# EFFECTS STUDY OF SOLID PROTUBERANCES PROPERTIES ON THE TURBULENT NATURAL CONVECTION PERFORMANCE INSIDE A VERTICAL CHANNEL

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**RESUME**

Dans cette étude, nous présentons des prédictions numériques des effets des propriétés géométriques, physiques et thermiques des protubérances, telles que la conductivité thermique des protubérances solides, les rapports d'aspect et les intensités du flux de chaleur exercées sur les faces externes des protubérances, sur la convection naturelle turbulente dans un canal vertical. Les résultats de l'analyse du transfert de chaleur sont obtenus en solutionnant les équations de l'écoulement de chaleur et des fluides en utilisant le modèle de la turbulence $k-\varepsilon$. Le système étudié est bidimensionnel, turbulent et permanant. La procédure numérique étend sur un code numérique qui modélise l'écoulement de chaleur et des fluides. Les résultats sont données en termes des champs et des profils des propriétés locales telles que la température, la vitesse, le coefficient de transfert de chaleur et l'intensité de la turbulence pour différents valeurs du rapport d'aspect des protubérances, la conductivité thermique et le flux de chaleur exercé sur les faces externes des protubérances. Les champs des propriétés d'air physiques et thermiques montrent des comportements différents en fonction de la valeur du rapport d'aspect des protubérances. Un effet positif sur le rendement de la convection naturelle turbulente a été prédit, dans la plage de 0 à 6,5%, par rapport à un canal vertical sans protubérances '. La présence des protubérances c'est un facteur important dans l'amélioration du rendement de la convection naturelle turbulente dans un canal vertical.

**ABSTRACT**

In this study, we report numerical predictions of the protuberances' geometrical, physical and thermal properties effects, such as thermal conductivity of the solid protuberances, aspect ratios of the protuberances and the uniform heat flux intensities, on the turbulent natural convection inside a vertical channel. Results of heat transfer analysis are obtained by solving the fluid and heat flow equations by using the kinetic energy and the dissipation rate $k-\varepsilon$ turbulence model. The studied system is two-dimensional, turbulent and steady state. The numerical procedure expands an existing computer code on fluid and heat flow modelling. Results are given in terms of fields and profiles of local temperature, velocity, convective heat transfer coefficient and turbulence intensity for various protuberances aspect ratios, protuberances' thermal conductivities and external heat fluxes applied on the external face of the protuberances. The distributions of thermal and physical air properties values show different thermal behaviours for different aspect ratio values. A positive effect on the turbulent natural convection performance, within the range of 0 up to 6.5 %, has been predicted compared to a vertical channel without protuberances'.

**KEYWORDS:** Turbulence, natural convection, vertical channel, solid protuberances.

## 1 INTRODUCTION

Properties of solids depend on their surface more often than their mass properties. Thus, the limits or boundaries surface topography with the external environment has a great importance on the interaction of the surface with the external environment. In order to optimize the functional properties of a fluid flow, it is essential to control the transfer of heat between the one hand the fluid and the wall and the other within the wall. This operation is based on the knowledge of the space-time evolution of the main physical parameters of the flow characteristics (temperature, velocity, humidity…etc).

The considered system is a vertical channel in which the transfer takes place by turbulent natural convection. On one of the plates are fixed to the channel rectangular section protuberances subjected to a uniform and constant density heat flux. These protuberances are formed of a saturated porous material with water or a non-porous material. The work is to characterize the natural convection based on the aspect ratio of the protrusions, the density of the heat flow, the nature of the used material, its water content…etc. Thus, the analysis of the physical parameters effect of the channel-protuberances system on the humidity and the temperature of the air flowing inside the vertical channel can help improving the mastering, for example, of the passive cooling.

The literature review shows that most of the existing works (description not detailed here) on these configurations are concerned with forced or mixed convection [3–14]. However, though the heat removal capacity of natural convection is relatively small compared to that of forced convection, natural convection may play an important role in the cooling of systems generating low powers; it provides low-cost and reliable cooling in such conditions.

Lin and Hsieh [15] studied experimentally natural convection in two vertical connected channels containing asymmetrically discrete heated ribs. The natural convection flow, which circulates from one channel to the other, was found to be laminar in one channel and turbulent in the other. Correlations were proposed for the Nusselt number in both channels.

Numerical predictions of periodically fully developed natural convection in a vertical channel, with surface mounted heat generating blocks, were reported in Kelkar and Choudhury [16]. Their results show that the mass flow rate induced by buoyancy forces increases at a rate less than the square root of the channel length. Natural convection in a horizontal channel heated from below and provided with six equidistant square adiabatic or isothermal bodies, placed in the interior, was studied by Lee et al. [17]. Results of a wide aspect ratio cell with periodic side boundaries and six internal bodies were compared against those obtained in the case of a unit cell containing a single body with either no-slip adiabatic side boundaries or periodic side boundaries and also against those corresponding to pure Rayleigh–Bénard convection. They found that, in the quasi-steady conduction state, the results of the wide aspect ratio case are identical to those of two unit cell cases. However, as convective motion sets in by increasing the Rayleigh number, the differences in the aspect ratio of the enclosure and in the boundary conditions have an effect on the resulting velocity and thermal fields. Hasnaoui et al. [18] studied natural convection inside a horizontal channel with adiabatic rectangular blocks placed on its lower wall and separated by isothermal heating surfaces. The numerical calculations, conducted in a representative module, showed that, depending on the values attributed to the governing geometric and thermophysical parameters, the final flow state achieved may be stationary, periodic or chaotic. Also, it was found that the relative height of the adiabatic blocks has a strong effect on the transition from steady-state solutions to periodic ones and on the destruction of the flow symmetry. A numerical study was performed by Amahmid et al. [19] on a similar geometry in which the rectangular heating blocks were maintained at a constant temperature. The effect of the calculation domain and the geometric and thermophysical parameters on the multiplicity of solutions was examined. It was shown that some solutions, obtained in the extended domain, could not be obtained in the elementary domain, since they do not satisfy the periodic conditions in the latter. The effect of the blocks' width on natural convection in a similar problem was also considered by Bakkas et al. [20]. They reported that this parameter has a significant effect on the flow and heat transfer in the channel.

