Laminar Flow around an Array of 3D Protruding Heaters Mounted in Cross-Stream Direction

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Abstract

Numerical analysis was performed to investigate the characteristics of the laminar fluid flow around an array of 3D protruding heaters mounted on the bottom substrate of a parallel plane channel using the ANSYS/Fluent® 14.0 commercial software. The fluid flow was considered to have constant properties under steady state conditions. In the channel inlet, the velocity profile was uniform. This problem is associated with forced flow over the electronic components mounted on printed circuit boards. The conservation equations and their boundary conditions were numerically solved in a single domain through a coupled procedure. The discretization of the equations was based on the Control Volumes Method. The algorithm SIMPLE was used to solve the pressure-velocity couple. Due to the non-linearity of the momentum equation, the correction of the velocity components and the pressure were under-relaxed to prevent instability and divergence. After a study of the computational mesh independence, the numerical results were obtained, displayed as a 3D non-uniform mesh with 212,670 control volumes. This computational mesh was more concentrated near the solid-fluid interface regions due to the larger primitive variable gradients in these regions. An investigation was done on the effects of the Reynolds numbers where the Reynolds numbers ranged from 100 to 300 and was dependent on the heights of the protruding heaters. The main characteristics of the fluid flow consisted of a small recirculation upstream of the heaters, the formation of horseshoe vortices around the protruding heaters? side walls and a large recirculation region downstream of the heaters. The fluid dynamics parameters of interest, the velocity profiles, local and average skin friction coefficient, pressure distribution and the Darcy-Weisbach friction factor, were found and compared to the results available in the literature.

Index terms— laminar flow, 3D protruding heaters, streamlines, recirculation, horseshoe vortices.

1 Introduction

he search for improvements and technological innovations by means of industrial development and academic research on the cooling of electronic equipment in the last two decades has been very intense. The most common method of heat transfer in source elements is still convection cooling, utilizing air as the work fluid. This choice was made because air is easily available, the devices required to move it are normally low cost, and it is 100% non-polluting **??**Nishida, 2012). In this work, problems motivated by the Level 2 of electronic packaging, associated with the thermal control of one row of 3D protruding heaters mounted on a printed circuit board (PCB) were considered, as shown in Fig. 1 (Alves, 2010). The available space for the heaters can be limited and the cooling process must be done through forced convection with moderate velocities (low Reynolds number) due to operational limitations and noise reduction. Under such conditions, there may not be enough space to work with heat sinks in these concentrate heat dissipation components. These components can be simulated by protruding blocks mounted on a parallel plate channel. Hwang & Yang (2004) presented a numerical study of the vortices structures of the flow (in a range of Reynolds numbers from low to moderate) around a cubic obstacle mounted on a plate in a channel. The main characteristics of the flow were horseshoe vortices upstream the obstacle, side vortices around the side faces of the cube, and "hair pin" vortices near the wake region. It was observed that as the flow approached the cube, an adverse pressure gradient produced a separate 3D boundary layer, allowing laminar horseshoe vortices to form. It was also noticed that as the Reynolds number increased, the structure of the horseshoe system became more complex and the number of vortices increased in pairs. Van Dijk& De Lange (2007) conducted a numerical study of a flow over one cubic obstacle mounted on the base of a parallel plate channel, considering either compressible or incompressible laminar flow. The Reynolds number was investigated in a range from 50 to 250, and the Mach number was varied between 0.1 and 0.6. The main flow characteristics around the obstacle were the formation of horseshoe vortices, vortices developing on the side T ( ) A walls of the obstacle, and, downstream of the obstacle there was a wake with two counter-rotating vortices. It was noticed that the shape and size of these flow characteristics are determined mainly by the Reynolds number, verifying that for greater Reynolds numbers, the horseshoe vortices as well as the wake region extended over a significantly broader area. The correlation between the separation and reattachment point position with the Reynolds number was also presented. Other studies relating to the flow around 3D protruding heater(s) were performed by Castro & Robins (1977), **??**ropea&Gacktatter (1985), Martinuzzi&Tropea (1993), Okamotoet al. **??**1997).

