly from the fact that this type of thermally driven flow is cheap, reliable, quiet and free maintenance and it is encountered in numerous applications such as the chimney, the solar panel, the trombe wall and double-skin facade as well.

Since the pioneering work of Elenbaas [1] natural convection in open smooth channels have been extensively studied over the last past decades, both experimentally [2] and numerically [3] for vertical or inclined configurations.

Several studies have shown that the inclusion of nano-particles in the coolant fluid or the change of the geometry with various shapes of protrusions or fins considerably increase the heat transfer energy. For the sake of the enhancements of the limited heat transfer performance of a heated vertical channel, numerous numerical and experimental works have modified the geometry of the simple channel configuration. Some of these works have investigated the effects of an adiabatic extension in the entry or exit section, as investigated by Lee [4] and the chimney effect as in [5]. Other works treated the influence of adding internal heated or adiabatic plates as in [6].

Another most important method to enhance the thermal performance of channels is by adding roughness elements (ribs) on one or both surfaces forming the vertical channel. Compared with the plentiful literature on natural convection in channels with smooth surfaces, relatively little works have treated with the effect of roughness elements on natural convection heat transfer in channels as those studied by [7, 8].

At the author knowledge and depending on the literature review the case of free convective heat transfer in an asymmetrically heated vertical channel filled of water and having two ribs symmetrically located on each wall, exactly attached above the heated zone in downstream of the channel seems not to have been investigated in details in the past and this has motivated the present study. Various aspect ratios of the ribs in the range of $0\leq Rs\leq 6b/18$ are studied for a fixed $Ra^* = 2.86\times 10^6$ and fixed $R_f = 5.2$. The adiabatic square ribs could be an effortless and cheap means that can be incorporated in natural ventilation systems (double skin facades, Trombe-walls…) to control the mass flow rate and heat transfer.

NOMENCLATURE

b	[m]	Channel width
Cp	[J/kg.K]	Specific heat capacity
e	[m]	rib height
g	[m/s^2]	Acceleration of gravity
k	[W/mK]	Thermal conductivity
L	[m]	heated length
$\overline{Nu}$	[-]	Averaged Nusselt number
p	[Pa]	Pressure
Pr	[-]	Prandtl number
Ra	[-]	Rayleigh number
*Ra**	[-]	Modified Rayleigh number
R_f	[-]	Channel aspect ratio
Rs	[-]	Ribs size
T	[K]	Temperature
u,v	[m/s]	Transversal and vertical velocity components
x,y	[m]	Transversal and vertical coordinates

Special characters

α	[m^2/s]	Thermal diffusivity
β	[1/K]	Volume expansion coefficient
φ	[W/m^2]	Heat flux density
ρ	[kg/m^3]	Density
ν	[m^2/s]	Kinematic viscosity
μ	[Pa.s]	Dynamic viscosity

Subscripts

0	Reference value

ANALYSIS AND MATHEMATICAL FORMULATION

The physical problem considered here is the natural convection in an asymmetrically heated vertical channel with two adiabatic square ribs as schematically presented in Figure 1. The vertical channel is made up of two parallel plates of height 2L and the left plate has a portion (L/2≤y≤3L/2) heated with a uniform heat flux density and of aspect ratio R_f is given as in Eq. (1). The remaining parts from the left wall and the whole right wall are adiabatic.

$$R_f = \frac{L}{b} \tag{1}$$

The two adiabatic ribs are located downstream of the channel at the end of the heated zone. In this study water is used as working fluid to neglect the effect of heat transfer by radiation and to only consider free convection effects. The fluid is Newtonian, incompressible and the physical properties (ρ, Cp, k, μ) of the fluid are temperature dependent. The viscous dissipation in the energy equation is neglected. The channel is initially immersed in a tank filled with water whose temperature is T_0 = 289K.

