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COMPARATIVE NUMERICAL AND EXPERIMENTAL RESULTS FOR THE CONJUGATE COOLING OF A DISCRETE HEATER IN A DUCT

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Abstract. *Forced airflow is the most used procedure for the cooling of electronic components mounted on circuit boards. Conductive boards may be used to enhance the components' cooling by conjugate forced convection-conduction. This cooling may be conveniently described by invariant conjugate coefficients expressed in matrix form. A numerical investigation was preformed to obtain these coefficients and to compare the results with previously obtained experimental values. The investigated configuration consisted of a protruding Aluminum heater mounted on the lower horizontal wall of a rectangular duct and cooled by forced airflow. Three distinct lower walls or substrate plates were considered in this numerical investigation: Adiabatic, Plexiglas and Aluminum. Uniform airflow velocity and temperature were assumed at the duct inlet and the heater was isothermal, located near the duct entrance. The conservation equations were solved with a zero-equations turbulence model in a single domain comprising the solid and fluid regions, ie, the airflow, the substrate wall and the heater. The numerical results for the convective and for the conjugate heater cooling were expressed as functions of the duct airflow Reynolds number in the range from 2,000 to 6,000. Their comparison with previous experimental results showed that the largest deviation was within 12%.*

Keywords: *conjugate cooling, invariant coefficients, numerical simulation*

1. INTRODUCTION

Electronic components assembled on circuit boards must operate below a maximum temperature specified by their manufacturers because otherwise they may cause failure of electronic equipments. According to Kraus and Bar-Cohen (1983), the components life cycle may be drastically reduced if they operate above their maximum values. Due to this problem, numerical and experimental models for a good thermal design of electronic equipment, circuit boards and components have been extensively studied. As the size of electronic components and equipment has been reduced, it has required more careful thermal design, usually employing heat transfer enhancement techniques. In many electronic equipments, circuit boards are closely spaced, constituting narrow channels convectively cooled by forced airflow. This lack of space may prevent the use of finned heat sinks attached to components with high heat fluxes. One alternative to enhance the cooling of these components is to use circuit boards as thermal conductors (Nakayama, 1997). In this case, the components are cooled by conjugate forced convection-conduction, comprising direct forced convection from their surfaces in contact with the airflow and conduction spreading through their contact with the board (substrate plate).

Moffat and co-authors (Moffat *et al*., 1985, Moffat and Anderson, 1990, Moffat, 1998 and 2004) developed an invariant descriptor for the forced convection cooling of electronic components mounted on circuit boards, the adiabatic heat transfer coefficient. The reference temperature associated to this coefficient is the heater adiabatic surface temperature (T_{ad}), which is defined as the heater equilibrium temperature when its power is turned off, while all the surrounding temperatures remain the same. To obtain this coefficient, they performed experiments considering the forced convection cooling of arrays of electrically heated blocks mounted on almost adiabatic substrate plates. Under these conditions, the electric power dissipation in the heaters is removed mostly by direct forced convection (q_{cv}) to the airflow. The experiments were performed with a single active heater at a time, so that

$$q_{cv} = h_{ad} A_h (T_h - T_{ad}) \qquad (1)$$

In Eq. (1) A_h is the heater surface area in contact with the airflow and T_h is the heater surface temperature. For low thermal conductivity substrate plates, q_{cv} may be well estimated from the heater power dissipation and Eq. (1) may be used to obtain the adiabatic heat transfer coefficient, h_{ad}. Once known as a function of the flow Reynolds number, this coefficient is invariant with the heater power dissipation. This is very convenient for the thermal design because it allows that Eq. (1) may be used to predict the heater temperature T_h under arbitrary heating conditions of the heaters array. In dimensionless form, h_{ad} may be expressed by the adiabatic Nusselt number Nu_{ad} as in Eq. (2).