Kwak and Song [21] reported numerical and experimental results on natural convection around horizontal downward facing plate with rectangular grooved fins with various aspect ratios. The numerical study, conducted in the whole domain, shows the existence of recirculation flows in the grooves, which prevents circulation of the main stream flow between the fins and reduces the heat transfer rate at the inner surface. The chimney effect and natural convection heat transfer in a two-dimensional horizontal channel with isothermally heated blocks and slots were studied numerically by El Alami et al. [22]. They showed that the block height has an important effect on the heat transfer and the flow rate generated by the chimney effect. Correlations evaluating the mean heat transfer and the flow rate were proposed by the authors. In the same configuration El Alami et al. [23] examined the effect of the spacing between the blocks on the cooling process of simulated electronic components. They showed that the flow structure and heat transfer depend significantly on the blocks spacing. Numerical experiments have been carried out by Arquis and Rady [24] to investigate natural convection heat transfer and fluid flow characteristics from a horizontal fluid layer with finned bottom surface. Useful guidelines to enhance the heat transfer rates from the finned surface have been suggested by the authors.

In the present paper, a numerical study of natural convection in a two dimensional vertical channel provided with heated solid protuberances releasing a uniform heat flux is carried out. The protuberances are mounted on the right side wall of the channel. The main objective of this study is to analyse the fluid flow and heat transfer characteristics inside the channel for various protuberances aspect ratios, thermal conductivity and uniform heat flux values. In the next parts, the study will be formulated then the numerical method will be validated against experimental data followed by description of the study approach. The results will be shown and discussed after that and finally we will conclude this paper by our findings and the expected future work in this part of studies.

## 2 STUDY FORMULATION

The study scheme sketched in Figure 1, is a vertical channel of height H containing three protuberances of height HP and Width LP mounted on its right side wall. The protuberances are connected with adiabatic surfaces of height HP and their surfaces release a constant heat flux QP. The bottom and the top sides of the channel are open. The remaining walls parts in the channel are considered as adiabatic surfaces.

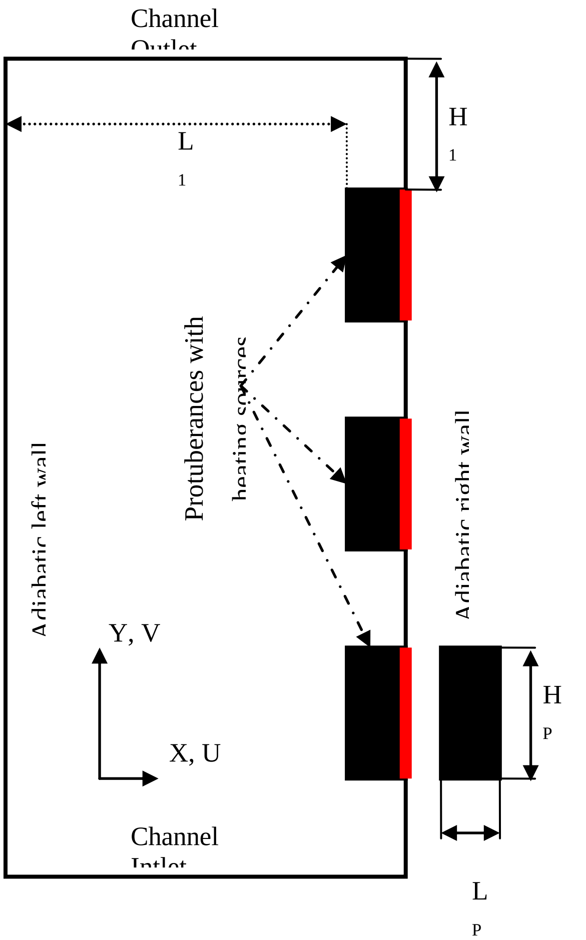


***Figure 1: Study Configuration***

***Table 1: Channel Size data***

| Element | Size |
|---|---|
| $H_1$ | 0.1 m |
| $H_p$ | 0.2 m |
| $L_p$ | See Table 2 |
| $L_1$ | See Table 2 |
| $L_1 + L_p$ (Channel width) | 1 m |
| $3 \times H_p + 4 \times H_1$ (Channel height) | 1 m |

***Table 2: Protuberances Aspect Ratios Data***

| Rp (Aspect ratio) | 0 | 0.4 | 0.8 | 1.2 | 1.6 | 2 |
|---|---|---|---|---|---|---|
| $H_p$ (m) | 0.2 | 0.2 | 0.2 | 0.2 | 0.2 | 0.2 |
| Lp (m) | 0 | 0.08 | 0.16 | 0.24 | 0.32 | 0.4 |
| $L_1$ (m) | 1 | 0.92 | 0.84 | 0.76 | 0.68 | 0.6 |

The mathematical model used is based on the hypothesis of two-dimensional steady flow of an incompressible fluid with constant physical properties and obeying to the Boussinesq approximation.