2 II.

3 Analysis

The basic configuration representing the treated problem for one of the 3D protruding heaters is indicated in Fig. 2. In this case, the channel has a height, H, length, L, and width, W. The substrate has the same length and width as the channel with a thickness, t. The heater has a length, L h , height, H h , width, W h and it is located at a distance, L u , from the channel entry. The space between the heaters is 2Ws. The cooling process occurred through a forced laminar flow with constant properties under steady state conditions. In the channel entrance, the flow velocity profile (u 0) was considered uniform. The mathematical model of the present problem was performed for a single domain: the solid regions (protruding heater and substrate) and the fluid flow in the channel. Due to the problem symmetries, the conservation equations were formulated for the domain with length, L, width, W/2 and height, (H + t), as Fig. 3 shows.

4 L u = 2H

L h = 0.75H The governing equations cover the conservation principles in the considered domain. Steady state conditions, constant properties and negligible viscous dissipation were assumed. The occasional effects of oscillation in the flow are not being considered in this modeling: a typical procedure adopted in similar problems, i.e. Alves&Altemani (2012), Zeng &Vafai (2009) e Davalath&Bayazitoglu (1987).L d = 3.75H u 0 x y L = 6.5H H h = 0.3H H t = 0.1H z x W s = 0.2H W h = 0.6H W = H ( ) A Year

? Mass Conservation (Continuity Equation)0 ? ? = u (1)

? Momentum Conservation (Navier-Stokes Equation)

( )2 p ? µ ?? = ?? + ? u u u (2)

The boundary conditions of the flow were uniform velocity (u 0) at the channel inlet, and null velocity at the solid-fluid interfaces (no-slip condition). At the channel outlet, the flow had its diffusion neglected in the x direction. In the solution domain at the lateral boundaries, the symmetry condition (periodic condition) was applied for the velocity fields (same geometry in each of the 3D protruding heater).

5 b) Fluidynamic Parameters of Interest

The solution of the governing equations output the velocity and pressure distributions in the considered domain. The numerical solutions of the primary variables distribution (u, v, w, p) were utilized to define the derived quantities. The Reynolds number in the channel was based on the protruding heater height (H h) and calculated as0 0 h h u H u H Re ? µ ? = = . (**3**)

The local skin friction coefficient, C f (?), can be written as( ) ( ) 2 0 2 p f C u ? ? ? ? = ? ? ? ? ? ? ? . (**4**)

? p (?) is the local sheer stress in a surface of the heater.

The mean friction coefficient, f C , can be written as2 0 2 p f C u ? ? = ? ? ? ? ? ? ? . (**5**)

p ? iis the mean shear stress at the heater surfaces.

The Darcy-Weisbach (or Moody) friction factor can be defined in terms of the total pressure drop in the channel (Î?"p) by the equation2 0 2 p H f u ? ? = ? ? ? ? ? ? (6)

6 c) Numerical Solution

The governing equations and their boundary conditions were numerically solved utilizing the Control Volume Method (Patankar, 1980) through the commercial software ANSYS/Fluent ® 14.0. The algorithm SIMPLE (Semi-Implicit Method for Pressure Linked Equations) was used to treat the pressure-velocity couple. The boundary conditions were applied at the edges of the analyzed domain (Fig. 3). After a mesh independency study, the numerical results were ( ) A Year obtained with a non-uniform 3D mesh containing 212,670 control volumes. This mesh was more concentrated in the regions near the solid-fluid interfaces due to the larger gradients in the primitive variables of these regions, as shown in Fig. 4. Due to the non-linearity in the Momentum Equation, the velocity components and the pressure correction were underrelaxed to prevent instability and divergence. The stop criteria of the iterative solving process was established for absolute changes in the primitive variables smaller than four significant figures between two consecutive iterations, while the global mass conservation in the domain was satisfied in all of the iterations. The numerical solutions were processed in a microcomputer with an Intel ® Core TM 2 Duo E7500 2,94GHz processor and 4GB of RAM. The processing time of a typical solution was approximately 10 minutes.

7 Results and Discussion

Typical geometry and property values, relevant to the electronic components mounted on printed circuit board cooling applications, were used to obtain the numerical results (BAR-COHEN et al., 2003). The geometric configuration showed in Fig. 2, were assumed considering a space H = 0.0254m between the parallel plates. Air was considered the cooling fluid. The fluid properties were considered constant, obtained at 300 K (INCROPERA et al., 2008). The effects of the Reynolds numbers Re = 100, 150, 200, 250, and 300 were investigated. According to Morris &Garimella (1996), the flow is laminar for this range of Re.

