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Improving the Efficiency of Wind Power System by Using Natural Convection Flows

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Abstract: In this paper a numerical study of natural convection in a two dimensional convergent channel, with or without rectangular block, is carried out. The block is placed at the channel outlet and its thermal conductivity is set equal to that of air. One of channel planes is heated at constant temperature T_H. The other one is maintained cold at $T_C < T_H$. The governing equations are solved using a finite volume method and the SIMLEC algorithm for the velocity-pressure coupling is used. Special emphasis is given to detail the effect of the block size and *Rayleigh* number on the dynamics of velocity, heat transfer and the debit generated by natural convection. Results are given for the following control parameters, $10^4 \leq Ra \leq 10^6$, Pr=0.71. The inlet and outlet opening diameters are, respectively, $C_1 = c_1/h$= 0.2 and $C_2 = c_2/h$ =0.1. Three values of the block height are considered: $B = b/h$=0.1, 0.14 and 0.2. These results show that the heat transfer and the mass flow rate variations with Ra are similar to those occurring in the case of the vertical isothermal parallel planes. The effect of the block height on the flow structure and heat transfer is negligible.

Keywords: Convergent channel/ Heat transfer/ wind system/ Numerical study/ block effect

Nomenclature

b	block height
B	dimensionless block height (b/h)
c	opening diameter
C	dimensionless opening diameter (c/h)
h	channel height
H	length of the channel inclined planes [=h/cos(γ)]

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γ	inclination angle
M	mass flow rate
n	normal coordinate
Nu	mean *Nusselt* number (eq.6)
Pr	*Prandtl* number ($Pr=\nu/\alpha$)
Ra	*Rayleigh* number ($Ra = \frac{g.\beta.\Delta T.c_1^3}{\alpha.\nu}$)
T	temperature of fluid
t	time
τ	dimensionless time ($t.\alpha/h^2$)
θ	dimensionless temperature of fluid [= $(T-T_c)/(T_H-T_C)$
u,v	velocities in x and y directions
U,V	dimensionless velocities in x and y directions
x,y	Cartesian coordinates
X,Y	dimensionless Cartesian coordinates [=$(x,y)/h$]
α	thermal diffusivity (m^2/s)
β	volumetric coefficient of thermal expansion (K^{-1})
λ	thermal conductivity of fluid ($W.K^{-1}m^{-1}$)
λ_b	thermal conductivity of the block ($W.K^{-1}m^{-1}$)
ν	kinematic viscosity of fluid (m^2/s fluid density (kg/m^3)
ψ	stream function
Ψ	dimensionless stream function (=ψ/α)

Subscripts

C	cold wall
H	heated wall
min	minimum
max	maximum

1 Introduction

Natural convection processes are widely used in thermal control of many systems because of its cheapness easy maintenance and reliability [Aouachria et al. 2009]. The design of natural convection thermal control systems by using simple relations is certainly appealing. Particular interest has been devoted to the channel configuration and several contributions have dealt with this geometry, as recently reviewed in [Auletta et al. 2003].

An interesting problem is the heat transfer in a vertical channel with two symmetric or asymmetric heated flat plates. In particular, when the channel is convergent, the determination of the thermal performance of these configurations is rather difficult, due to the large number of thermal and geometric variables [Sparrow et al. 1988, Said. 1996].

The first numerical and experimental study of natural convection in water in a convergent vertical channel was carried out by [Sparrow et al. 1988]. Natural convection in a convergent channel with finite-thickness principal walls at uniform heat flux was investigated numerically. Heat conduction in the walls and the effects of walls emissivity were taken into account. Laminar two-dimensional steady-state conditions were assumed. Results in terms of wall dimensionless temperature profiles as a function of convergence angle, channel spacing, and heat flux were given for various values of wall emissivity [Bianco et al. 2006].