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$$Nu_{ad} = \frac{h_{ad} L_h}{k} \tag{2}$$

In Eq. (2), L_h is the block heater edge along the flow and k is the air thermal conductivity. As the substrate plate thermal conductivity increases, conduction becomes an important path for the heater cooling, causing an enhanced heater cooling by conjugate forced convection-conduction. Thus, in this case, Eq. (1) alone will not be accurate enough to predict T_h because only a fraction of the heater power dissipation will be transferred directly to the airflow by convection. Nakayama (1997) reports several investigations of the literature related to this conjugate cooling. Davalath and Bayazitoglu (1987), Kim and Anand (1994 and 1995) investigated discrete protruding heaters mounted on a conductive substrate plate cooled by channel flow. They reported results for the conjugate cooling in terms of global thermal resistances. Convective cooling results were also presented by Nusselt numbers based either on the fluid temperature at the channel inlet or on its mean mixed value just upstream of the heater. Thereby their results depend on the power dissipation in the heaters. For this reason, their results are reported for uniform heating of the heaters' array.

Alves and Altemani (2012) developed an invariant descriptor for the conjugate cooling in the form of dimensionless conjugate coefficients g^+_{ij} relating the temperature of each heater of an array mounted on a conductive substrate plate to the power dissipation in all the heaters. They performed two-dimensional numerical simulations of the conjugate forced convection-conduction cooling of three protruding heaters mounted on the conductive lower wall of a parallel plates channel by forced laminar airflow. Considering an array of N heaters, the n^{th} heater average surface temperature ($T_{h,n}$) above the channel inlet flow temperature (T_0) was related to the power dissipation q_i in each heater by a N-square matrix of conjugate coefficients g^+_{ni} as indicated in Eq. (3).

$$\begin{bmatrix} \Delta T_1 \\ \Delta T_2 \\ \vdots \\ \Delta T_N \end{bmatrix} = \frac{1}{\dot{m} c_p} \begin{bmatrix} g^+_{11} & g^+_{12} & \cdots & g^+_{1N} \\ g^+_{21} & g^+_{22} & \cdots & g^+_{2N} \\ \vdots & \vdots & \ddots & \vdots \\ g^+_{N1} & g^+_{N2} & \cdots & g^+_{NN} \end{bmatrix} \begin{bmatrix} q_1 \\ q_2 \\ \vdots \\ q_N \end{bmatrix} \tag{3}$$

The channel flow rate is represented by $\dot{m}$ and c_p is the fluid specific heat. Under conditions of constant properties, flow rate and fixed geometry and materials, the conjugate coefficients are invariant with the power dissipation in the heaters array.

In the case where only a single heater is mounted on the substrate plate, the conjugate heat transfer may be described just by the dimensionless conjugate coefficient g^+_{11}, which can be obtained from Eq.(3) in the form

$$\dot{m} c_p (T_{h,1} - T_0) = g^+_{11} q_{cj} \tag{4}$$

The purpose of the present work was to perform numerical simulations of the conjugate cooling of a protruding heater mounted on the lower wall of a rectangular duct with the same configuration of the apparatus reported by Loiola and Altemani (2012) and to compare the numerical and experimental results. Besides the Aluminum and Plexiglas plates reported in the experimental work, simulations were also performed for an adiabatic plate in order to obtain numerical results for the heater convective cooling. The simulations were three-dimensional, steady state, assuming uniform airflow velocity and temperature at the rectangular duct inlet. The results were expressed by the adiabatic Nusselt number Nu_{ad} and by a single conjugate coefficient g^+_{11}. Both are invariant with the heater power dissipation, but dependent on the channel flow Reynolds number Re_D, which was based on the duct hydraulic diameter. The Reynolds number was defined as function of the airflow mass rate $(\dot{m})$, the dynamic viscosity of the air (μ) and the duct cross section perimeter (p_w) as indicated in Eq.(5).