In Figure 5, the streamlines over a 3D protruding heater, in a perspective view, are presented for Reynolds numbers of 100, 200, and 300. The main characteristics of the laminar flow are the horseshoe vortices which start upstream the heater and develop around the heater's lateral surfaces; a small recirculation upstream the protruding heater; the detachment of the fluid's boundary layer at the top of the heater causing a recirculation (reverse flux); and a large recirculation region downstream the heater due to the flow reattachment. It is interesting to state that the fluid flow development around the 3D protruding heaters' lateral surfaces does not freely happen due to the small space between the heaters. The recirculation length (L rec) downstream the protruding heater, or the distance between the base of the heater's rear surface and the reattachment point of the fluid's boundary layer, is presented in function of Reynolds in Tab. 1. The same results are shown in Fig. 11, where it is observed that the recirculation length varies linearly with Re. A correlation with deviations smaller than 0.35% is presented in Eq. (**??**). From all presented results, the greatest length (L rec) was approximately 2.75H, ensuring that the recirculation is always in the studied domain. The results for the Darcy-Weisbach(or Moody) friction factor and the mean friction coefficient can be correlated with deviations smaller than 1.5% using 0 359 0 051 , f C , Re ? = , (**8**)0 359 0 204 , f , Re ? = . (**9**

8 Conclusions

In this work, a numerical analysis of the laminar flow around an array of 3D protruding heaters mounted on the bottom wall (substrate) of a parallel plate channel was made utilizing the commercial software ANSYS/Fluent ® 14.0. Air was considered as the cooling fluid. The cooling process occurred through a forced laminar flow with constant properties under steady state conditions. At the channel's inlet, the velocity profile of the flow was uniform. The conservation equations and the respective boundary conditions were numerically solved in a single domain that incorporated the regions of solid and fluid, through a coupled procedure utilizing the Control Volume Method. The occasional effects of oscillation in the flow were not considered. Due to the problem symmetries, the basic configuration of the problem was reduced to the one in Fig. 2 and the solution domain utilized was showed in Fig. 3. Typical geometry and property values, relevant to the electronic components mounted on printed circuit board cooling applications, were used to obtain the numerical results. The geometric configuration showed in Fig. 2, were assumed considering a space H = 0.0254m between the parallel plates. The effects of the Reynolds number, based on the protruding heaters height, were inspected for Re = 100, 150, 200, 250, and 300. The flow in the channel was always laminar for the range of Re investigated.

The behavior of the laminar flow over the 3D protruding heaters mounted in cross-stream direction was showed through the streamlines. The streamlines over a 3D protruding heater were presented for Reynolds numbers of 100, 200, and 300. The main characteristics of the laminar flow were the horseshoe vortices which start upstream the heater and develop around the heater's lateral surfaces; a small recirculation upstream the protruding heater; the fluid's boundary layer detachment at the top of the heater causing a recirculation (reverse flux); and a large recirculation region downstream the heater due to the flow reattachment. The recirculation length (L rec) downstream the protruding heater varies linearly with Re. The velocity magtintudes, the recirculation directions and the pressure distributions at the different regions of the air laminar flow, were presented for the planes xy, xz e yz. The local skin friction distribution on the walls of the parallel plate channel and on the 3D protruding heater surfaces, were also showed.

8 CONCLUSIONS

It is interesting to state that the fluid flow development around the 3D protruding heaters lateral surfaces does not freely happen due to the small space between the heaters. The fluid-dynamic symmetry conditions of the blocks were dominant and the corresponding flow was different than a single 3D protruding heater with free domain in the cross-stream direction to the flow. [1] [2] [3] [4]

1

Figure 1: Figure 1 :

[2]© 2013 Global Journals Inc. (US) velocity component values. A negative velocity value (u) represents a reverse flux in relation to the main flow.

[3]© 2013 Global Journals Inc. (US) notation presented in Fig. 19 is used in this work.