A numerical study of natural convective flows, mainly for high *Rayleigh* numbers, in a sloped converging channel, for different inclination and convergence angles has been carried out by [Kaiser et al. 2008], taking into account the lacks of the literature on some aspects of this configuration. Two-dimensional, laminar, transitional and turbulent simulations were obtained by solving the fully elliptic governing equations using two different general-purpose codes: Fluent and Phoenics. Their Results show that for sloped and convergent channels, the average Nusselt number can be fitted to those obtained for vertical channels by modifying the *Rayleigh* number. Generalized correlation for *Nusselt* number has been reported. In view of these studies it can be observed that results of heat transfer in convergent channels with obstacle have not been performed yet, at our knowledge. Existing studies, in the majority of cases, consider asymmetrical heating and do not assess in a systematic manner the influence of geometric parameters such as the converging angle, the block size and the aspect ratio of the channel.

In this paper, our aim is to study the block size and *Rayleigh* number effect on the flow structure, heat transfer and the kinetic energy of wind at the channel outlet. In particular, our objective is to analyse the possibility to improve the wind power system performance by using natural convection flow and simulating a wind system support as a convergent channel heated by solar energy. The originality of our work lies in the combination of wind and solar systems for enhancing wind speed at the channel outlet.

2 Physical problem and governing equations

The studied configuration is shown in Figure 1. Note that, the convergence angle is constant in this study. The channel left wall is heated at constant temperature

T_H and the right one is maintained cold at $T_C < T_H$. A heat conductor rectangular block is introduced at the channel outlet.

The flow is considered steady, laminar incompressible and the Boussinesq approximation has been applied. The thermo-physical quantities are assumed to be constant except for the density in the buoyancy force. With these assumptions, the dimensionless governing equations can be written as:

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

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

$$\frac{\partial V}{\partial t}+U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\Pr(\frac{\partial^2 V}{\partial Y^2}+\frac{\partial^2 V}{\partial Y^2})+\Pr.Ra.\theta \tag{3}$$

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

Referring to fig. 1, the dimensionless variables are:

$$X=\frac{x}{h},\quad Y=\frac{y}{h},\quad U=\frac{u.h}{\alpha},\quad V=\frac{v.h}{\alpha},\quad \theta=\frac{T-T_C}{T_H-T_C},\quad P=\frac{(p+\rho.g.y)}{\rho.\alpha^2}$$

$Ra=\frac{g.\beta.\Delta T.c_1^3}{\alpha.\nu}, \Pr=\frac{\nu}{\alpha}$ with $(\Delta T = T_H - T_C)$

Dirichlet conditions have been assumed for most of the quantities at the channel inlet. We consider the inlet velocity and temperature profiles uniforms. This selected set of boundary conditions considers that the incompressibility constraint is extended up to the border. We finally close the Boussinesq system by imposing Bernoulli conditions on pressure at the channel inlet. Hence, the boundary conditions associated to the system of the equations (1-4) are detailed as bellow:

θ=1; $U=V$=0 (on the channel plane on the left)

θ=0; $U=V$=0 (on the channel plane on the right)

θ=U=0, $V=\frac{M}{c_1}$, $P=-\frac{M^2}{2}$ (at the inlet opening)

The mass flow rate M, is calculated at the channel outlet as:

$$M=\int_{0.25}^{0.75} V dx|_{y=H} \tag{5}$$

At the outlet opening, $\frac{\partial U}{\partial y}=0$ while$V$and$T$ are extrapolated by adopting similar processes as shown in references [Tomimura et al. 1988, El Alami et al. 2008,

