23rd ABCM International Congress of Mechanical Engineering
December 6-11, 2015, Rio de Janeiro, RJ, Brazil

EVALUATION OF THE CONJUGATE HEAT TRANSFER OF TWO HEATERS IN A TOP INLET DUCT

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Abstract. *The present work reports an experimental investigation of the conjugate forced convection-conduction cooling of two protruding aluminum heaters mounted on the horizontal wall of a rectangular duct. The heating was obtained by independent electric power dissipation in each heater and the cooling was performed by forced airflow with a top inlet in the duct. The heaters were cooled mainly by direct forced convection to the airflow at the surfaces in contact with the flow and by conduction to the duct wall at their interface. The conjugate heat transfer from the heaters was described by means of a matrix* G^+ *of dimensionless conjugate coefficients* g^+_{ni}*, which are invariant with the conjugate heat loss from each heater. This was shown by several experimental tests with distinct air flow rates and power dissipation in both heaters. The tests were performed using an apparatus consisting of two symmetrical ducts, each with a top inlet, separated by a common wall, denominated the substrate plate. Two heaters were mounted on each side of the substrate plate and the conjugate coefficients were evaluated for each heater, using one active heater per duct at a time. In order to explore the effect of the conductive heat losses from the heaters on the conjugate coefficients, two substrate plates were employed in the experimental tests: one was made of aluminum and the other of acrylic. For the acrylic plate, the heaters' conductive losses were comparatively small and thus the convective losses were also evaluated and described by means of the adiabatic heat transfer coefficient,* h_{ad}*. This coefficient is invariant with the rate of convective heat loss from each heater to the airflow. The experimental tests were performed under steady state conditions for a range of the Reynolds number based on the duct hydraulic diameter from 1200 to 3000.*

Keywords: *conjugate heat transfer, invariant descriptors, experimental investigation*

1. INTRODUCTION

Electronic components mounted on a circuit board require careful thermal design. Such components dissipate electric power as heat by ohmic effect, leading to an increase in temperature. If their temperate goes above a tolerable limit, the life cycle of such components may be greatly reduced (Kraus & Bar-Cohen, 1983) possibly leading to permanent damage on the equipment. Circuit boards are usually assembled with small spacing, thus constituting compact ducts, through which air flows to convectively cool the components. The circuit board, or substrate, also acts as a thermal conductor, transferring heat by conduction. This constitutes a conjugate forced convection-conduction cooling mechanism, as the components lose heat both by forced convection to the airflow and by conduction to the substrate.

The rate of convective heat transfer q_{cv} from a heater may be conveniently described in terms of the adiabatic convective coefficient h_{ad}, described by Moffat and co-authors (Moffat et al., 1985, Moffat and Anderson, 1990, Moffat, 1998, Moffat, 2004). This coefficient is dependent on the geometry of both the channel and the heaters, the properties of the fluid and also on the flow Reynolds number. It does not depend, however, on the power dissipated by the heaters, making it an invariant descriptor of the convective heat transfer. For a heater transferring a heat rate q_{cv} by convection, at a temperature T_h and with a convective surface area A_h, the adiabatic convective coefficient h_{ad} may be evaluated according to Eq. (1).

$$h_{ad} = \frac{q_{cv}/A_h}{T_h - T_{ad}} \tag{1}$$

In this equation, the temperature T_{ad} refers to the adiabatic temperature of the heater. This is the temperature that it would achieve in equilibrium should its power be turned off, while all of the surrounding temperatures remain the same and it does not exchange heat by conduction or radiation. The adiabatic coefficient h_{ad} may also be expressed in a dimensionless form as an adiabatic Nusselt number Nu_{ad} shown in Eq. (2), for a heater with edge L_h along the air flow and a fluid with thermal conductivity k.