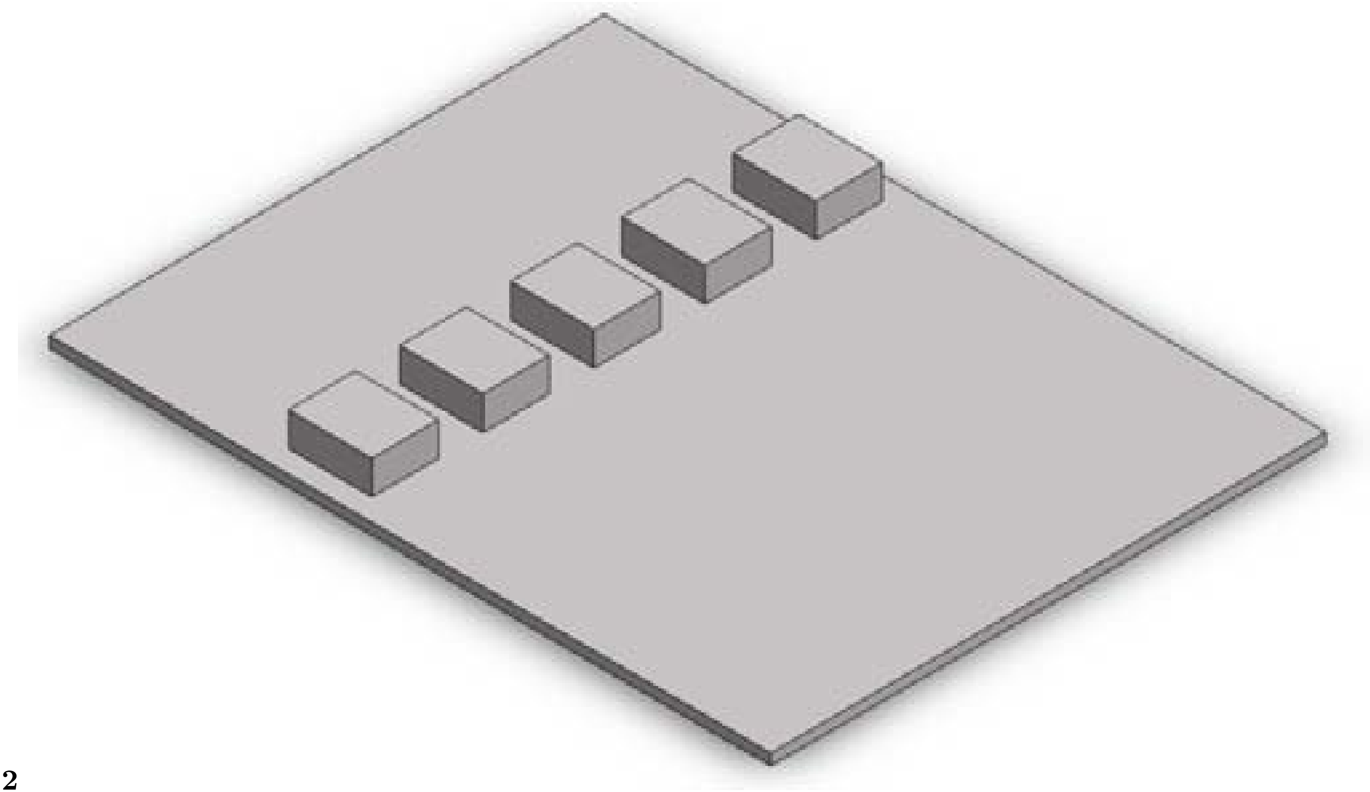

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Figure 2: Figure 2 :