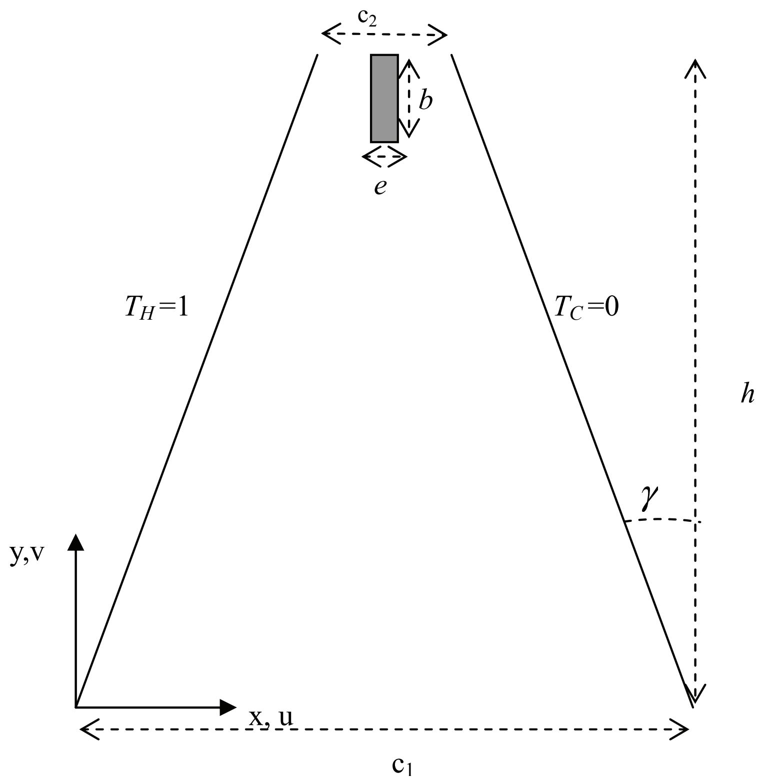


Figure 1: Convergent channel with a block at the outlet

Tmartnhad et al. 2008] (their second spatial derivative terms in the vertical direction are equal to zero).

For reasonably high values of *Ra* ($Ra \geq 5x10^3$), any region outside the channel can be neglected: the upstream thermal diffusion through the inlet is considered negligible.

The local *Nusselt* number over the surface blocks (active walls) is calculated as:

$$Nu = \frac{\partial T}{\partial n}|_{heated \quad wall} \tag{6}$$

The mean and normalised *Nusselt* number over the heated wall is:

$$\overline{N}u = \frac{1}{H}\int_0^H \frac{\partial T}{\partial x}|_{y=y_1}\, dy \tag{7}$$

y_1 is the equation representing the inclined heated wall and H is the length of the channel inclined plane defined as : $H = \frac{h}{\cos\alpha}$

3 Numerical method

The results presented in this work have been obtained by using numerical procedure based on a finite volume technique [Achoubir et al. 2008, Semma et al. 2010]. The governing equations were solved considering the boussinesq approximation. Two dimensional flows are considered in this study and the velocity – pressure coupling is solved by using SIMPLEC algorithm [Van doormaal et al. 1984].

As a result of a grid independence study, a grid size of 60x300 was found to model accurately the flow fields described in the corresponding results. Time steps considered are ranging between 10^{-5} and 10^{-4}.

The accuracy of the numerical model was verified by comparing our results with those obtained by [De Val Devis et al. 1984, Le Queré et al. 1985] for natural convection in differential heated cavity, table 1, and then with the results obtained by [Kalache et al. 1985] in a trapezoidal cavity, table 2.

Finally, we have confronted our results to those proposed by [Derayaud and Fichera. 2002] in a vertical channel with two ribs symmetrically placed on the channel walls, table 3. Good agreement was obtained in Ψ_{max} and Nu terms. When a steady – state is reached, all the energy furnished by the hot wall to the fluid must leave the channel through the cold surface and trough the outlet opening. This energy balance was verified by less than 3% in all cases considered here.

Table 1: Comparison of our results and those of [De Val Devis et al. 1984, Le Queré et al. 1985]

Ra	[De Val Devis et al. 1984]	[Le Queré et al. 1985]	Present study	Maximum deviation
10^4	Ψ_{max}= 5.098	——	Ψ_{max}=5.035	1.2%
10^5	Ψ_{max}=9.667	——	Ψ_{max}=9.725	0.6%
10^6	Ψ_{max}=17.113	16.8110	Ψ_{max}=17.152	0.2%
10^7	——	30.170	Ψ_{max}=30.077	0.3%

4 Results and discussion

In this paper, the analysis is focused on heat transfer rate across the hot wall, flow and thermal fields for range of *Rayleigh* number: $5\text{x}10^3 \leq Ra \leq 10^6$, and other parameters of the problem (h=5, C_1=0.20, C_2=0.1 and Pr=0.71). *Rayleigh* number

Table 2: Comparison of our results and those of [Kalache et al. 1985]

Gr	Present study	[Kalache et al. 1985]	Maximum deviation
2.5×10^3	Ψ_{max}=5.42	Ψ_{max} =5.37	2.4 %
5×10^3	Ψ_{max} =9.74	Ψ_{max} =9.77	4.1 %
10^4	Ψ_{max} =15.43	Ψ_{max} =15.41	0.2%

Table 3: Comparison of our results and those of [Derayaud and Fichera. 2002]

Ra=10^5 (A=5)	[Derayaud and Fichera. 2002]	Present study	Maximum deviation
Ψ_{max}	151.51	152.85	0.9%
M	148.27	151.72	2.2%

values range was chosen in favour of natural convection role. Results are presented in the channel without and with the block at the outlet. The thermal conductivity of the block is set equal to that of air. Streamlines and isotherms are not presented for the intermediate value *B*=0.14.