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

When the substrate's thermal conductivity is sufficiently low, the heater will dissipate most of its power by forced convection. In this situation, Eq. (1) may be used to accurately predict its equilibrium temperature. However, as the thermal conductivity of the substrate increases, the rate of conductive heat loss may become of the same order of magnitude as q_{cv}. The heater is then cooled by a conjugate forced convection-conduction mechanism. The adiabatic convective coefficient is still an invariant descriptor in this situation. However, Eq. (1) will not be able to accurately predict the heater's equilibrium temperature, as only a fraction of its heat loss will occur by convection.

The conjugate heat transfer may be described by the invariant g^+_{ni} coefficients presented by Alves (2010) and Alves and Altemani (2012). These coefficients may be expressed as a square matrix G^+ of order N equal to number of heaters in an array. Thus, for an array of N heaters cooled by forced convection with a mass flow rate $\dot{m}$ of a fluid with specific heat c_p and inlet temperature T_0, the temperature T_h of each heater transferring heat at a rate q_{cj} by conjugate forced convection-conduction may be evaluated as

$$\begin{bmatrix}(T_h\text{-}T_0)_1\\(T_h\text{-}T_0)_2\\\vdots\\(T_h\text{-}T_0)_N\end{bmatrix} = \frac{1}{\dot{m}c_p}\begin{bmatrix}g^+_{11} & g^+_{12} & \cdots & g^+_{1N}\\ g^+_{21} & g^+_{22} & \cdots & g^+_{2N}\\ \vdots & \vdots & \ddots & \vdots\\ g^+_{N1} & g^+_{N2} & \cdots & g^+_{NN}\end{bmatrix}\begin{bmatrix}q_{cj,1}\\q_{cj,2}\\\vdots\\q_{cj,N}\end{bmatrix} \tag{3}$$

In this equation, the diagonal coefficients g^+_{nn} represent the self-heating effect of the n^{th} heater. The remaining non-diagonal g^+_{ni} terms of the G^+ matrix indicate the effect that the i^{th} heater's conjugate heat transfer has on the equilibrium temperature of the n^{th} heater. Maintaining the same geometry, fluid properties and duct mass flow rate, the conjugate coefficients are also independent of conjugate heat loss from the heaters. They are invariant descriptors of the conjugate heat transfer. When only two heaters are mounted on the substrate plate, Eq. (3) assumes the form

$$\begin{bmatrix}(T_h\text{-}T_0)_1\\(T_h\text{-}T_0)_2\end{bmatrix} = \frac{1}{\dot{m}c_p}\begin{bmatrix}g^+_{11} & g^+_{12}\\ g^+_{21} & g^+_{22}\end{bmatrix}\begin{bmatrix}q_{cj,1}\\q_{cj,2}\end{bmatrix} \tag{4}$$

The purpose of the present work was to obtain an experimental evaluation of the heat transfer from two heaters mounted in a top inlet duct in terms of the adiabatic coefficient h_{ad}, evaluated for both heaters, and the conjugate coefficients g^+_{ni}. Two distinct substrate plates were used: one made of acrylic, with a relatively low thermal conductivity, and one of aluminum, with a comparatively high thermal conductivity. These materials where chosen in order to investigate the effect of conduction on the conjugate heat transfer. Tests were performed for Reynolds numbers in the range from 1200 to 3000, based on the hydraulic diameter of the duct. For a duct perimeter P_d, mass flow $\dot{m}$ and fluid of dynamic viscosity μ, this Reynolds number was defined as

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

2. MATERIALS AND METHODS

The experiments were performed using a pair of symmetrical ducts. This assembly consisted of an acrylic duct divided in two by a substrate plate at half of its height, as shown in Fig. 1. Two different substrate plates were used: one made of acrylic and another made of aluminum. Each substrate plate (acrylic or aluminum) was inserted into slots along the duct lateral walls, allowing for one substrate plate to be easily removed and replaced by the other.