Before proceeding with the effects study, we have selected the following data:

R. Bessaih and M. Kadja [27] in their study have used a grid for the rectangular channel of 3xL height and 1xL width of 32x90 nodes which means 2880 nodes and in this study we use only 1xL height and 1xL width which means one third of the [27] grid nodes are acceptable for this study grid nodes (value will be 960).

Based on the above data, the following meshes (Figures. 2/3/4/5/6/7) have been built and refined based on the follow options: 10x10, 20x20, 30x30, 35x35, 40x40 and 50x50 structured grids for each protuberances aspect ratio. The selected and refined mesh for this study configuration is 40x40 which means 1600 nodes (see below Table 3) based on the comparison between the Maximum, Average and minimum values of the temperature, velocity, turbulent kinetic energy and the turbulent dissipation rate at the outlet of the vertical channel.

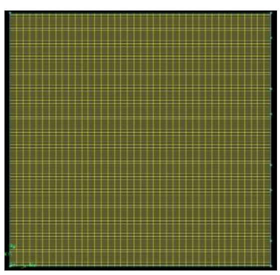

***Figure 2 : Mesh for***

***0 aspect ratio***

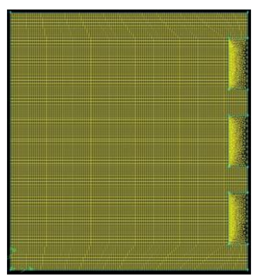

***Figure 3: Mesh for***

***0.4 aspect ratio***

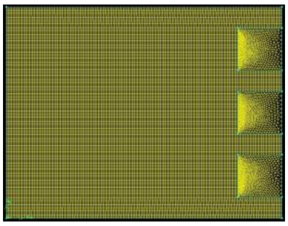

*Figure 4: Mesh for 0.8 aspect ratio*

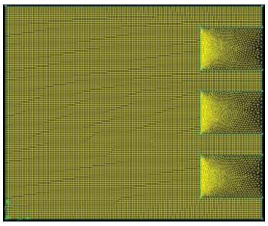

*Figure 5: Mesh for 1.2 aspect ratio*

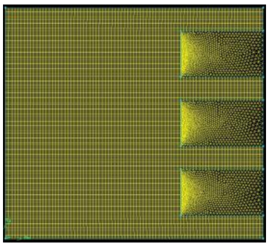

*Figure 6: Mesh for 1.6 aspect ratio*

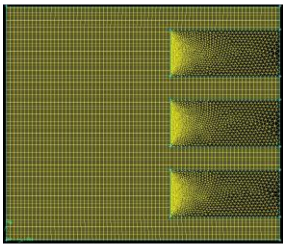

*Figure 7: Mesh for 2 aspect ratio*

*Table 3: Selected Meshes for different protuberances aspect ratios*

| Aspect Ratio | Selected Mesh |
|---|---|
| 0 | 40x40 |
| 0.4 | 40x40 |
| 0.8 | 40x40 |
| 1.2 | 40x40 |
| 1.6 | 40x40 |
| 2 | 40x40 |

After the selection of the meshes' sizes, we have identified the parameters required for the effects study of the geometrical, physical and thermal protuberances properties on the turbulent natural convection inside the vertical channel and given them the following values:

- Protuberances aspect ratios 0 up to 2.
- Uniform heat flux on the external face of the protuberances: from 100 up to 8000 [W/m$^2$].
- Protuberances thermal conductivity: from 16.27 up to 297.73 [W/m.K] by using the follow materials: Gold, Aluminium, Nickel and carbon Steel (check the below Table 4).

Without forgetting the other walls boundaries conditions which they are assumed adiabatic, for that we have selected the gypsum as a material with a thermal conductivity of 0.5 [W/m.K].

*Table 4: Protuberances Effects Study Cases*

| **Material** | **Conductivity [W/m.K]** | **Rp** | **Flux (W/m²)** | | | | | |
|---|---|---|---|---|---|---|---|---|
| | | | **100** | **500** | **1000** | **2000** | **4000** | **8000** |
| **Carbon Steel** | **16.27** | **0** | Case1 | Case2 | Case3 | Case4 | Case5 | Case6 |
| | | **0.4** | Case7 | Case8 | Case9 | Case10 | Case11 | Case12 |
| | | **0.8** | Case13 | Case14 | Case15 | Case16 | Case17 | Case18 |
| | | **1.2** | Case19 | **Case20** | Case21 | Case22 | Case23 | Case24 |
| | | **1.6** | Case25 | Case26 | Case27 | Case28 | Case29 | Case30 |
| | | **2** | Case31 | Case32 | Case33 | Case34 | Case35 | Case36 |
| **Nickel** | **91.74** | **0** | Case37 | - | - | - | - | - |
| | | **0.4** | Case38 | - | - | - | - | - |
| | | **0.8** | Case39 | - | - | - | - | - |
| | | **1.2** | Case40 | - | - | - | - | - |
| | | **1.6** | Case41 | - | - | - | - | - |
| | | **2** | Case42 | - | - | - | - | - |
| **Aluminium** | **202.4** | **0** | Case43 | - | - | - | - | - |
| | | **0.4** | Case44 | - | - | - | - | - |

