THE EFFECT OF THE THICKNESS OF THE PLATES IN LAMINAR FORCED CONVECTION IN HORIZONTAL CHANNEL WITH HEAT GENERATING BOARDS COOLED BY NANOFLUIDS

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Abstract. *This paper describes the study of the behaviour of two-dimensional and parallel horizontal flat plates subjected to laminar forced convection of nanofluid water/copper. The governing equations are solved numerically using the finite volume technique and the algorithm SIMPLE. After the simulations the temperature and velocity fields were obtained for the following variables: plate thickness and conductivities, spacing between boards and inlet velocities.*

Keywords*: Forced Convection, Numerical Methods, Nanofluids.*

1. INTRODUCTION

The process of heat transfer which gets along with conduction and convection is extremely important due to its applications in various areas of engineering, such as cooling of electronic equipments and turbine blades, aerospace, nuclear reactors, heat exchangers, among others. The calculation procedure basically consists in subdivide the physical domain into two, usually the fluid and solid region. It shoud be noted that the thermophysical properties of the fluid has an important role in the mechanism of heat removal. On the other hand, the internal heat generation of the solid region is an interesting subject involving heat transfer. The question that covers the maintenance of the temperature in a safe level from integrated circuits is one direct application of that topic because the increasing interest in recent years owing to miniaturization of electronic components, which enhances the dissipated power per unity of area (Bechtold et al. (2005)).

The literature shows that the efforts are directed to analytical and empirical models to solve the problem described. Dorfman (2010) made a complete review from the principal methods for simple two-dimensional flow, where the form of the velocity flow field shall be assumed. More recently semi-analytical treatments are used to described the phenomena, such as the works of Rashidi and Erfani (2009), which involves the homotopy analysis method, and He (2010), regarding the homotopy perturbation method. Khaleghi et al. (2007), in turn, applied the variational iteration method to solve nonlinear heat transfer equations, while Zhou (1986) resorted the differential transformed method. Bautista and Mendez (2006) investigated conjugate heat transfer analysis in a discrete heat source with internal heat generation in a rectangular-channel laminar flow. Olarewaju et al. (2012) inquired the internal heat generation effect on thermal boundary layer with a convective surface boundary condition over a flat plate. They accomplished that the plate surface temperatures exceed the temperature of the fluid on the lower surface of the plate because of strong internal heat generation. None of the articles cited so far have worked with nanofluids. These suspended nanoparticles in a fluid have been widely studied due to your verified enhancement on thermal condutctivity of the fluids. The concentration of the particles in volume should be less than 1% and size of the particles between 1 and 100 nanometers. The assumption of the nanofluids behaving as a single-phase fluids is adopted in computational fluid dynamics. The term nanofluid was first used by Choi and Eastman (1995), who proposed the suspension of metal nanoparticles in a base fluid, like water, ethylene-glycol and oil. Other kinds of particles may be oxids, carbides and nanotubes. Reddy et al. (2015) studied the influence of viscous dissipation on natural convection boundary layer flow and heat transfer of different nanofluids along a vertical plate with internal heat generation. The work of Rana and Bahrgava (2011) investigated the heat transfer enhancement in mixed convection flow of nanofluids along a vertical plate with a heat source/sink with a finite element method. Sidik et al. (2014) presented a summary of the recent developments in research on the synthesis and characterization of stationary nanofluids and. The main challenges observed for them are the long term stability of nanoparticles dispersion, increased pressure drop and pumping power, nanofluids thermal performance in turbulent flow and fully developed region, etc. Alawi et al. (2014) analyzed the natural convection heat transfer in horizontal concentric annulus between outer cylinder and inner flat tube using nanofluids.

This work has a main goal to obtain the Nusselt numbers of an arrangement of horizontal parallel plates, with constant internal heat generation, for a range of values of distance between boards, inlet velocities, plate conductivity and thickness. Nanofluids Water/Cupper with 1% concentration of particles in volume and laminar flow regime were used. The governing equations were approximated numerically using the technique of finite volume (Patankar (1980)). The conjugate heat transfer was solved by using a domain that includes the fluid a solid region, where the viscosity of the solid region was set equal to a very large number.

2. PROBLEM FORMULATION

The geometry analyzed in this work is shown in Fig. 1. The dotted region corresponds to the domain used at work. The gray part represents the plate area. The physical properties of the nanofluids are assumed to be constant.