$$Re_D = \frac{4\dot{m}}{\mu\, p_w} \tag{5}$$

2. MODEL DESCRIPTION

The numerical simulations were performed to evaluate the conjugate cooling of a heater in a rectangular duct by forced airflow, as indicated in Fig. 1. A protruding heater was mounted on the duct lower horizontal wall, centered at 25 mm from the duct entrance. The simulations were performed considering uniform airflow velocity and temperature (T_i=18°C) at the duct inlet, and the heater was isothermal (T_h = 22°C). Thus, all air properties were taken at 20°C. The duct lower wall was 2mm thick and the simulations were performed considering two distinct lower walls, associated to Aluminum and Plexiglas, as in the experimental tests of Loiola and Altemani (2012), and also an adiabatic surface used to evaluate the adiabatic Nusselt number. A perfect thermal contact was assumed at the interface of the heater and both

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conductive lower walls. The remaining duct surfaces were considered smooth and adiabatic. The duct cross section was $W = 0.160$ m and $H = 0.020$ m, with a length $L = 0.200$ m, as in Fig. 1, while the heater had a square base with an edge $a = 0.050$ m and height $h = 6$ mm. Considering the symmetry plane shown in Fig. 1 of the duct and heater, the simulations were performed for only half of the duct, in order to save computer storage memory and processing time.

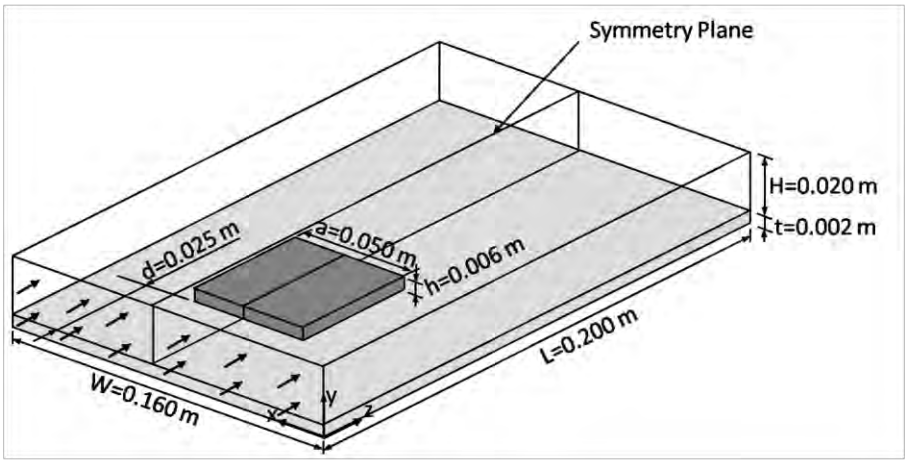

Figure 1. Diagram of the heater in the rectangular duct

The software PHOENICS (CHAM, 2009) was employed for the three dimensional simulations of the duct flow and heat transfer. Turbulent flow was solved by the Reynolds-averaged conservation equations using the zero-equation turbulence model L-VEL (Spalding *et al.*, 1996) included in the software. A non-uniform numerical grid was used, more refined close to the duct walls and the heater. After several numerical tests, the reported results were obtained considering 180 grid points in the axial direction and 80x70 grid points in the numerical domain, ie, the duct length and half the duct cross-section, as indicated in Fig. 1. Although the duct used in the experiments was 0,30 m long, in the simulations the duct length was only 0.20 m because it was verified after initial numerical results that the recirculation behind the heater did not extend to 0.20 m duct length.

The experimental results of Loiola and Altemani (2012) were obtained in laboratory with a rectangular duct containing a protruding Aluminum heater mounted on the lower horizontal wall (substrate plate). The rectangular duct had the same cross section of the numerical tests but a longer length, as previously explained. Two distinct substrate plates with identical dimensions were used as the duct lower wall - one was made of Plexiglas and the other of Aluminum. The remaining lateral and top walls of the duct were very smooth, made of Plexiglas. The heater was mounted on the Plexiglas substrate with reduced conductive thermal contact in order to enhance the convective cooling from the heater to the airflow in order to obtain the adiabatic Nusselt number through Eq. (1) and (2). On the other hand, a good thermal contact was sought for the heater with the Aluminum substrate, to enhance the conjugate cooling and the conjugate coefficient for both substrates was obtained from Eq. (4).