4.1 Flow structure and Thermal field

We present isotherms and streamlines, in the case of convergent channel without block, for *Ra* = 10^4, 10^5, and 6x10^5, as shown in figures 2-a, 2-b and 2-c, respectively. We can notice in all these figures that the flow structure, represented by streamlines, is not complex and made only by open lines. As expected, two different trends have been obtained: fully developed regime for low *Ra* values ($Ra<10^5$), and boundary layer regime, for high *Ra*. Hence, at low *Ra*, the channel is full by the open lines, fig. 2-a; the flow is a *thermosyphon* kind. When *Ra* exceeds 10^5, the boundary layer regime is developed along the hot wall, figs. 2-b and 2-c. Isotherms are too tight near this active wall, showing a good heat exchange.

In the case of the channel with block, the effect of *Rayleigh* number, on the flow structure and on the heat transfer has been analyzed for various values of the block height. The flow structure changes when the block size increases.

Thus, for *B* = 0.1, a recirculating cell appears just below the block and streamlines are deformed in this zone, fig.3-a, for *Ra*=10^4. The recirculating cell constrains the fresh air jet to pass close the active wall. It size decreases with *Rayleigh* number as shown in fig.3-b for *Ra*=10^5. Corresponding isotherms show a good heat exchange a long the lower part of the hot wall. They are distorted around the block and we can notify that the major part of the open lines pass between the hot wall and the block. When *Ra* exceeds 10^5, boundary layer flow is created along the upper part

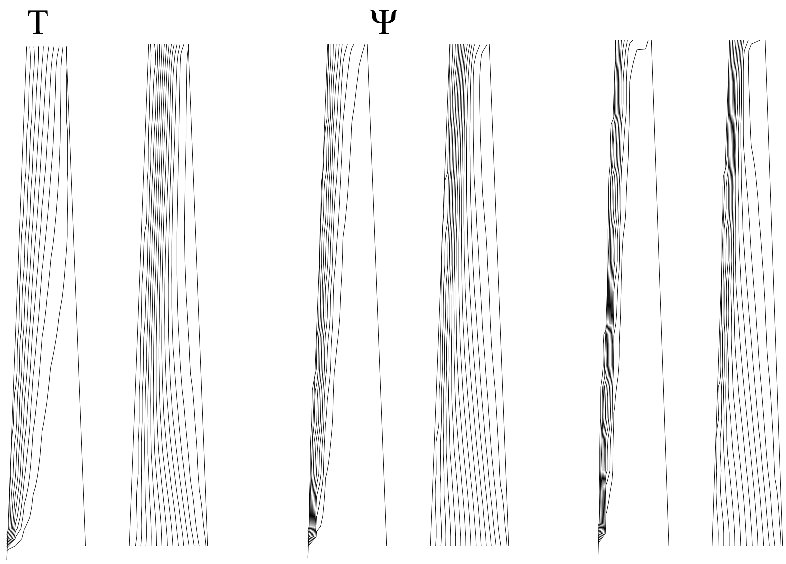


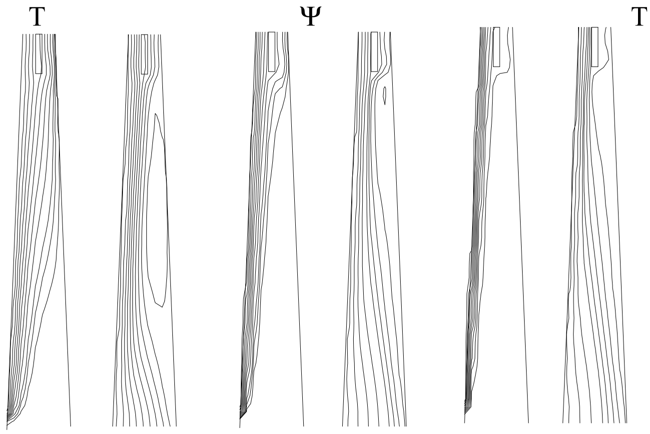
Figure 2: Isotherms and streamlines in the channel without block

of the hot wall and isotherms are too tight along this wall.