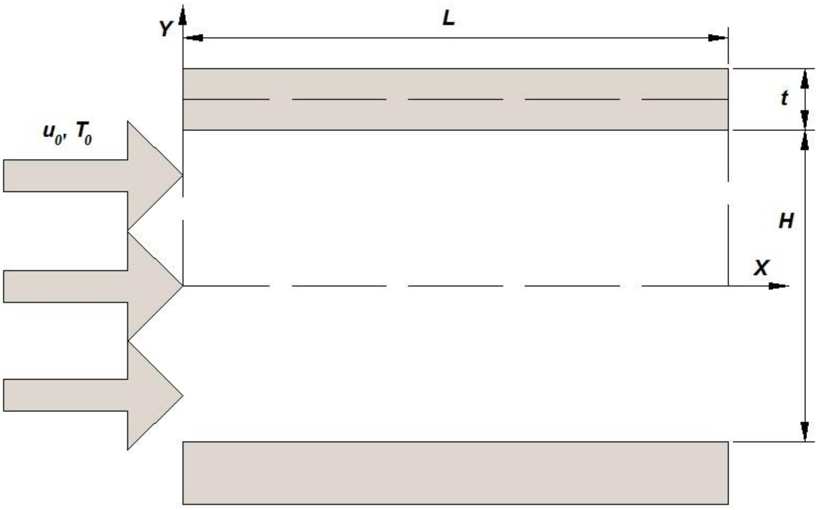

Figure 1. Geometry and Computational Domain.

The governing equations in a dimensionless form for steady-state laminar, incompressible and two-dimensional fluid flow are:

Continuity Equation

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

Momentum Equation

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\frac{1}{Re_L}\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right) \tag{2}$$

$$U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\frac{1}{Re_L}\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right) \tag{3}$$

Energy Equation

$$U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}=\frac{1}{Re_L\cdot Pr}\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right) \tag{4}$$

Energy Equation in the solid region

$$R\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right)+1=0 \tag{5}$$

And the dimensionless parameters defined as follow:

$$X = x/L;\quad Y = y/L;\quad U = u/u_o;$$
$$V = v/u_o;\quad P = p/\rho_{nf}\cdot u_o^2;\quad \theta = k_{nf}(T - T_o)/q'''L^2;\qquad (6)$$
$$Pr = \nu_{nf}/\alpha_{nf};\quad Re_L = u_o\cdot L/\nu_{nf};\quad R = k_w/k_{nf}$$

Where: u, v – velocities (m/s); p – pressure (N/m^2); T – temperature (K); x – horizontal coordinate (m); y – vertical coordinate (m); L – channel length (m); H – distance between plates (m); t – plate thickness (m); ρ – density (kg/m^3); k – thermal conductivity (W/mK); P – dimensionless pressure; R – conductivity ratio; Re_L – Reynolds Number; Pr – Prandtl Number; q''' – internal heat generation of the plates (W/m^3); α – thermal diffusivity (m^2/s); ν – kinematic viscosity (m^2/s); μ – dynamic viscosity (Pa·s); C_p – specif heat (J/kgK); Subscripts: o – inlet; w – wall; nf – nanofluid.

The boundary conditions are shown in Fig. 2.