3. NUMERICAL PROCEDURE

For the numerical domain, encompassing only half of the duct solid and fluid regions presented in Fig. 1, the Reynolds averaged mass, momentum and energy conservation equations were solved in Cartesian three dimensional coordinates. The air properties were assumed constant at 20ºC and the Aluminum and Plexiglas properties were obtained from tabulated values. The conservation equations were expressed by

$$\frac{\partial V_i}{\partial x_i} = 0 \tag{6}$$

$$\frac{\partial V_i V_j}{\partial x_j} = -\frac{1}{\rho}\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left[(\nu + \varepsilon_M)\left(\frac{\partial V_i}{\partial x_j} + \frac{\partial V_j}{\partial x_i}\right)\right] \tag{7}$$

$$\frac{\partial (V_j T)}{\partial x_j} = \frac{\partial}{\partial x_j}\left[(\alpha + \varepsilon_H)\frac{\partial T}{\partial x_j}\right] \tag{8}$$

The Cartesian coordinates are represented by x_i and x_j, the velocity components in the coordinate directions are indicated by V_i and the flow pressure and temperature respectively by P and T. The fluid properties are the density ρ, the kinematic viscosity ν, and the thermal diffusivity α. The turbulent diffusivities ε_M and ε_H were obtained from the L-VEL turbulence model included in the software PHOENICS, as described by Spalding *et al.*, (1996). The boundary conditions comprise uniform airflow velocity and temperature at the duct inlet and the heater was assumed isothermal. The lower duct wall was included in the numerical domain in order to evaluate the conjugate heater cooling, with its lower surface assumed adiabatic. The remaining lateral and top duct walls were adiabatic. The simulations were performed in a microcomputer (Intel Core i7-2600, 3.4 GHz with 8GB RAM), demanding about seven hours and 5000 iterations to reach convergence for each case. The numerical results were obtained from these simulations and they will be compared to the previous experimental results.

3.1 Numerical Test

In order to provide confidence to adequate usage of the computational package PHOENICS, an initial simulation was performed and the results were compared to a solution from the literature. The case considered was the thermal entrance region of developed airflow in a parallel plates channel with uniform heat flux on one wall while the other was adiabatic. The obtained numerical results were compared to analytical results presented by Kays and Crawford, (1993).

Figure 2 shows the geometry used for the numerical test, with the indicated channel height and length. The lower channel wall had a uniform heat flux equal to 500 W/m^2 while the upper wall was adiabatic. The airflow was developed with an average velocity w_m = 0.193 m/s and its inlet temperature was T_i = 20°C. The Reynolds number based on the channel hydraulic diameter was Re_D = 500, corresponding to the laminar regime.

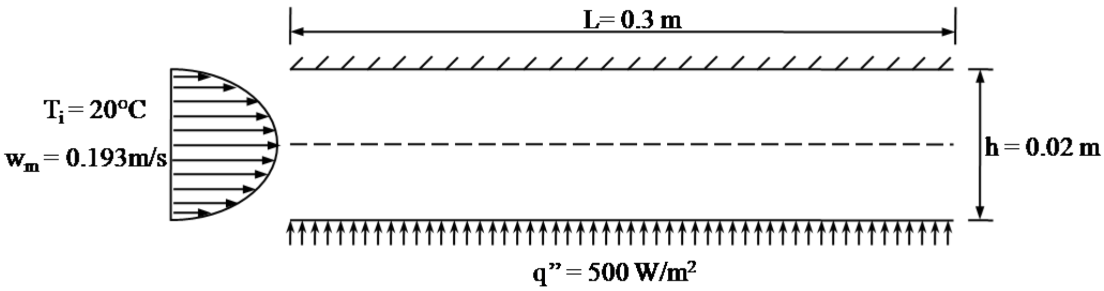

Figure 2. Configuration employed for the numerical test

The results were obtained for a uniform numerical grid in each direction, with 70 points along the channel height and 150 in the flow direction. The numerical and the analytical results for the local Nusselt number along the active wall (Nu_z), based on the channel hydraulic diameter, are presented in Fig. 3, indicating that they agreed within 1.1%.