Both ducts had 20 mm height, 160 mm width, 250 mm length with a top inlet. Two pairs of heaters were placed 55 mm away from the lateral walls on each side of the substrate plate, totalizing four heaters. The first pair was placed directly below the airflow inlet of each duct, 25 mm away from the closed front wall. The second pair was placed 25 mm downstream from first one. A side view of the duct, showing the positioning of the heaters is shown in Fig. 1(b). The two upstream heaters were named Heater 1a and Heater 1b, while the two downstream heaters were named Heater 2a and Heater 2b. All heaters had a 50x50 mm cross section and 6 mm height. They were made from two aluminum pieces encasing an electrical resistance of around 8 Ω made of a Chromel wire with a Teflon coating. All heater surfaces were polished to a mirror-like finish in order to reduce thermal losses by radiation. After assembly, the outer walls of the acrylic duct were covered in aluminum foil, in order to reduce radiation losses, and it was also thermally insulated with a 100-mm-thick layer of polyurethane foam.

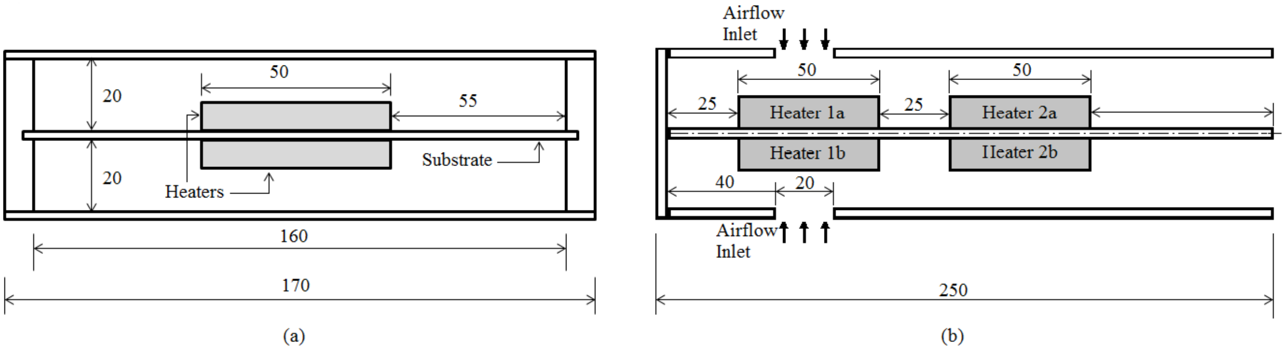

Figure 1. Front view (a) and side view (b) of the duct – all dimensions are in millimeters.

The double duct was attached to a plenum box and air flow was forced by a fan. Air from the laboratory was suctioned by the fan, flowing into the two ducts before being discharged into the first chamber of the box. Then it crossed a long radius nozzle with internal diameter of 17 mm into the second chamber before being discharged to the environment outside of laboratory. A schematic view of this apparatus is shown in Fig. 2.

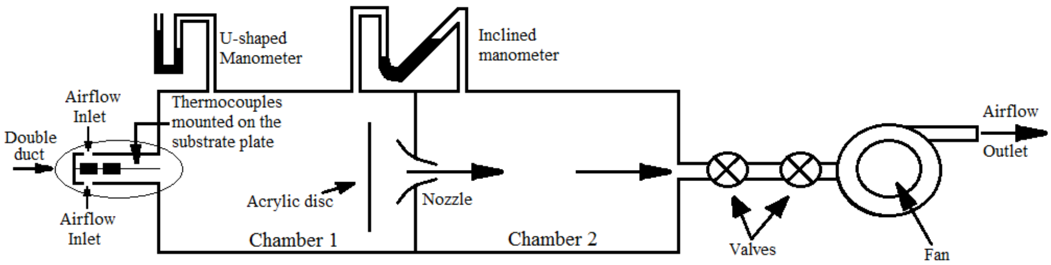

Figure 2. Schematic of the experimental apparatus

The pressure difference between the two chambers was measured using an inclined manometer filled with ethanol (relative density 0.7876). This value was then used in the nozzle calibration curve in order to obtain the mass flow rate through the plenum box. Since the two ducts were symmetrical, it was assumed that half of the airflow passed through each duct. A U-shaped manometer filled with water measured the gauge pressure in Chamber 1, while atmospheric pressure was obtained using a digital barometer (Sodmex – Brazil).