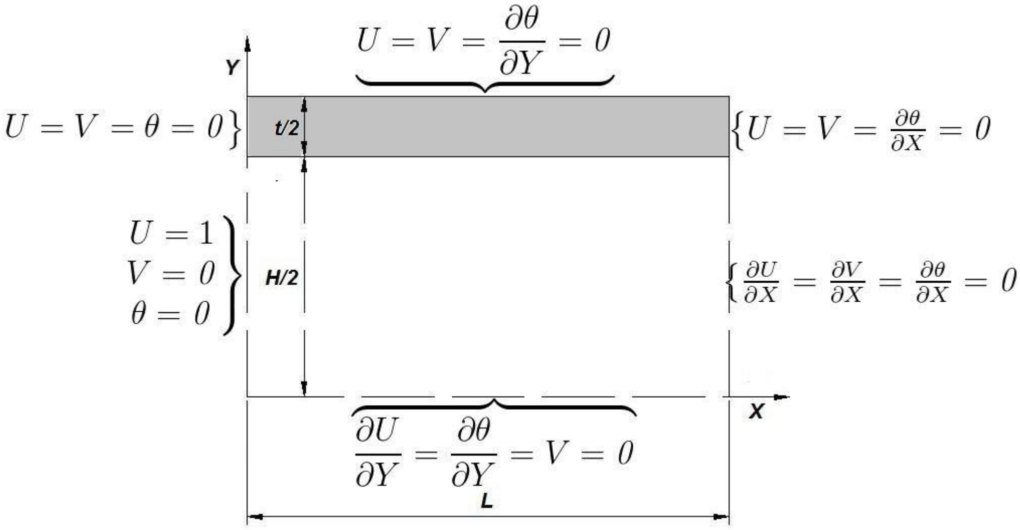

Figure 2. Boundary Conditions.

The physical properties of the nanofluids are obtained by the following equations, and Tab. 1 summarizes the values used in the present work. Hence:

$$\mu = \mu_{bf}(1 + 2.5\phi + 6.2\phi^2)\qquad (7)$$

$$\rho = (1 - \phi)\rho_{bf} + \phi\rho_{np}\qquad (8)$$

$$C_p = \left[(1 - \phi)\rho_{bf}C_p + \phi\rho_{np}C_{p_{np}}\right]/\left[(1 - \phi)\rho_{bf}C_{p_{bf}} + \phi\rho_{np}\right]\qquad (9)$$

$$k = k_{bf}(11.6\phi^2 + 9.6\phi + 1)\qquad (10)$$

The meaning of the following subscripts are: bf – base fluid; np – nanoparticles. The symbol ϕ denotes the volume concentration of the nanoparticles. The dynamic viscosity correlation (Eq. (7)) was proposed by Batchelor (1977), while Eqs. (8) and (9) were obtained from Zhou et al. (2010). The thermal conductivity may be found in the work of Wang et al. (2010), which corresponds to a fit of the experimental data. The nanofluid is hypothetically considered as a single phase fluid.

Table 1. The Physical Properties of Base Fluid, Nanoparticles and Nanofluids.

Properties	Fluid Base (H_2O)	Nanoparticles (Cu)	Nanofluids (H_2O/Cu) 1%
k	0.604	400	0.663
C_p	4179	383	3863.3
ρ	997.1	8954	1076.7
μ	$1.0\cdot10^{-3}$	-	1.0257

The Local Nusselt Number is defined as follow:

$$Nu_x = h\cdot D_h/k_{nf} = q'''\cdot t\cdot D_h/k_{nf}\left(T_{x,s} - T_b\right)\qquad (11)$$

Where D_h is the Hydraulic Diameter and the subscript s denotes the plate surface and b is the bulk temperature, expressed as:

$$T_b = \int uTdy / \int udy \tag{12}$$

Finally, the dimensionless form of Nusselt Number is:

$$Nu_X = 2(t/L)(H/L)/\left(\theta_{X,s} - \theta_b\right) \tag{13}$$

The average Nusselt Number is calculated as follows:

$$\overline{Nu}_X = \int Nu_X dX / \int dX \tag{14}$$

3. NUMERICAL PROCEDURE

The governing equations are solved using the finite volume method developed by Patankar (1980) with staggered grid for velocity field while the scalar quantities were stored in the center of these volumes. The convective terms of momentum and energy are approximated by Power-Law scheme. The SIMPLE algorithm is applied to solve the coupling of pressure-velocity. The Tri-Diagonal Matrix Algorithm (TDMA) method is employed to figure out the discretized set of equations and the programming language of the code is FORTRAN. The adopted convergence criterion is:

$$\left|\frac{\varphi^{k+1} - \varphi^k}{\varphi^k}\right| \leq 10^{-6} \tag{15}$$

Where φ performs U, V, θ and the maximum residual in the continuity equation and K the current iteration. The machine used to perform all of the calculations is a microcomputer with Intel® i7 (2.2 GHz) processor and it has 8 GB of RAM.

3.1 Grid independence Study

Two uniform grids were analyzed in this work: 84 × 402 ($Y \times X$) e 166 × 802 ($Y \times X$). The problem choosed was to obtain the maximum temperature of the channel, Average Nusselt Number and pressure at the centerline of the channel ($Y = 0$, $X = 1$) for $H/L = 0.25$, $Re_L = 100$, $Pr = 6$, R = 50 and $t/L = 0.01$. The results are summarized at Tab. 2, where it is clear that the results for 84 × 402 points provides good results compared with the 166 × 802 with less computational effort.