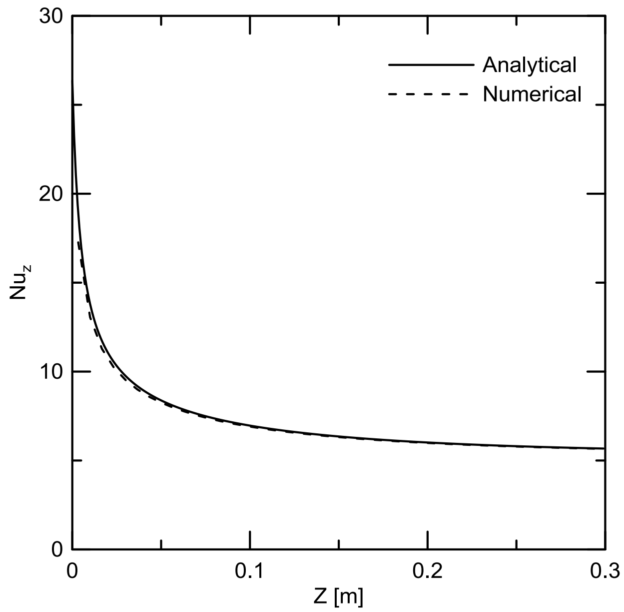

Figure 3. Analytical and numerical local Nusselt number

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4. RESULTS AND DISCUSSION

The numerical simulations performed with the adiabatic substrate plate had the purpose to obtain the adiabatic Nusselt number as a function of the duct airflow Reynolds number. In the previous experimental investigation, the adiabatic Nusselt number was evaluated from tests with the Plexiglas substrate, where the heater was mounted with a low conductive thermal contact in order to promote the convective cooling. The simulations performed for the Plexiglas and the Aluminum substrates were used to evaluate just the conjugate coefficient g^{+}_{11}. In both investigations, numerical and experimental, the average airflow velocity in the duct ranged from about 1 m/s to 3 m/s, corresponding to Reynolds number from 2,000 to 6,000.

Adiabatic Substrate Plate. The numerical results for Nu_{ad} were obtained considering that the heater was mounted on a perfectly insulated substrate plate, while the experimental results were obtained from tests with the heater temperature at 40°C mounted on the Plexiglas substrate. Both results are presented in Fig. 4 as functions of Re_D and they were grouped in two sets indicative of the flow regime. In the laminar range, the numerical results are almost parallel but about 2 % above the experimental results. In the turbulent regime, the experimental distribution of Nu_{ad} presents a steeper function of the Reynolds number than the numerical results, but they agreed within 4 %.