When increasing the block height, the mainly observation is that the recirculating cell size becomes more and more important and open lines are much distorted just below the block as we can see for example in figs.3-b, and 4-b, respectively for B=0.1 and 0.2. Corresponding isotherms show the development of thermal boundary layer and heat exchange is slightly enhanced referring to the last case (B=0.1).

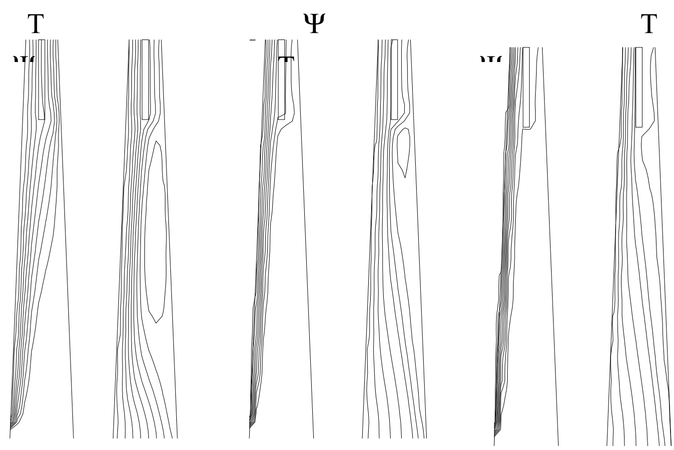
Generally, the block height variation effect on both heat exchange and flow structure is limited. In our sense, that because of the block size variation is made in the path of the flow, especially since the block is considered conductor of heat.

The mainly objective of this work is to evaluate the cinematic energy quantity which can be transferred by the flow to the turbine modulated by the block, at the channel outlet. Hence, we calculate the vertical outlet velocity profile in the cases of channel with and without block for different values of *Rayleigh* number, as shown in figs.5-a,b,c,d, respectively for Ra=10^4, 10^5 and 6x10^5. We can notify that the block existence modifies strongly the outlet velocity profile at low Ra number as shown if fig.5-a, for Ra=10^4. When Ra increases, the block effect on the velocity profile becomes more and more limited, figs.5-b and 5-c. This because increasing Ra leads to the development of the boundary layer regime along the upper



a) Ra=10^4 b) Ra=10^5 c) Ra=$6x10^5$

Figure 3: Isotherms and streamlines in the channel with the block, B=0.1



a) Ra=10^4 b) Ra=10^5 c) Ra=$6x10^5$

Figure 4: Isotherms and streamlines in the channel with block, B=0.2

part of the channel and so the block position is out of the flow path.

To maintain the block effect on the outlet velocity profile important, we must move the block toward the hot wall as well as *Ra* increases.