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | | **0.8** | Case45 | - | - | - | - | - |
| | | **1.2** | Case46 | - | - | - | - | - |
| | | **1.6** | Case47 | - | - | - | - | - |
| | | **2** | Case48 | - | - | - | - | - |
| **Gold** | **297.73** | **0** | Case49 | - | - | - | - | - |
| | | **0.4** | Case50 | - | - | - | - | - |
| | | **0.8** | Case51 | - | - | - | - | - |
| | | **1.2** | Case52 | - | - | - | - | - |
| | | **1.6** | Case53 | - | - | - | - | - |
| | | **2** | Case54 | - | - | - | - | - |

## 3 STUDY APPROACH, RESULTS AND DISCUSSION

### 3.1 Study Validation

The numerical method developed in this study was first validated using the experimental data from F. Ampofo and T.G. Karayiannis [26] corresponding to a free protrusion geometry. They studied the turbulent natural convection flow in a closed cavity, differentially heated from the sides

with adiabatic horizontal walls. Figure 7 shows the comparison between the air temperature at the cavity middle height measured by a laser-Doppler Anemometer LDA and micro-diameter thermocouple which helps to eliminate the above mentioned problems associated with a multi-sensor hot and cold wire anemometry system. For Rayleigh number equal to $1.5 \times 10^9$.

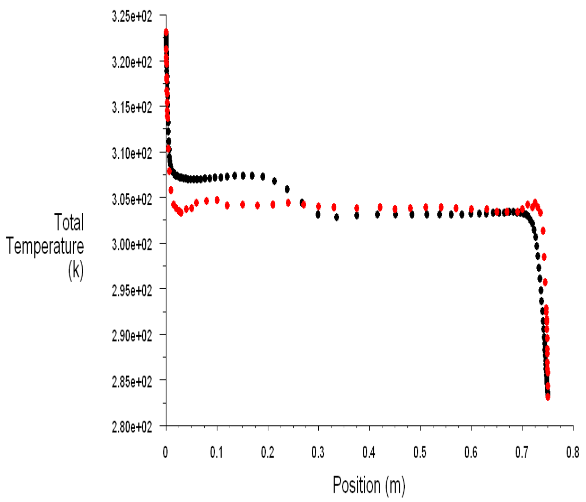


***Figure 8: Temperature profile in the middle height of the cavity: comparison computations and experimental data***

It is clear that the predicted temperature contains the three distinct regions of the flow: the upward and downward buoyancy generated near wall regions and the core region which is stagnant. These predictions are in close agreement with measurements the black curve for the study method results and the red curve for the experimental data (2.4% as a deviation between the study and experimental average temperature values). The Figure 9 represents the schematic diagram of the cavity.

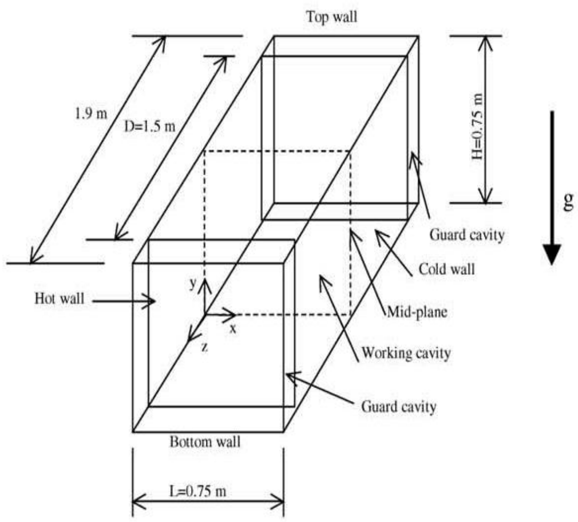


***Figure 9 : Experimental three-dimensional schematic diagram of the air filled cavity***

### 3.2 Study Approach

Our adopted approach of the plan for the effects study is based on the following study cases shown in the Table 5: in each case we will take set of values regarding the three parameters protuberances geometrical aspect ratio, thermal conductivity and the uniform heat flux applied on the protuberances external faces.

The code used for the numerical simulation is based on the below characteristics:

- The finite volume method to discretize the partial differential equations of the mathematical model: Pressures have used the PRESTO scheme and the second order upwind scheme for the momentum, turbulent kinetic energy, turbulent dissipation rate and energy terms.
- Steady state simulation.
- Standard k-$\varepsilon$ Turbulence Model with near wall treatment.
- SIMPLE algorithm for the velocity-pressure coupling.
- Under relaxation factors: pressure 0.3, density 1, Body forces 1, momentum 0.7, turbulent kinetic energy 0.8, turbulent dissipation rate 0.8, turbulent viscosity 1 and energy 1.
- Convergence absolute criteria: continuity 0.5%, velocity 0.5%, energy 0.001%, turbulent kinetic energy 0.5% and turbulent dissipation rate 0.5%.
- Prandtl number 0.71.