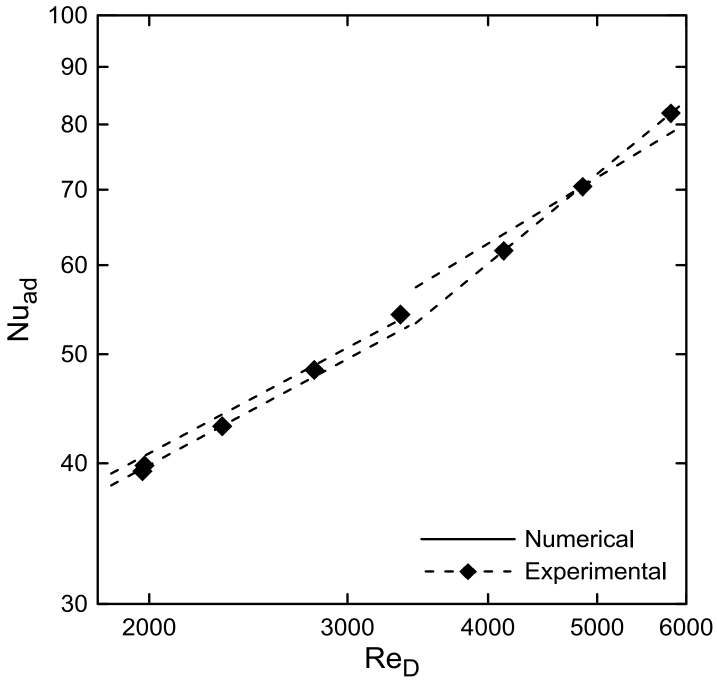

Figure 4. Adiabatic Nusselt number for the adiabatic/Plexiglas substrate.

Plexiglas Substrate Plate. This substrate was selected due to its relatively low thermal conductivity, as compared to the Aluminum substrate. In perspective, the heater mounted on the Plexiglas substrate will be cooled mostly by convection and the power dissipation needed for it to attain a fixed temperature will be smaller than that for the Aluminum substrate under the same flow conditions. As a consequence, the conjugate coefficients must be larger for the Plexiglas substrate than for the Aluminum substrate. These trends can be observed from the results of Figs. (5) and (6). For these two substrate plates, the simulations were performed assuming a perfect thermal contact at the heater-substrate interface. For the Plexiglas plate, the results for g^{+}_{11} are presented in Fig. (5). In the Re_D range indicative of laminar flow regime the simulation results were within 5% above and almost parallel to the experimental results. In the turbulent flow region, the simulations presented a slope with Re_D similar to that of the laminar regime, while the experiments indicated a pronounced decrease of the slope, as compared to the laminar range of Re_D. The relative largest deviation of the numerical and the experimental results was around 12%, attained for the largest Re_D, about 5,800.

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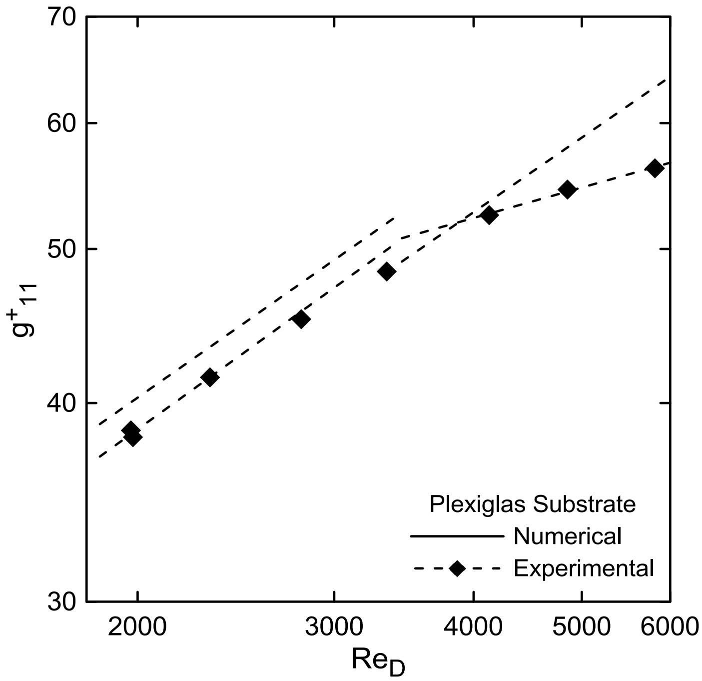

Figure 5. Conjugate coefficient for the Plexiglas substrate.

Aluminum Substrate Plate. This substrate plate was chosen to enhance the conjugate forced convection-conduction heater cooling. In this case, the heater-substrate thermal contact for the simulations was perfect and for the experimental investigation a good thermal contact was provided. Thus, a better agreement should be expected between the simulation and the experimental results for the Aluminum substrate than for the Plexiglas substrate. As indicated in Fig. 6, the numerical and the experimental results for this substrate are relatively closer to each other for both flow regimes although their slopes with Re_D are not quite the same, ie, they are not parallel to each other. The largest relative deviation of the experiments and simulations is around 5% in the investigated Re_D range.