Table 2. Results for the two grid points evaluated.

Grid	θ_{max}	P (X = 1, Y = 0)	Nu_{av}	Time (hr)
84 × 402	$1.422 \cdot 10^{-3}$	0.508	9.125	3
166 × 802	$1.415 \cdot 10^{-3}$	0.507	9,219	9

The mesh 84 × 402 has 32800 control volumes, 400 in the horizontal direction and 82 the vertical one. The length of each control volume, both in X and Y direction (*XCV* and *YCV*), is 0.0025. This ratio is used to generate H/L lengths equals to 0.2 and 0.3, providing, respectively, 44 and 64 points in vertical direction, while in horizontal direction the number of points are kept in 402. The plate region has, in all cases, two control volumes.

3.2 Code validation

The results of the computational code were compared with the numerical results of Silva and Ganzarolli (2002), which obtained maximum dimensionless temperature profiles as a function of a dimensionless pressure head N defined as:

$$N = \left(\frac{\Delta p L^2}{Pr \cdot \mu \cdot \alpha}\right) \tag{16}$$

Results are shown in Fig. 3 for the dimensionless distance between the plates equal to $H/L = 0.05$.The plate thickness adopted was $t/L = 0.006$ and $R = 30$. It was varied the Reynolds Number to obtain the different pressure values. The fluid in this validation has a Prandtl Number equal to 0.71.There is good agreement between the results of the present work and the ones of Silva and Ganzarolli (2002).

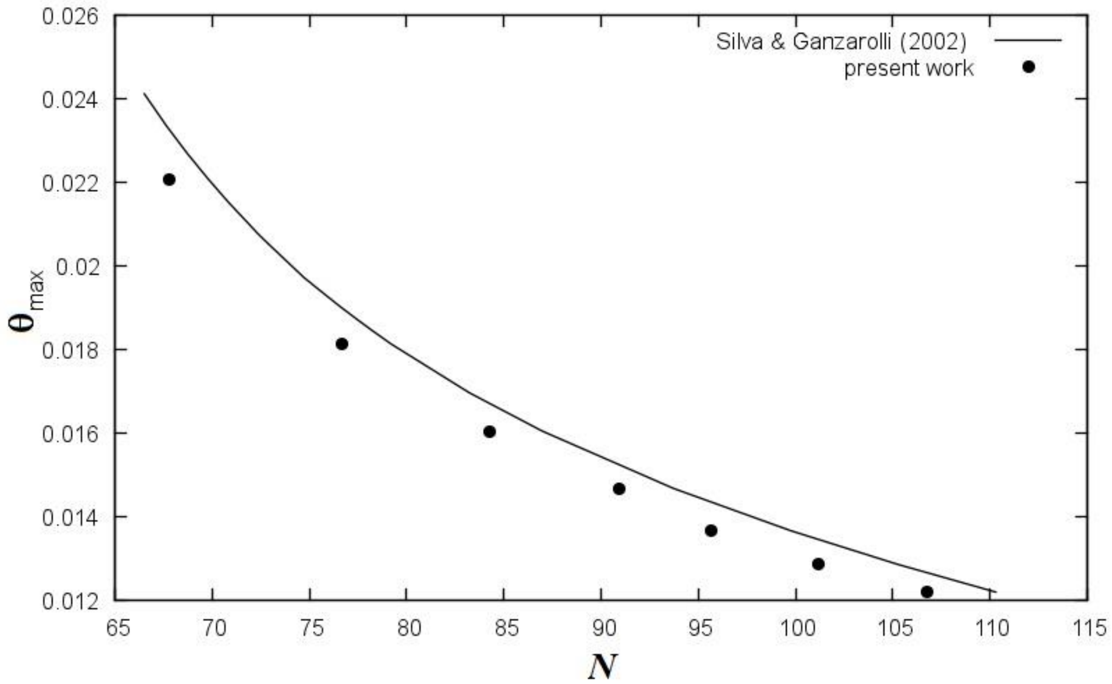

Figure 3. Comparison between numerical results of maximum temperature in function of *N*. (*Pr* = 0.7, *H/L* = 0.05, *t/L* = 0.006, *R* = 30)

4. RESULTS AND DISCUSSION

A numerical analysis was accomplished on laminar forced convection in a horizontal channel. The distances between boards *(H/L)* chosen for the study were: 0.2, 0.3 and 0.4. The $(H/L)Re_L$ values used in simulation corresponding to 10, 100 and 1000. This dimensionless group represents The Reynolds Number evaluated as a function of the distance between plates. For each distance *H/L* was calculated the local Nusselt Number. The dimensionless thickness of the plate *(t/L)* was adopted as 0.01 and 0.02. Two values of *R* were specified: 50 and 150. The Prandtl Number of nanofluid Water/Copper is equal to 6.