The governing equations are:

- Continuity equation:

$$\frac{\partial \rho u}{\partial x} + \frac{\partial \rho v}{\partial y} = 0 \tag{2}$$

- Momentum equations:

$$\left(\frac{\partial u}{\partial t} + u\frac{\partial \rho u}{\partial x} + v\frac{\partial \rho u}{\partial y}\right) = -\frac{\partial P}{\partial x} + \frac{\partial}{\partial x}\left(\mu\frac{\partial u}{\partial x}\right) + \frac{\partial}{\partial y}\left(\mu\frac{\partial u}{\partial y}\right) \tag{3}$$

$$\left(\frac{\partial v}{\partial t} + u\frac{\partial \rho v}{\partial x} + v\frac{\partial \rho v}{\partial y}\right) = -\frac{\partial P}{\partial y} + \frac{\partial}{\partial x}\left(\mu\frac{\partial v}{\partial x}\right) + \frac{\partial}{\partial y}\left(\mu\frac{\partial v}{\partial y}\right) - \rho g \tag{4}$$

- Energy equation:

$$\rho C_p\left(\frac{\partial T}{\partial t} + u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y}\right) = \frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right) \tag{5}$$

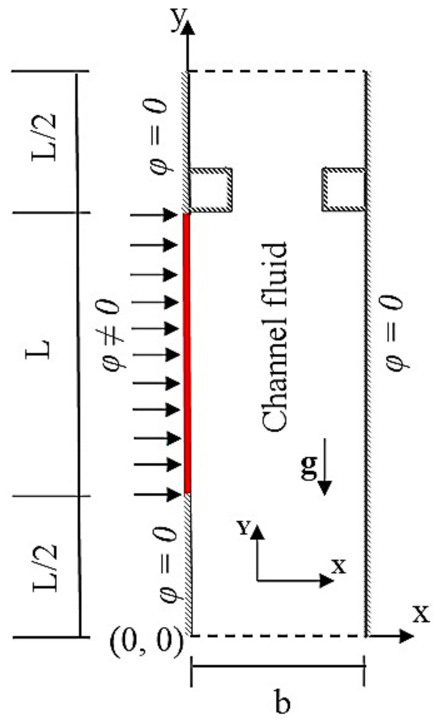

Figure 1 Two-dimensional representation of the channel

The velocity and temperature boundary conditions used in this problem are as follows

On the right wall x=b and 0≤y≤2L: $u = v = 0; \partial T/\partial Y = 0$

On the left wall

- x = 0; 0 ≤ y ≤ L/2 and 3L/2 ≤ y ≤2L: $u = v = 0; \varphi = 0$
- x = 0; L/2≤ y ≤ 3L /2 : $u = v = 0; \varphi > 0$

The boundary conditions at the inlet and outlet of the channel are *a priori* unknown since the driving flow is located within the computation domain (interactions between the channel and its external environment).

Once the flow and thermal fields are computed, the surface averaged Nusselt number was calculated at the hot wall using the formula:

$$\overline{Nu} = \frac{\bar{h}L}{k} \tag{6}$$

While, $\bar{h} = \frac{1}{L}\int_{L/2}^{3L/2} \frac{\varphi}{(T_w - T_0)} dy$ (7)

NUMERICAL FORMULATION AND VALIDATION

The governing equations are solved using the finite-volume method in a staggered grid system. Numerical simulations have been performed with the commercial Ansys Fluent® CFD code. The calculation domain is covered with a uniform grid with refined mesh in the vicinity of the walls. The resulting sets of algebraic equations for each variable after discretization were solved using the line by line procedure, which is the combination of the tri-diagonal matrix algorithm (TDMA) and the Gauss-Seidel iterative technique. During the iterative process, residues for each algebraic equation are monitored and the solution is considered to be converged when the normalized residual of each algebraic equation is less than a prescribed value of 10^{-6} for all the governing equations.

For the aim of validating our numerical results, the present temperature differences in the vicinity of the heated wall are compared with previous experimental data [9]. The same case as in [9] with the same boundary conditions is reproduced. As shown in Figure 2, a very good agreement is found with a maximum difference equal to 9.47%. This gives more consistency to our numerical results.