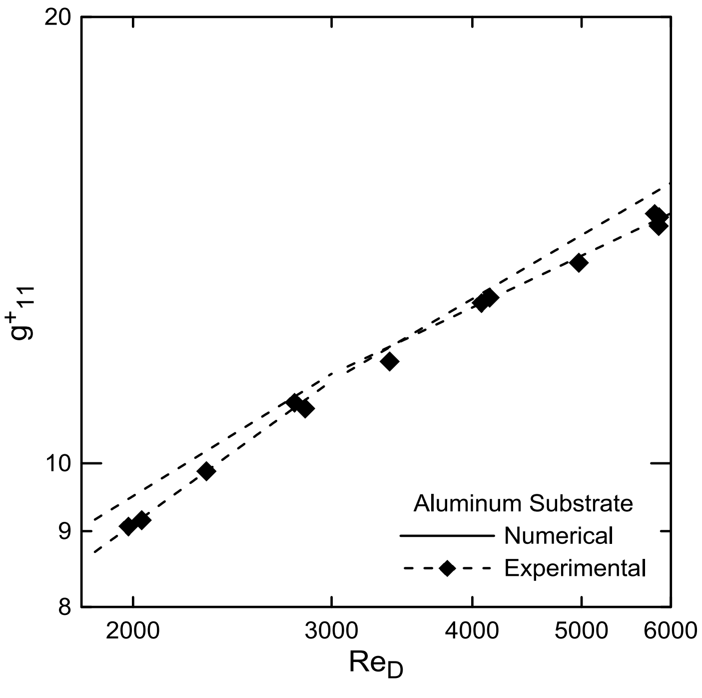

Figure 6. Conjugate coefficient for the Aluminum substrate.

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5. CONCLUSIONS

Comparisons were made between numerical and experimental results for two dimensionless coefficients (Nu_{ad} and g^+_{11}) related to the forced airflow cooling of a protruding heater mounted on the lower horizontal wall of a rectangular duct. For the numerical simulations, a relatively simple turbulence model was employed and the results were obtained under steady state conditions for the time averaged Reynolds conservation equations. Care was taken to extend the calculation domain to a distance downstream the protruding heater beyond the flow separation and recirculation zone.

The simulations to obtain the adiabatic Nusselt number Nu_{ad} were performed with an adiabatic substrate plate, to restrict the heat transfer to convection only. On the other hand, the experimental values were obtained with the heater mounted on the Plexiglas substrate plate with low conductive thermal contact, in order to reduce its conductive cooling. The indicated numerical and experimental results for Nu_{ad} agreed within 4 %.

The simulations to obtain the conjugate coefficient g^+_{11} were performed considering both the Plexiglas and the Aluminum substrate plates with a perfect heater-substrate plate thermal contact. The experimental results were obtained with a good heater thermal contact with the Aluminum substrate, but with a minimized thermal contact with the Plexiglas substrate plate. Thus, the simulations to obtain the conjugate coefficient g^+_{11} presented a better agreement with the experimental data for the Aluminum substrate than for the Plexiglas substrate. The largest deviations between the numerical and the experimental results for g^+_{11} were around 5 % for the Aluminum substrate and 12 % for the Plexiglas substrate. Comparatively, the conjugate coefficients for the Aluminum substrate are quite smaller than those for the Plexiglas substrate. This indicates that under the same flow conditions and power dissipation, the heater temperature will be noticeably smaller when mounted on the Aluminum substrate.

The numerical and the experimental results for both invariant coefficients were obtained for quite distinct heater and flow temperatures, so that the similar results indicate their invariant nature with the heater power dissipation. This is a very convenient feature from the thermal design point of view.

6. ACKNOWLEDGEMENTS

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