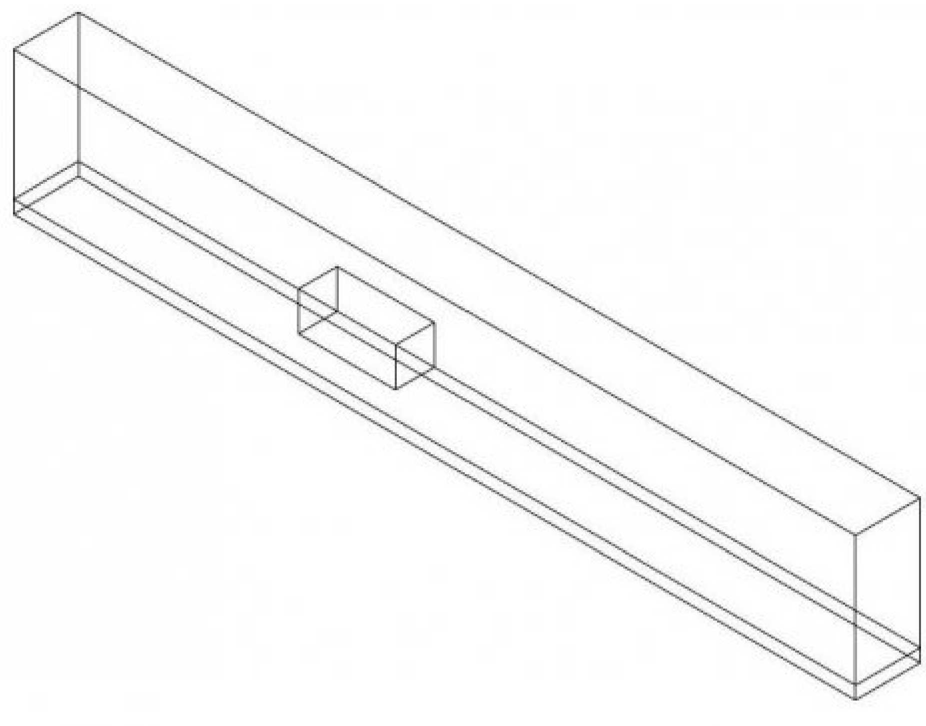

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Figure 3: Figure 3 :