### 3.3 Study Results

By using the above data and simulation approach characteristics we got a set of results and as an example the case20 has the below air temperature and velocity distributions inside the vertical channel shown in the Figures 10-11:

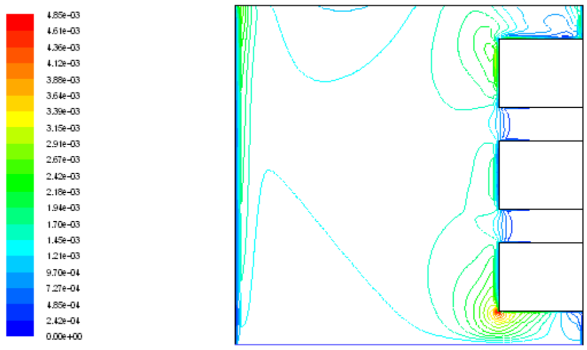


***Figure 10: Air velocity inside the vertical channel for the case 20***

The study effects results can be shown by combining two of the three parameters protuberances aspect ratio, thermal conductivity and the uniform heat flux, for that purpose we have selected combinations in which we have the below graphs (Figures 12 up to 19).

As summary effects study data, we have collected them in the below Table 6.

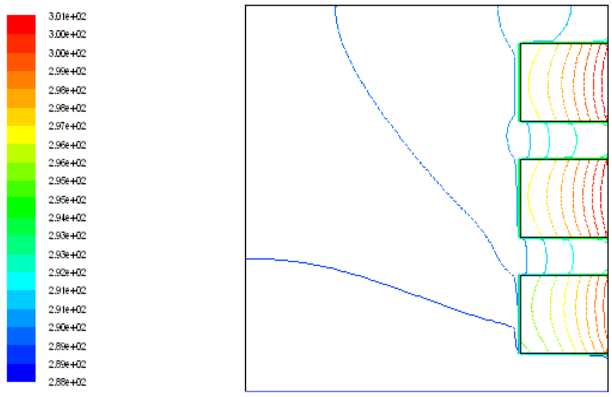


***Figure 11 : Air temperature inside the Vertical channel for the case 20***

***Table 5: Summary Effects Study Data***

| *Protuberances Parameter* | *Range* | *Channel Air average temperature* [C] | *Channel convective heat transfer coefficient* [W/m$^2$.k] | *Channel turbulence intensity* [%] | *Channel Rayleigh number* [-] |
|---|---|---|---|---|---|
| *Aspect ratio* | *0-2* | *0% up to 6.5%* | *0% up to 200%* | *-7% up to 2%* | *0% up to 65%* |

### 3.4 Study Discussion

We can understand from the below Figures 12 up to 19 that the energy, momentum and turbulence parameters have various changes caused by the three independent study parameters aspect ratio, thermal conductivity and uniform heat flux compared to the aspect ratio case of 0. These effects are described case by case and as per the below dependent parameters. We have tried to present these effects by putting the channel without protuberances Rp=0 as a reference and have calculated the deviation of the each cases to the reference case in order to predict the performance changes of the turbulent natural convection characteristics inside the channel and deduce the difference between the cases. In the following paragraphs we have discussed these deviations and changes of the turbulent natural convection characteristics:

#### *3.4.1 Channel Air average temperature deviation*

The Figure 12 of the channel Air average temperature deviation versus the thermal conductivity and aspect ratios of the protuberances for a uniform heat flux of 100 [W/m2] shows that the average temperature deviation is directly proportional to the aspect ratios and inversely proportional to the thermal conductivity. As an overall reading, the average temperature deviation is increasing within the range of 0 up to 2% compared to the protuberances' with an aspect ratio of 0.

For Figure 17 the channel Air average temperature deviation versus the applied heat fluxes and aspect ratios of the protuberances for carbon steel protuberances shows that the average temperature deviation is directly proportional to the aspect ratios and the uniform heat fluxes. As an overall reading, the air average temperature is increasing within the range of 0 up to 6.5% compared to the protuberances' with an aspect ratio of 0.

That means that the air average temperature is increasing by the increasing of the aspect ratio of the protuberances which means the heat exchange surface of the protuberances is participating in the enhancement of the heat transfer inside the channel.

#### *3.4.2 Channel Air heat transfer coefficient deviation*

The Figure 13 of the channel Air heat transfer coefficient deviation versus the thermal conductivity and aspect ratios of the protuberances for a uniform heat flux of 100 [W/m2] shows that the heat transfer coefficient deviation is directly proportional to the protuberances' aspect ratios and no changes with the thermal conductivities. As an overall reading, the heat transfer coefficient is increasing within the range of 0 up to 200% compared to the protuberances' with an aspect ratio of 0.

For Figure 16 the channel air heat transfer coefficient deviation versus the applied heat fluxes and aspect ratios of the protuberances for carbon steel protuberances shows that the heat transfer coefficient deviation is directly proportional to the protuberances' aspect ratios and no changes with the uniform heat fluxes. As an overall reading, the heat transfer coefficient is increasing within the range of 0 up to 200% compared to the protuberances' with an aspect ratio of 0.

That means that the air average heat transfer coefficient is increasing by the increasing of the aspect ratio of the protuberances which means the heat exchange surface of the protuberances is participating in the enhancement of the heat transfer inside the channel.

#### *3.4.3 Channel turbulence intensity deviation*

The Figure 14 of the channel turbulence intensity deviation versus the thermal conductivity and aspect ratios of the protuberances for a uniform heat flux of 100 [W/m2] shows that the channel turbulence intensity deviation is directly proportional to the thermal conductivity and a nonlinear variation versus the aspect ratios with a highest point related 0.4 as an aspect ratio. As an overall reading, the turbulence intensity is changing (increasing or decreasing) within the interval of -7% up to 2% compared to the protuberances' with an aspect ratio of 0.