The conjugate problem (solid/fluid region) is adopted to solve the governing equations, and the temperature profile from energy equation is obtained separately from mass and momentum equations, after the determination of the velocity and pressure fields.

Figure 4 is the plot of Nusselt Number along the dimensionless length of the channel for the variables studied in the present work. The graphics show that the Nusselt Number is very high at the beginning of the entrance region of the channel and decreases strongly. The charts (a), (b) and (c) are from the dimensionless plate thickness equal to 0.01, while the others ones are from *t/L* equal to 0.02. The first consideration is that by increasing The Reynolds Number also increases The Local Nusselt Number value, regardless of the distance *H/L*. It indicates the temperature decreases with increasing mass flow within the channel. This effect is also observed in The Average Nusselt Number (Tab. 3). When analyzing the effect of increase in conductivity heat transfer a decrease in both Local and Average Nusselt Number is observed, when the plate thickness and Re_H are kept constant. The result is more visible near the channel entrance for The Local Nusselt Number. From a certain value of *X*, The Local Nusselt Number for higher *R* becomes slightly larger. The higher The Reynolds Number, the greater the value of *X* where the inversion occurs. The effect of the growth of the plate thickness causes an increase in Local Nusselt Number. The aumengtation, which occurs in the cited parameter, is not proportional to the increase in plate thickness. The variation is related to the value of the internal resistance that occurs with changes in board thickness.

Table 3 presents the results of Average Nusselt Number for all variables involved in this study. The raising of the mass flow (for a fixed *H/L*) it provides the increase of Nusselt Number, and the same behavior is observed when increasing the distance between the plates (for a fixed Re_H) . As mentioned above, the increase in *R* factor indicates a diminution in Average Nusselt Number, and it can be explained since the increase in conductivity k_s leads to lower conduction resistance. However, to change the value of *R* from 50 to 150 does not match the values of Nusselt Number are three times smaller. The variation of Nusselt Number, however, does not happen linearly with Re_H. By comparing the variation of Average Nusselt Number to the thickness of the plate it is observed that there is an augmentation from the first to enlarge the latter, when fixed *H/L* and Re_H.