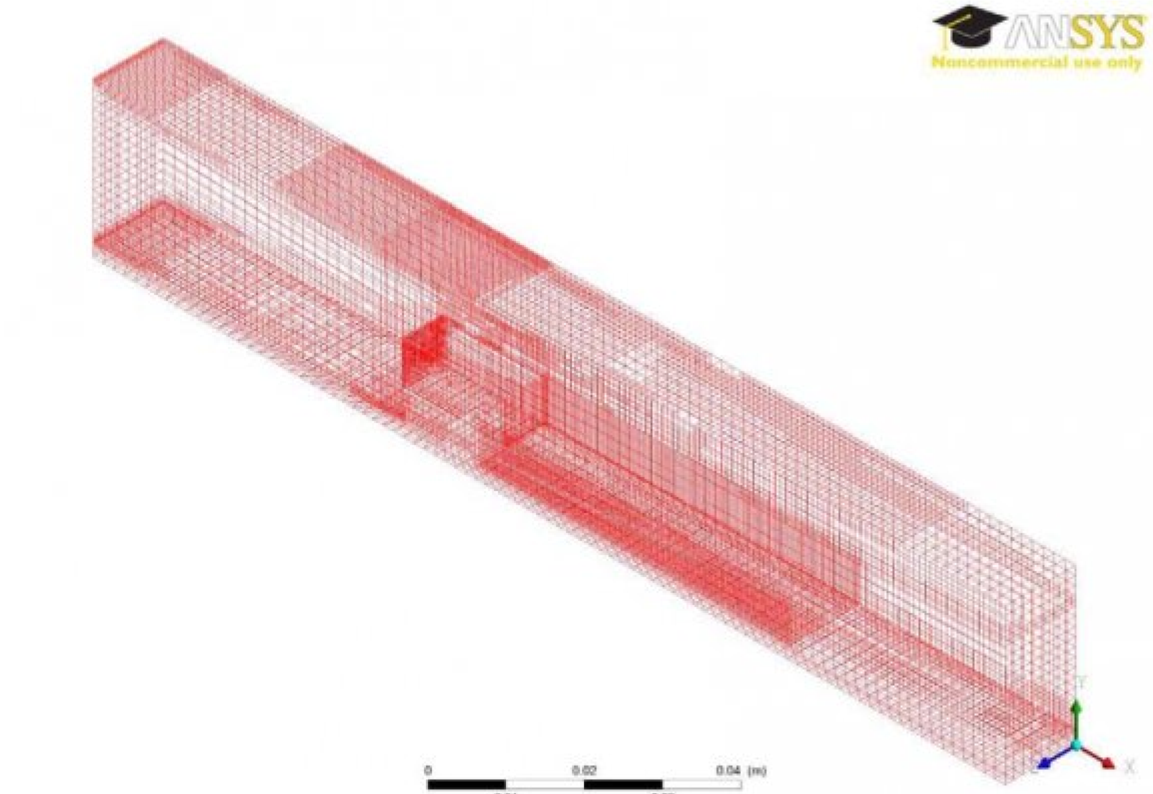

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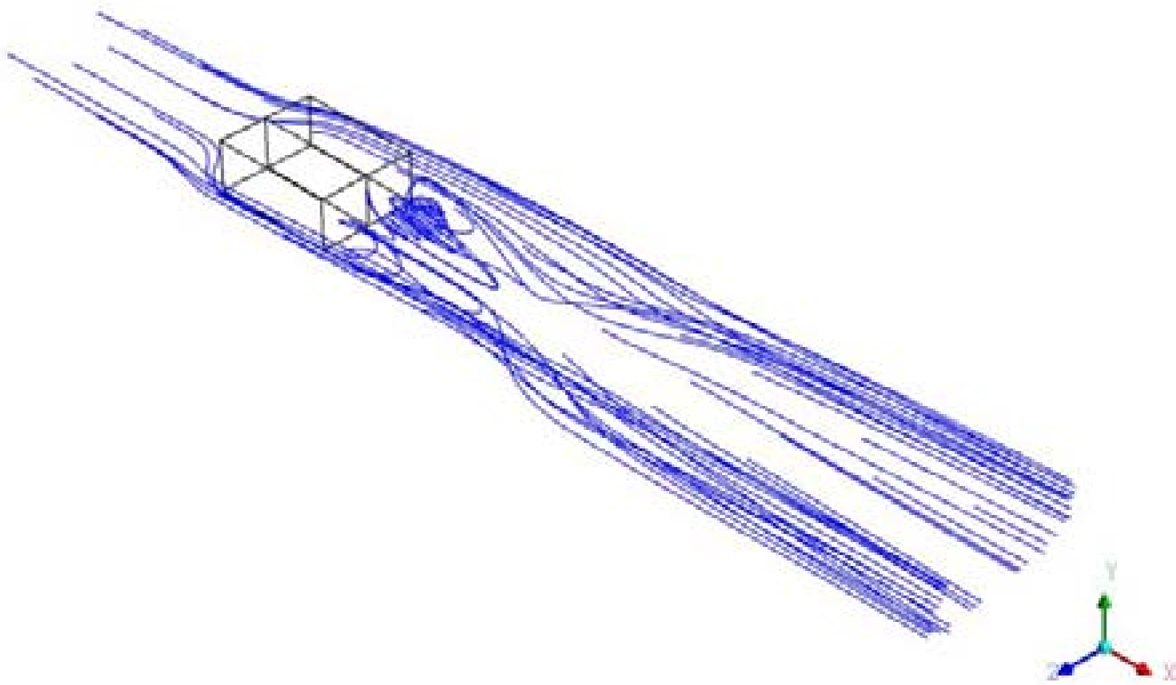
Figure 4: Figure 4 :