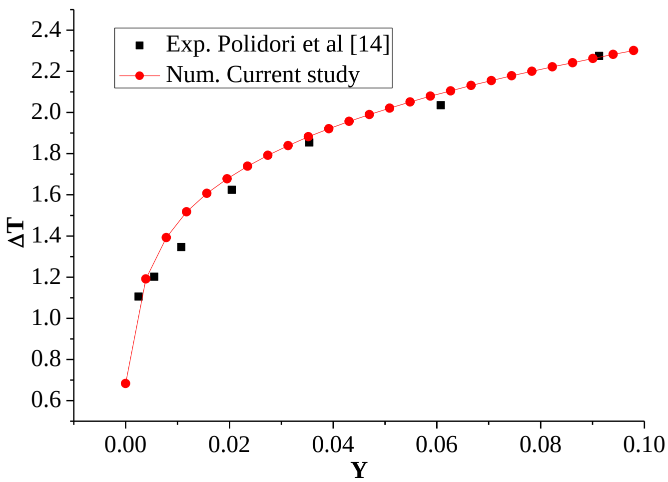

Figure 2 Comparison of the current study results with those experimental [9]

RESULTS AND DISCUSSION

Heat transfer and flow dynamics in asymmetrically heated vertical rib-roughened channel of two adiabatic ribs attached symmetrically to each wall are investigated numerically by carrying out simulation with water as working fluid, thus allowing neglecting radiation effects. The present parametric study has been carried out over reasonable ranges of ribs aspect ratios of $0 \leq Rs \leq 6b/18$ for a fixed modified Rayleigh number $Ra^* = 2.86\times 10^6$ and for a fixed aspect ratio of the heated part $R_f = 5.2$. The modified Rayleigh number based on the heat flux density (φ) and the thickness of the channel (b) also takes into account the aspect ratio of the heated part R_f through Eqs. (1) and (8). Concerning the value of the heated part aspect ratios R_f, it is within the range of those found for horizontally divided double-skin facades of high-rise buildings [3].

$$Ra^* = \frac{g\beta\phi b^4}{k\nu^2}\frac{b}{L}Pr \qquad (8)$$

1. STREAMLINES AND MASS FLOW RATE

Figure 3 represents the effect of the two square adiabatic ribs size on the flow structure in the studied channel. Generally, it can be seen that the flow is boundary layer type for all cases and develops along the left heated wall. Moreover, the existence of pocket-like streamlines (reversed flow) near the adiabatic right wall in the exit of the channel are also observed. The formation of the pocket-like shape streamlines in the channel outlet is due to cooler ambient fluid sucking through the channel outlet due to not enough feeding channel inlet. The penetration depth of the reversed flow is almost until more than the half of the channel for Rs= 0 (no ribs fig. 3a) and it slightly diminishes with the increase in the ribs sizes (for Rs = b/18 and 2b/18: Figures. 3b, 3c). The shape of the streamlines is almost similar except for Rs= b/18 (Figure 3b) where a small secondary vortex is observed beneath the right rib. For Rs = 3b/18 (Figure. 3c), the recirculation zone is sheared by the ribs to the half size for Rs= 0 (no ribs) located in the upper adiabatic extension of the channel above the ribs. The important observation is that at this Rs value (= 3b/18) there is an important quantity of fluid exiting the channel about more than one time and half compared to that for the case with no ribs.