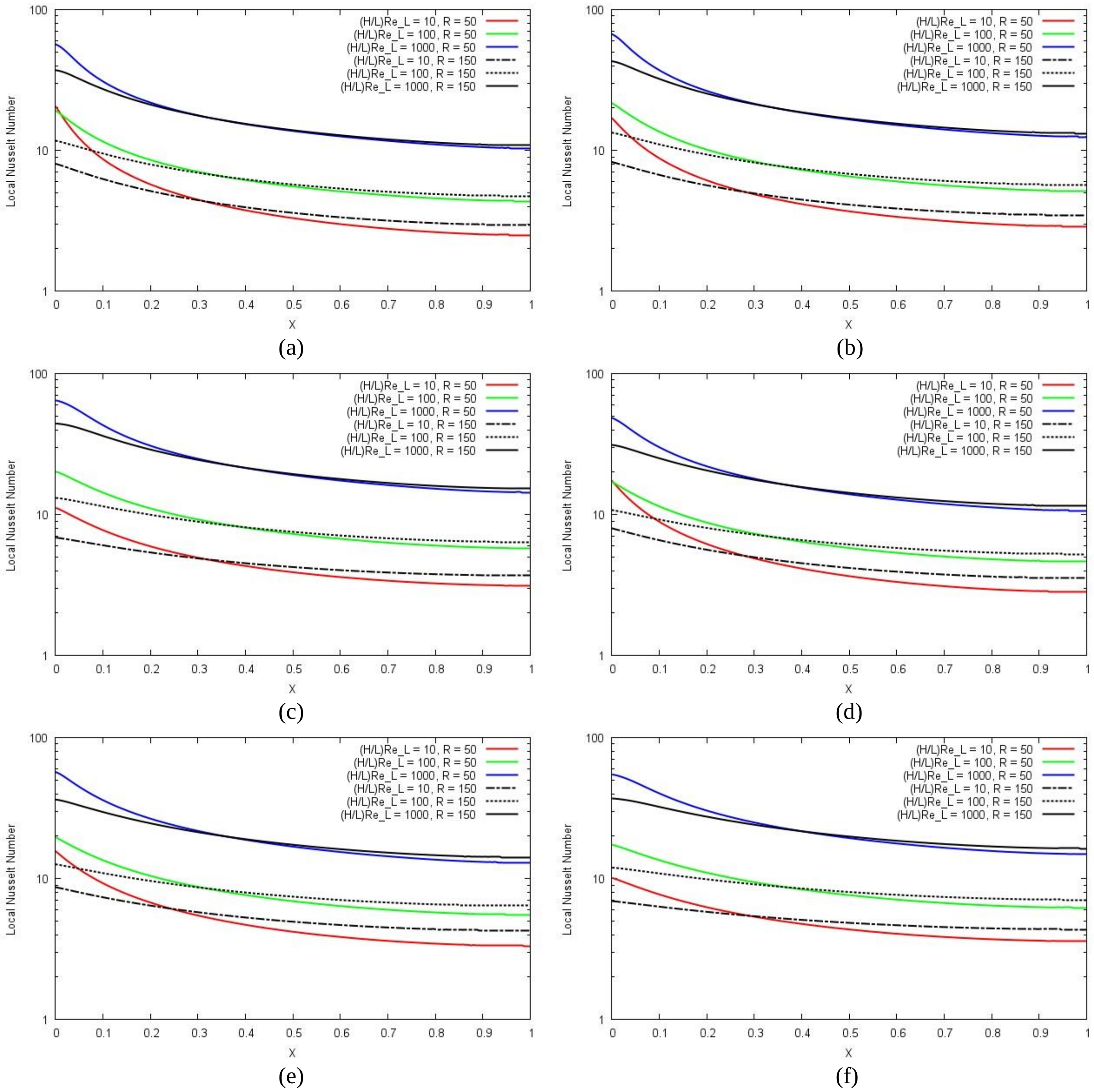

Figure 4. Local Nusselt Number: (a) *H/L* = 0.20, *t/L* = 0.01; (b) *H/L* = 0.30, *t/L* = 0.01; (c) *H/L* = 0.40, *t/L* = 0.01; (d) *H/L* = 0.20, *t/L* = 0.02; (e) *H/L* = 0.30, *t/L* = 0.02; (f) *H/L* = 0.40, *t/L* = 0.02.

Table 3. Average Nusselt Number.

R = 50							
t/L = 0.01				*t/L* = 0.02			
$(H/L)Re_L$	*H/L* = 0.20	*H/L* = 0.30	*H/L* = 0.40	$(H/L)Re_L$	*H/L* = 0.20	*H/L* = 0.30	*H/L* = 0.40
10	4.644	4.715	4.888	10	4.890	5.040	5.337
100	6.861	8.058	8.739	100	6.989	8.292	8.798
1000	17.409	20.953	23.582	1000	17.630	21.174	24.110
R = 150							
t/L = 0.01				*t/L* = 0.02			
$(H/L)Re_L$	*H/L* = 0.20	*H/L* = 0.30	*H/L* = 0.40	$(H/L)Re_L$	*H/L* = 0.20	*H/L* = 0.30	*H/L* = 0.40
10	4.128	4.622	4.576	10	4.645	5.407	5.114
100	6.451	7.615	8.255	100	6.709	8.087	8.529
1000	16.346	19.719	22.139	1000	16.596	19.881	22.459

5. CONCLUSIONS

It was obtained numerically values of the Nusselt Number (Local and Average) for a laminar forced convection of Water/Copper nanofluid, 1% of volume concentration of the particles, in a two-dimensional horizontal channel composed by flat plates with internal heat generation. Results shown that the increase in Reynolds Number, for a fixed plates spacing, denotes a growth of Nusselt Number. It was noticed that the increment of R factor corresponds to a reduction of The Nusselt Number. The augmentation of the plate thickness results in an increase in The Nusselt Number, if it was fixed the spaced between the boards (varying the Reynolds Number) or if it was fixed the Reynolds Number (varying the plate thickness).

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