For Figure 18 the channel turbulence intensity deviation versus the applied heat fluxes and aspect ratios of the protuberances for carbon steel protuberances shows that the channel turbulence intensity deviation is directly proportional to the thermal conductivity and a nonlinear deviation versus the aspect ratios with a highest point related 0.4 as an aspect ratio. As an overall reading, the turbulence intensity is changing (increasing or decreasing) within the interval of -7% up to 2% compared to the protuberances' with an aspect ratio of 0.

That means that the average turbulence intensity is increasing by the increasing of the aspect ratio of the protuberances in some values but there are reverse effect when the aspect ratio is too high which means the turbulence intensity will be inversely affected if the aspect ratio is high for our case more than 0.4. In general, the aspect ratio can help positively in the enhancement of the heat transfer by increasing the turbulence intensity.

#### *3.4.4 Channel Rayleigh number deviation*

The Figure 15 of the channel Rayleigh number deviation versus the thermal conductivity and aspect ratios of the protuberances for the uniform heat flux of 100 [W/m2] shows that the Rayleigh number deviation is directly proportional to the protuberances' aspect ratios and no variation with the thermal conductivities. As an overall reading, the Rayleigh number is increasing within the range of 0 up to 65% compared to the protuberances' with an aspect ratio of 0.

For Figure 19 the channel Rayleigh number deviation versus the applied heat fluxes and aspect ratios of the protuberances for carbon steel protuberances shows that the Rayleigh number deviation is inversely proportional to the uniform heat fluxes and directly proportional to the aspect ratios from 100 up to 4000 [W/m2] then this observation is not valid from 4000 up to 8000 [W/m2] in which a parabolic variation is shown. As an overall reading, the Rayleigh number is changing (increasing or decreasing) within the deviation interval of 0% up to 65% compared to the protuberances' with an aspect ratio of 0.

That means that the Rayleigh number is affected positively by the presence of the protuberances and therefore the heat transfer by natural convection inside the channel is improved.

These parameters changes are based generally on the effect of the protuberances aspect ratio changes which affects the heat exchange surfaces and in the same time the size of protuberances will be as barriers for air movement and or extra heat exchange surfaces.

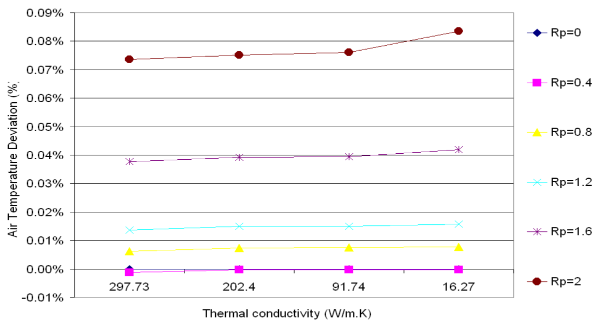


***Figure 12 : Channel Air temperature deviation versus the thermal conductivity and aspect ratios of the protuberances for the uniform heat flux of 100 [W/m2]***

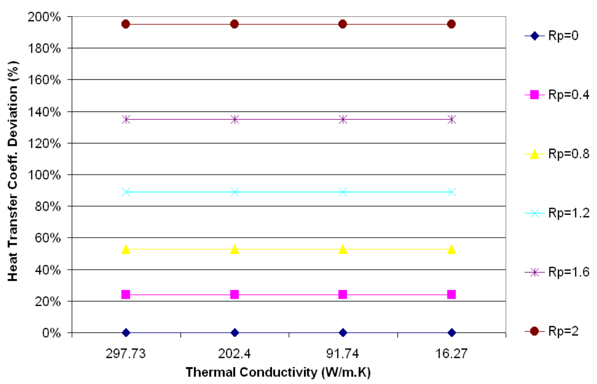


***Figure 13 : Channel Air heat transfer coefficient deviation versus the thermal conductivity and aspect ratios of the protuberances for the uniform heat flux of 100 [W/m2]***

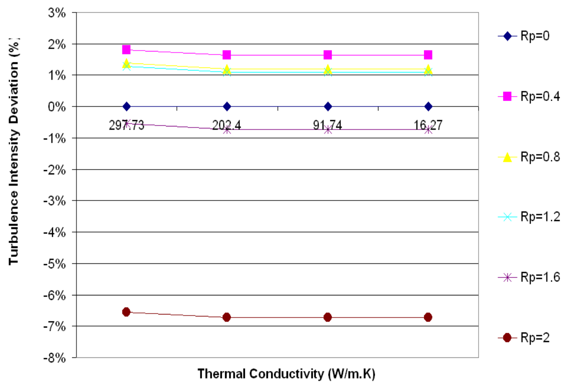


***Figure 14 : Channel turbulence intensity deviation versus the thermal conductivity and aspect ratios of the protuberances for the uniform heat flux of 100 [W/m2]***

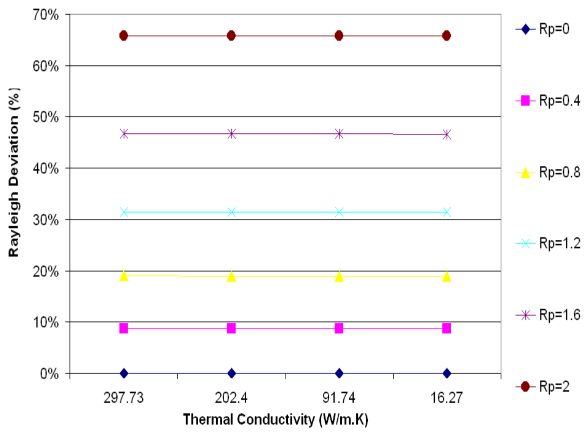


***Figure 15: Channel Rayleigh number deviation versus the thermal conductivity and aspect ratios of the protuberances for the uniform heat flux of 100 [W/m2]***