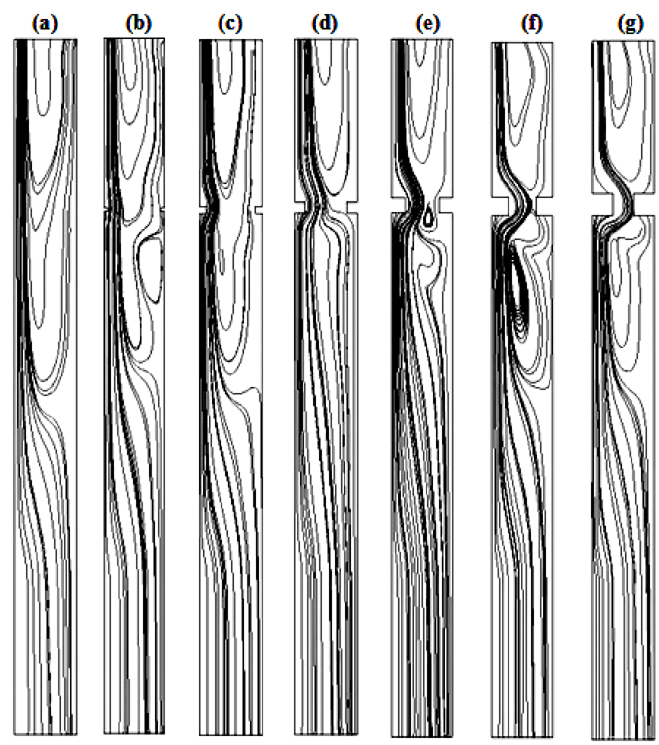

Figure 3 Streamlines pattern for different ribs sizes Rs: (a) Rs = 0 no ribs; (b) Rs = b/18; (c) Rs = 2b/18; (d) Rs = 3b/18; (e) Rs = 4b/18; (f) Rs = 5b/18 and (g) Rs = 6b/18.

When the ribs size increases $Rs \geq 4b/18$, the formation of two small vortices below the ribs in the centreline of the channel is noticed.

Figure 4 shows the variation of the mass flow rate as a function of the ribs sizes ($0 \leq Rs \leq 6b/18$). It can be seen that the variation of the mass flow rate within the channel shows a peak corresponding to ribs size equal to 3b/18 and leading to a maximal flow rate enhancement of 67.25% compared to the reference case. Then it decreases gradually with the Rs augmentation. For the case Rs= 3b/18 and as mentioned before, the penetration of recirculation zone is clipped by the two ribs to about the half of its original depth for Rs = 0 and it is located above the ribs (see Figure 3d) offering a large passage for the feeding fluid.

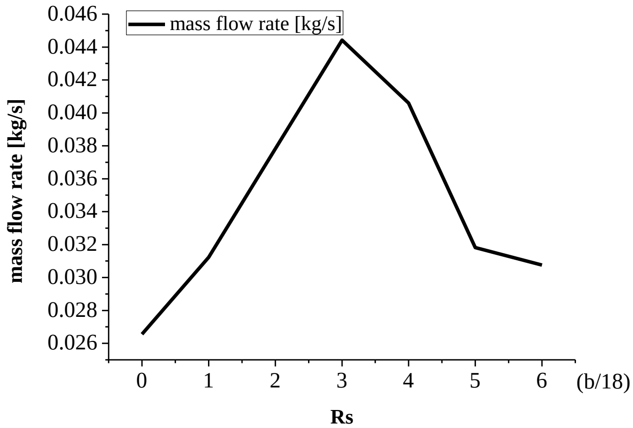

Figure 4 Mass flow rate variation as a function of ribs size

Then, the increase of the ribs sizes diminishes the passage of the fluid and the appearance of the two small vortices below the ribs blocks the fluid passing through the mini channel leading to a decrease in the mass flow rate.

2. AVERAGE NUSSELT NUMBER AND TEMPERATURE DISTRIBUTION

The effect of ribs sizes on the heat transfer within the studied channel is reported as the temperature difference and the average Nusselt number at the heated wall for $Ra^* = 2.86\times10^6$ and R_f = 5.2. Figure 5 displays the comparison of temperature difference profiles along the heated wall for all studied cases (Rs = 0 to 6b/18). Generally, the same tendency was seen that the temperatures profiles at the heated wall increase with the vertical direction. The minimum wall temperatures are found for the case Rs = 3b/18 which mean the best extraction of heat from the heated wall**.**