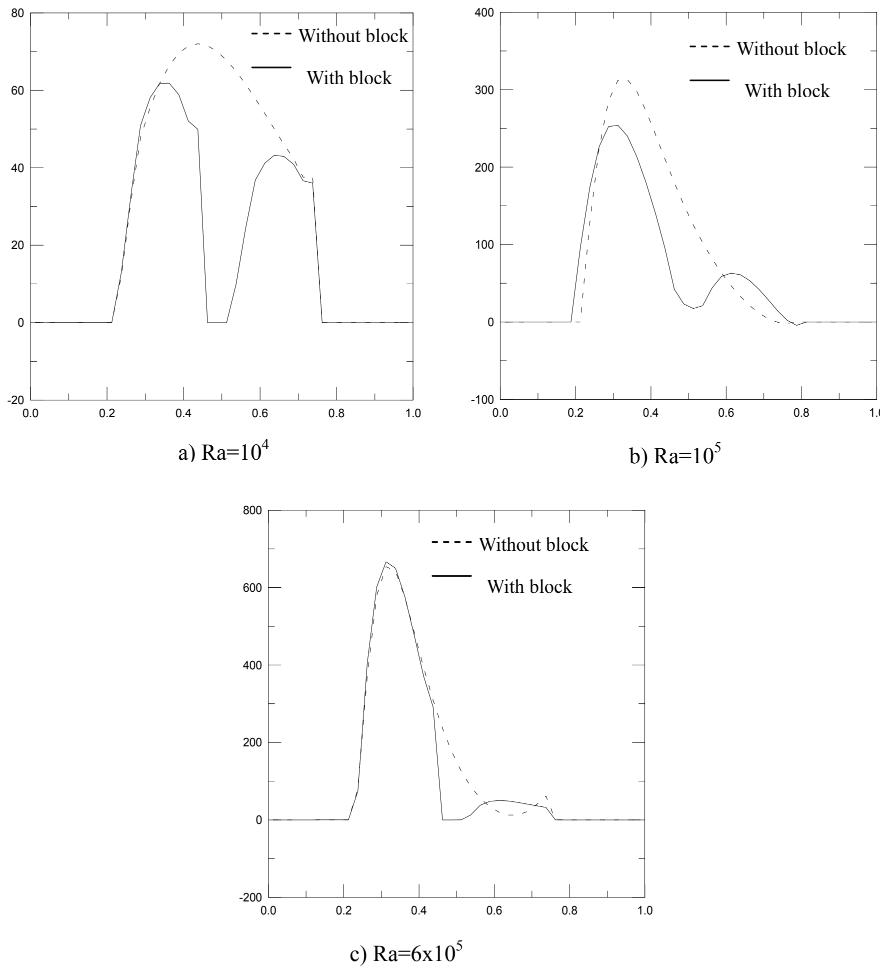


Figure 5: Outlet velocity profile for different values of *Ra*

In order to evaluate the wind energy, may be transmitted to the turbine (represented by the block), we calculated the maximum difference ΔV_{max} between the flow speed leaving the channel with and without block for different values of *Rayleigh* number. The evolution of ΔV_{max} depending on *Ra* is shown in fig.6. We notice that ΔV_{max}, generally, varies between 100 and 400 in the chosen range of *Ra*. It increases with the *Rayleigh* number until it reaches a maximum and then fall slightly in the area of high *Ra*. For both heights of the block, the maximum of ΔV_{max} is obtained around *Ra* = 3×10^5. The two curves are indistinguishable between *Ra* = 10^4 and 10^5, while an important gap between them is observed around *Ra*=10^5. We explain the existence of ΔV_{max} maximum by the fact that increasing *Ra*, the vertical flow velocity increases, which explains the increase of ΔV_{max}. Contrarily, the thickness of the boundary layer decreases because of the chimney effect development. Consequently, the block is located outside of the dynamic boundary layer, so it has no effect on the flow.

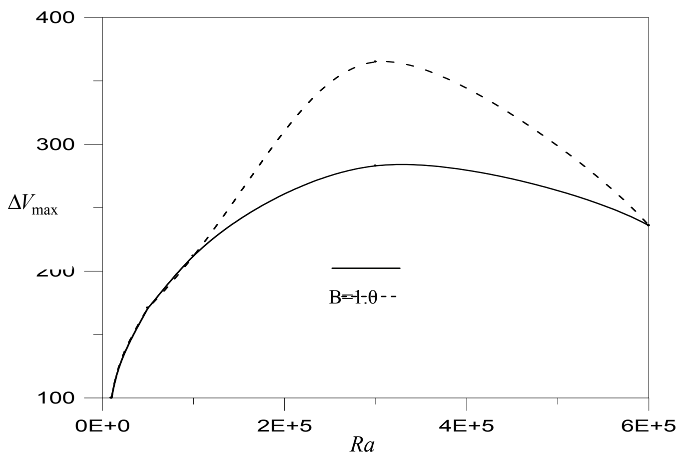


Figure 6: ΔV_{max} variation with *Rayleigh* number for different values of the block height

4.2 Heat transfer and mass flow rate

The heat transfer in the channel is calculated in term of the average *Nusselt* number, using the equation (5). Its variation according to the *Rayleigh* number (Ra) [Ben-Arous et al. 2008] is presented in fig. 7, for different values of the bloc height B. We note that, the *Nusselt* number increases linearly (in log-log) with *Ra*. It should be noted that there is no gap between the *Nu* curves and so, the bloc height variation is without any effect on the heat exchange throw the channel.