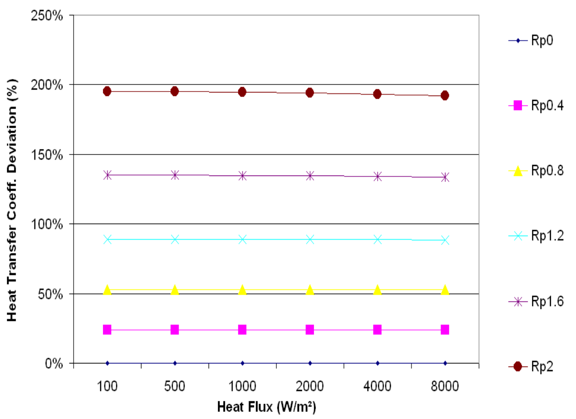


***Figure 16 : Channel Air heat transfer coefficient deviation versus the applied heat fluxes and aspect ratios of the protuberances for carbon steel protuberances***

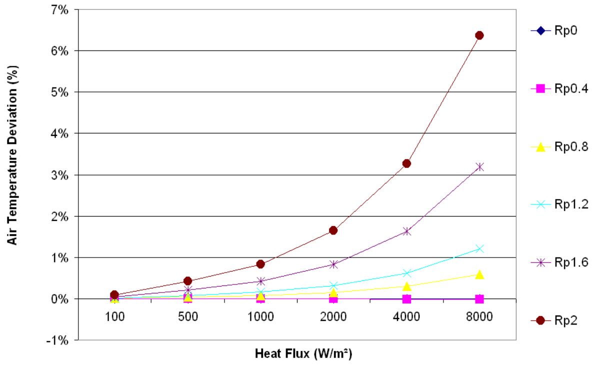


***Figure 17 : Channel Air temperature deviation versus the applied heat fluxes and aspect ratios of the protuberances for carbon steel protuberances***

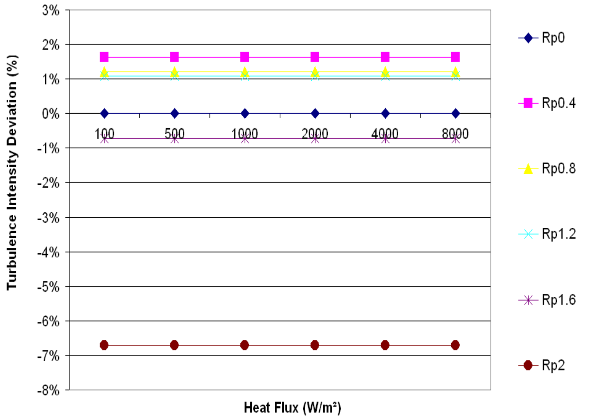


***Figure 18: Channel turbulence intensity deviation versus the applied heat fluxes and aspect ratios of the protuberances for carbon steel protuberances***

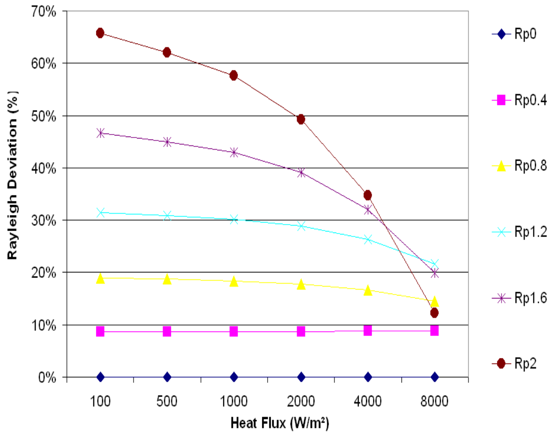


***Figure 19: Channel Rayleigh number deviation versus the applied heat fluxes and aspect ratios of the protuberances for carbon steel protuberances***

## 4 CONCLUSION

From the study results and findings, we can conclude that the solid protuberances have an effect on the turbulent natural convection performance inside the vertical channel compared to the channel without protuberances and the magnitude of the effect has been quantified between 0 and 6.5% based on the protuberances characteristics (aspect ratios 0 up to 2, thermal conductivities 16.27 up to 297.73 [W/m.K] and the applied uniform heat fluxes 100 up to 8000 [W/m2]). In addition, the effect of the aspect ratio is more visible than the one of the thermal conductivity and or the applied uniform heat flux.

For future studies we suggest to study the protuberances' shape and/or the porosity on the turbulent natural convection performance inside the vertical channel.

## NOMENCLATURE

k: Turbulent kinetic energy

H: Height [m].

L: Width [m].

Q: Uniform Heat Flux [W/m$^2$]

P: Protuberance.

Rp: aspect Ratio.

Greek Symbols:

$\varepsilon$ : Turbulent dissipation rate.

Subscript:

1, 2, 3: sequential numbering.

P: Protuberance

## REFERENCES


**[1]** Soret-induced convection of a micropolar fluid in a vertical channel, Computers and Mathematics with Applications 62 (2011) 725–736.