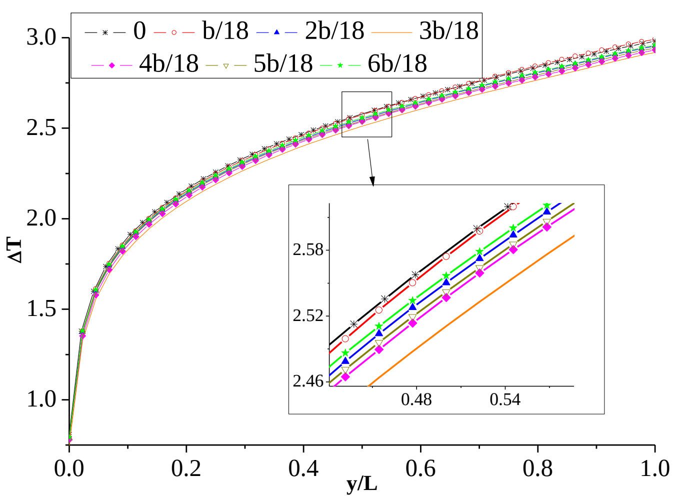

Figure 5 Temperature profiles along the vertical heated wall for all studied ribs sizes

This result indicates that the maximum heat transfer rate characterized by heat transfer coefficient $h = \varphi/(T_w\text{-}T_0)$ is found for Rs= 3b/18 as confirmed by the average Nusselt number plots for all ribs sizes presented in Figure 6. These plots indicate that the heat transfer rate increases with increasing ribs sizes until it reaches its maximal value for Rs= 3b/18 then it decreases with Rs augmentation. It is recalled that at this Rs value the temperatures of the heated wall reach their minimum values while a maximum flow rate is observed (i.e. maximum quantity of the cold fluid coming from the channel inlet due to the clipping of the reversed zone by the two ribs).

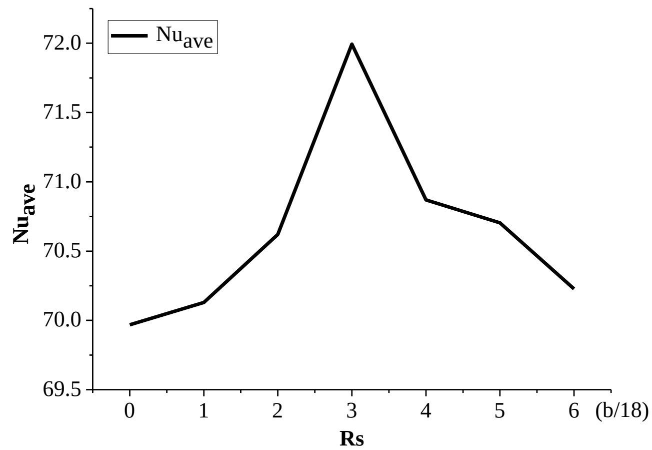

Figure 6 the variation of the average Nusselt number along the heated wall for the various ribs sizes Rs

CONCLUSION

The effect of adiabatic square ribs on the heat transfer and fluid flow in an asymmetrically heated channel have been numerically analyzed for modified Rayleigh number equal to 2.86×10^6, channel aspect ratio equal to 5.2 and for seven sizes of the square ribs. Overall, the heat transfer and flow characteristics are strongly affected by the ribs sizes.

The subsequent is a concise summary of the core results of this study

- A boundary layer type flow in the vicinity of the heated wall is noticed for all ribs sizes.
- According to the size of ribs, the heat transfer rate and the mass flow rate can be enhanced or reduced.
- The size of ribs Rs= 3b/18 is the optimal size which offers more improvement of heat transfer and mass flow rates compared with the case with no ribs. The improvement of the mass flow rate and heat transfer is due to the shearing of the recirculation zone at the channel outlet, which offers more section passage to the cold fluid coming from channel inlet.
- The recirculation zone was always noticed at the channel outlet whatever the ribs size. On the other hand, the increase in the size of this zone diminishes considerably the induced flow in the channel. The recirculation length gradually decreases with increasing in ribs size showing its minimal

length at Rs = 3b/18 then it increases with the increase of ribs sizes showing two small vortices in the vicinity of the two ribs.

In order to access more details on the way convective heat transfer occurs in roughened vertical channel, further experiments will be conducted to get the expected effect of the ribs location with respect to the channel length. Different locations will be studied, namely in the upper, middle and lower part of the heating area.

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