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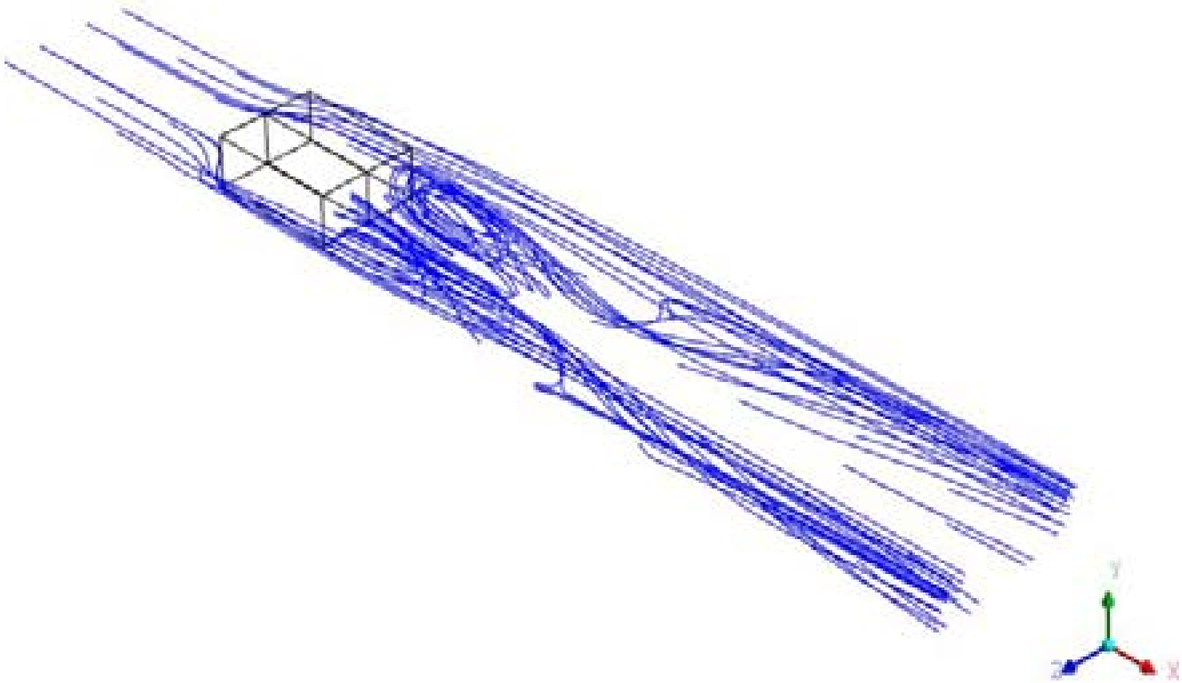

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Figure 6: Figure 6 : 7 ,

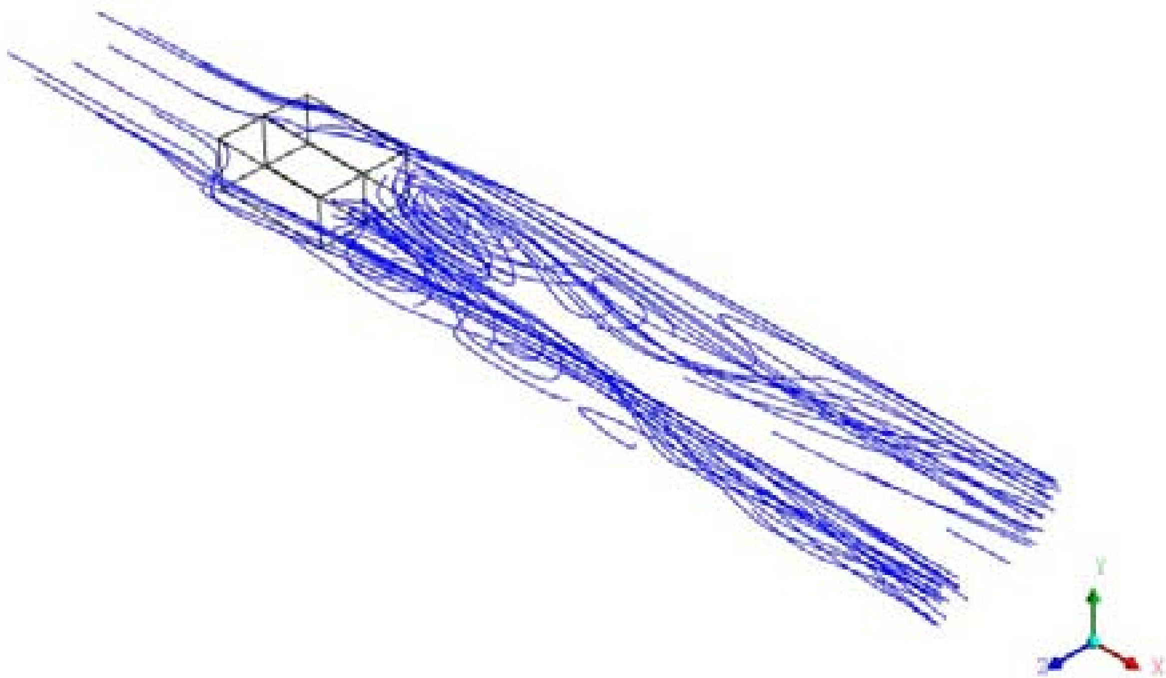

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Figure 7: Figure 7 :Figure 8 :Figure 9 :