In the range of *Rayleigh* number values ($10^4 \leq Ra \leq 8 \times 10^5$) the numerical values of *Nu* are given by the correlation with *Ra* as:

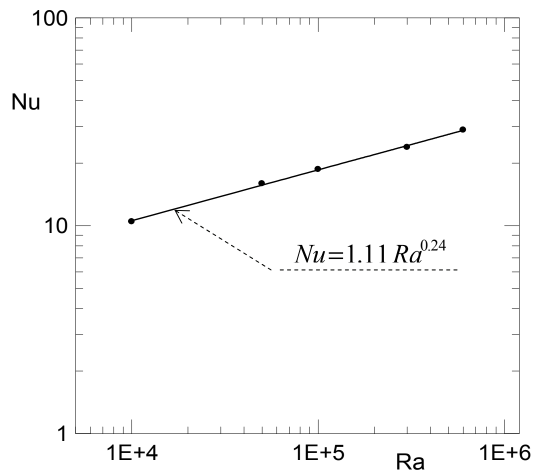


Figure 7: *Nusselt* variation with *Rayleigh* number for B=0.10 ; 0.14 ; 0.20

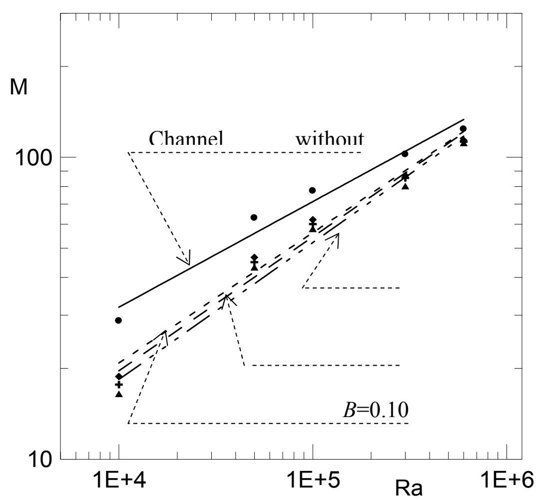


Figure 8: Mass flow rate variation with *Rayleigh* number for *B*=0.10 ; 0.14 ; 0.20

Nu=1.11 $Ra^{0.24}$ (5) which can be rewritten in the same meaner of those obtained in the literature as:

Nu=1.10 $Ra^{1/4}$ (6) with maximum deviation less than 3%. There exist some similar relations in the literature in the cases of smoothed vertical channel (*Nu*=0.619 $Ra^{1/4}$ [Sparrow and Azevedo. 1985])

The other outcome of the problem is the rate of induced mass flow M. In fig. 8, we present M variation with Ra for different values of B, with and without block. We note, in this figure, there is a marked difference between the mass flow rate curve in the channel without obstacle and those obtained for different heights thereof. This gap is progressively reduced as Ra increases. This is explained again by the narrowing of the dynamic boundary layer described above. The mass flow rate correlations with Ra are shown below:

$M = 1.272\,Ra^{0.35}$without block

$M = 0.33\,Ra^{0.44}$B=0.10

$M = 0.28\,Ra^{0.45}$B=0.14

$M = 0.39\ Ra^{0.43}$B=0.20

The maximum deviation is less than 3% for all these cases.

5 Conclusion

A numerical investigation has been conducted to evaluate heat and mass flow rate induced by natural convection in a finite vertical and convergent channel with heated rectangular block at the outlet. Cases for intermediate values of *Rayleigh* number and for different values of the block height have been considered. Most of such cases correspond to isothermal asymmetric heating conditions. The case of convergent channel without the block has been also considered (for comparison).

The results show that there is a slight modification in the flow structure and thermal field due to the block existence. The effect of the block height variation on the flow structure and on heat exchange is relatively limited. However, there is a significant gap between the curves representing the mass flow in cases with and without block. This variation, due to the fall of velocity in the vicinity of the block, is evaluated in the form of kinetic energy eventually transferred to the block (representing the turbine of a wind system).