**[2]** Assunta Andreozzi, Antonio Auletta, Oronzio Manca, Entropy generation in natural convection in a symmetrically and uniformly heated vertical channel, International Journal of Heat and Mass Transfer 49 (2006) 3221–3228.

**[3]** Young TJ, Vafai K., Convective cooling of a heated obstacle in a channel. Int. J. Heat Mass Transfer 1998 (41) 3131–48.

**[4]** Young TJ, Vafai K., Convective flow and heat transfer in a channel containing multiple heated obstacles. Int. J. Heat Mass Transfer 1998 (41) 3279–98.

**[5]** Leung CW, Chen S, Chan TL. Numerical simulation of laminar forced convection in an air-cooled horizontal printed circuit board assembly. Numer. Heat Transfer Part A 2000 (37) 373–93.

**[6]** Herman C, Kang E. Experimental visualisation of temperature fields and study of heat transfer enhancement in oscillatory flow in a grooved channel. Heat Mass Transfer 2001 (37) 87–99.

**[7]** Korichi A, Oufer L. Numerical heat transfer in a rectangular channel with mounted obstacles on upper and lower walls. Int. J. Therm. Sci. 2005 (44) 644–55.

**[8]** Moon JW, Kim SY, Cho HH. Frequency-dependent heat transfer enhancement from rectangular heated block array in a pulsating channel flow. Int. J. Heat Mass Transfer 2005 (48) 4904–13.

**[9]** Chen S, Liu Y, Chan SF, Leung CW, Chan TL. Experimental study of optimum spacing problem in the cooling of simulated electronic package. Heat Mass Transfer 2001 (37) 251–7.

**[10]** Chen S, Liu Y. An optimum spacing problem for three-by-three heated elements mounted on a substrate. Heat Mass Transfer 2002 (39) 3–9.

**[11]** Najam M, El Alami M, Oubarra A. Heat transfer in a ‘‘T” form cavity with heated rectangular blocks submitted to a vertical jet: the block gap effect on multiple solutions. Energy Convers Manage 2004 (45) 113–125.

**[12]** Baskaya S, Erturhan U, Sivrioglu M. Experimental investigation of mixed convection from an array of discrete heat sources at the bottom of a horizontal channel. Heat Mass Transfer 2005 (42) 56–63.

**[13]** Dogan A, Sivrioglu M, Baskaya S. Experimental investigation of mixed convection heat transfer in a rectangular channel with discrete heat sources at the top and at the bottom. Int. Common Heat Mass Transfer 2005 (32) 1244–52.

**[14]** Premachandran B, Balaji C. Mixed convection heat transfer from a horizontal channel with protruding heat sources. Heat Mass Transfer 2005 (41) 510–8.

**[15]** Lin TY, Hsieh SS. Natural convection of opposing/assisting flows in vertical channels with asymmetrically discrete heated ribs. Int. J. Heat Mass Transfer 1990; 33(10):2295–309.

**[16]** Kelkar KM, Choudhury D. Numerical prediction of periodically fully developed natural convection in a vertical channel with surface mounted heat generating blocks. Int. J. Heat Mass Transfer 1993; 36(5):1133–45.

**[17]** Lee JR, Ha MY, Balachandar S, Yoon HS, Lee SS. Natural convection in a horizontal layer of fluid with a periodic array of square cylinders in the interior. Phys Fluids 2004; 16(4):1097–117.

**[18]** Hasnaoui M, Bilgen E, Vasseur P. Natural convection above an array of open cavities heated from below. Numer. Heat Transfer Part A 1990;18:463–82.

**[19]** Amahmid A, Hasnaoui M, Vasseur P. Multiplicité des solutions en convection naturelle dans une géométrie répétitive. Int J Heat Mass Transfer 1997;40(16):3805–18.

**[20]** Bakkas M, Amahmid A, Hasnaoui M. Steady natural convection in a horizontal channel containing heated rectangular blocks periodically mounted on its lower wall. Energy Convers Manage 2006 (47) 509–28.

**[21]** Kwak CE, Song TH. Natural convection around horizontal downward-facing plate with rectangular grooves: experiments and numerical simulations. Int. J. Heat Mass Transfer 2000 (43) 825–38.

**[22]** El Alami M, Najam M, Semma E, Oubarra A, Penot F. Chimney effect in a ‘‘T” form cavity with heated isothermal blocks: the blocks height effect. Energy Convers Manage 2004 (45) 3181–91.

**[23]** El Alami M, Najam M, Semma E, Oubarra A, Penot F. Electronic components cooling by natural convection in horizontal channel with slots. Energy Convers Manage 2005 (46) 2762–72.

**[24]** Arquis E, Rady M. Study of natural convection heat transfer in a finned horizontal fluid layer. Int. J Therm. Sci. 2005 (44) 43–52.

**[25]** Assunta Andreozzi, Antonio Campo, Oronzio Manca, Compounded natural convection enhancement in a vertical parallel-plate channel, International Journal of Thermal Sciences 47 (2008) 742–748.

**[26]** A. Ampofo, T.G. Karayiannis, Experimental benchmark data for turbulent natural convection in an air filled square cavity. Int. J. Heat Mass Transfer 46 (2003); 3551–3572.

**[27]** R. Bessaih, M. Kadja, Turbulent natural convection cooling of electronic components mounted on a vertical channel, Applied thermal engineering 20 (2000), 141-154.