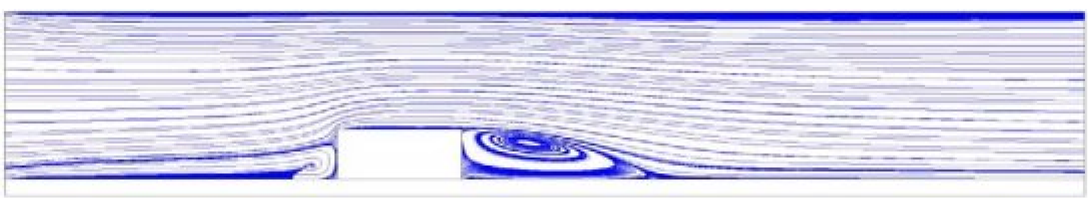

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Figure 8: Figure 10 :

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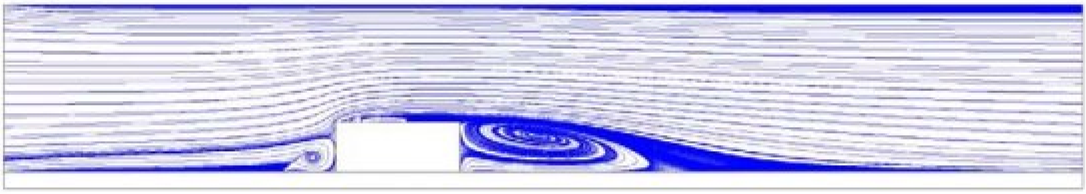

Figure 9: Figure 11 :Figure 12 :Figure 13 :Figure 14 :

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Figure 10: Figure 15 :Figure 16 :Figure 17 :Figure 18 :Figure 20

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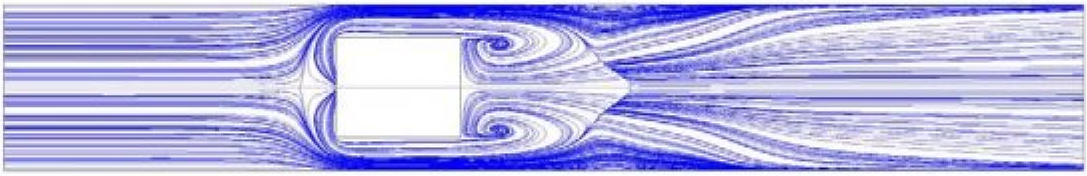

Figure 11: Figure 20 :

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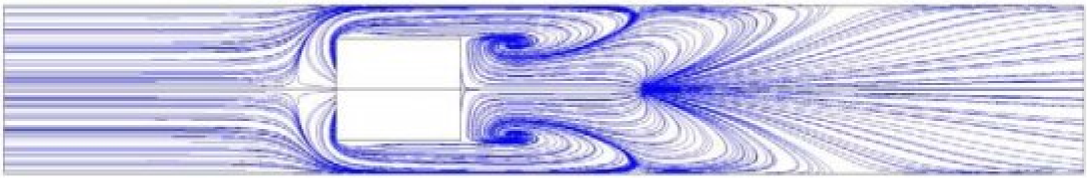

Figure 12: Figure 21 :

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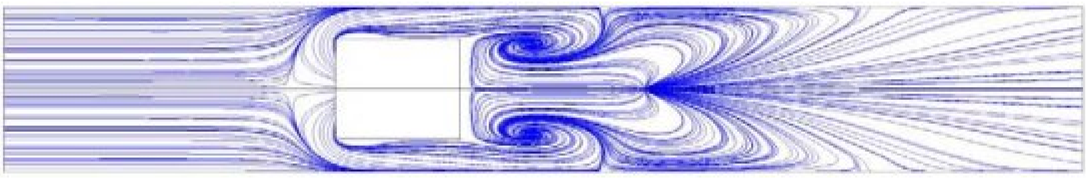

Figure 13:)Figure 23 :Figure 24 :

1

Re	L rec /H
100	1.19
150	1.60
200	2.00
250	2.43
300	2.84

(=) 0 0083 0 3602 rec L H . Re . + , (7)

Figure 14: Table 1 :

.1 Acknowledgements

The authors recognize the Universidade Tecnológica Federal do Paraná -UTFPR/Campus Ponta Grossa.

.2 ( )

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