This allowed us to conclude that a significant improvement of performance of wind power systems is possible using this method based on solar radiation collector (represented by the convergent channel considered here).

References

Achoubir K., Bennacer R., Cheddadi A., El gnaoui M., Semma E. (2008): Numerical Study of Thermosolutal Convection in Enclosures Used for Directional Solidification (Bridgman Cavity), *Fluid Dyn. Mater. Process.*, vol. 4 no. 3, pp. 199-210.

Aouachria Z. (2009): Heat and Mass Transfer Along of a Vertical Wall by Natural convection in Porous media, *Fluid Dyn. Mater. Process.*, vol. 5 no. 2, pp. 137-148.

Auletta A.,Manca O.,Musto M., Nardini S. (2003): Thermal design of symmetrically and asymmetrically heated channel-chimney systems in natural convection, *Applied Thermal Engineering* vol. 23, pp. 605–621.

Ben-Arous A. M., Busedra A. A. (2008): Mixed Convection in Horizontal Internally Finned Semcircular Ducts, *Fluid Dyn. Mater. Process.*, vol. 4 no. 4, pp. 255-262.

Bianco N., Langellotto L., Manca O., Naso V. (2006): Numerical analysis of radiative effects on natural convection in vertical convergent and symmetrically heated channels. *Numerical Heat Transfer*, Part A, vol. 49, pp. 369–391.

Desrayaud G, Fichera A. (2002): Laminar natural convection in a vertical isothermal channel with symmetric surface mounted rectangular ribs. *Int J Heat Fluid Flow*. vol. 23, pp.519–29.

De Vahl Davis G. (1983): Natural convection of air in a square cavity: a bench mark numerical solution. *Int J Numer Meth Fluid*, Vol.3, pp. 249–64.

El Alami M., Semma. E. A., Najam M, Boutarfa R. (2008): Numerical Study of Convective Heat Transfer ina Horizontal Channel. *Fluid Dyn. Mater. Process.*, Vol.151, no.1, pp.1-9.

Kaiser A. S., Zamora B., Viedma A. (2008): Numerical correlation for natural convective flows in isothermal heated, inclined and convergent channels, for high Rayleigh numbers. *Comput Fluids*. Vol.38, No.1, pp 1-15.

Kalache D., Penot F., Le Quéré F. (1985): Numerical investigation of the validity of the two-dimensional assumption in the computation of natural convection within a trapezoidal cavity Num. Methods in laminar and turbulent flow, 4th. Int. Conf. SWANSEA, UK, pp. 829-840.

Le Queré P., Alziary De Roquefort T. (1985): Computation of natural convection in twodimensional cavities with chebyshev polynomials. *J Comput Phys.*, vol. 57, pp. 210–28.

Said S. A. (1996): Investigation of natural convection in convergent vertical channels, *International Journal of Energy Research* vol. 20, pp. 559–567.

Semma E. A ., El Ganaoui M ., Timchenko V., leonardi E. (2010): Thermal Modulation effects on Thermosolutal Convection in a Vertical Bridgman Cavity, *Fluid Dyn. Mater. Process.*, Vol.6(3), pp. 233-250.

Sparrow E. M., Azevedo F. A. (1985): Vertical-channel Natural Convection spanning between the fully developed limit and the single-plate boundary-layer limit. *Int. J. Heat Mass Transfert*, Vol.28, No. 10, pp. 1847-1857.

Sparrow E. M.,Ruiz R., Azevedo L. F. A. (1988): Experiments and numerical investigation of natural convection in convergent vertical channels, *International Journal of Heat and Mass Transfer* vol. 31, pp. 907–915.

Tmartnhad I., El Alami M., Najam M., Oubarra A. (2008): Numerical investigation on mixed convection flow in a trapezoidal cavity heated from Below. *Journal Energy conversion and Management*, vol. 49 no. 11, pp. 3205-3210.

Tomimura T., Fujii M. (1988): Laminar mixed heat transfer between parallel plates with localized heat sources, In proc. Int. Symp. On Cooling Technology for Electronic Equipment, Honululu, pp. 233-247.

Van Doormaal J. P., Raithby G. D. (1984): Enhancements of the SIMPLE method for predicting incompressible fluid flows. *Numerical Heat Transfer*, vol. 7, pp. 147-163.