Temperature was measured by using type E thermocouple wires (Omega Engineering, USA). Heaters 1a and 2a had two thermocouples each, while Heaters 1b and 2b had a single thermocouple. Additional thermocouples were positioned along the substrate plates, airflow inlet and the upper wall of the duct. The lower duct had only the two thermocouples used to measure the temperatures of Heaters 1b and 2b. Temperature measurements were read on a digital temperature indicator (Omega Engineering, model DP41-TC, USA) with a resolution of 0.1 ºC. Two independent DC power supplies (HP, model 6296, USA) were used to independently control the power dissipation of the pair of active heaters.

Experimental tests were initially with only one pair of active heaters at a time, i.e only Heaters 1a and 1b would be powered on while Heaters 2a and 2b would be powered off or vice-versa. Power dissipation was controlled so that both active heaters would achieve the same temperature, thus maintaining the thermal symmetry between the two ducts. Around 20 tests were run for each substrate plate, being 10 tests with each pair of heaters active at a time. Half of the tests with each pair were made maintaining the heater's temperatures close do 40 ºC, while in the other half the temperatures were close to 50 ºC. This procedure was adopted in order to verify that both the adiabatic convective coefficient and the conjugate coefficients are invariant to the power dissipation in the heaters. During the tests, data was collected every 30 minutes and consisted of the thermocouple readings, the height of fluid in both manometers, power dissipation on both active heaters and atmospheric pressure. The system was assumed to be under steady state conditions when thermocouple readings were only 0.1 ºC apart in two consecutive measurements, after two to three hours from the start of the test. Only measurements obtained in this condition were used to evaluate the results. Some of the experiments were repeated in order to check the obtained values.

Tests performed with the acrylic substrate plate were used to evaluate both coefficients. Special care was taken to reduce the thermal contact between the heaters and the substrate plate, emphasizing the convective heat transfer. Small, triangular pieces of 0.25mm-thick insulation tape were fixed on the corners of the heaters, reducing their contact with the acrylic substrate. The resulting gaps between the edges of the heaters and the substrate plate were sealed using silicone rubber, thus preventing an air flow below the heaters. Due to the symmetry of the experimental apparatus, both active heaters in each test needed approximately the same electric power in order to achieve the same temperatures. Thus, the average rate of convective heat transfer q_{cv} from each heater was evaluated as

$$q_{cv} = q_h - (q_w + q_{cd} + q_{rad}) \tag{6}$$

In this equation, q_h represents the average electric power dissipated by the two active heaters. Heat losses are represented by the three other terms: q_w indicates average thermal losses by both the thermocouple and power wires, q_{cd} represents average conduction losses to the substrate plate and q_{rad} represents average thermal losses by radiation from the heaters' surfaces to the surroundings.

In order to evaluate the conjugate coefficients g^+_{ni}, the average rate of conjugate heat transfer q_{cj} of the active pair of heaters was also evaluated as shown in Eq. (7). In this case, only losses by conduction from the thermocouple and power wires and by radiation are considered, since the conductive heat transfer to the substrate plate is part of the conjugate mechanism. Just like on Eq. (6), q_h, q_w and q_{rad} refer to the average values obtained from the two active heaters.

$$q_{cj} = q_h - (q_w + q_{rad}) \tag{7}$$

Experimental tests were also performed using the aluminum substrate plate, in order to obtain the conjugate coefficients when conduction corresponds to a larger fraction of conjugate heat transfer. The thermal contact between the heaters and the substrate plate was enhanced by a thin layer of thermal grease. The tests with the aluminum substrate indicated that the power dissipation needed to achieve the same temperature in both active heaters was not as uniform as in the case with the acrylic substrate. Thus, in this case the power dissipation q_h and the two losses indicated in Eq. (7) were considered just for one side of the duct, i.e, for Heaters 1a and 2a.

3. RESULTS

The experimental tests were run for average air velocities inside each duct varying from 0.5 m/s to 1.4 m/s, corresponding to Reynolds numbers Re_D in the range from 1200 to 3000. Experimental data obtained with the acrylic substrate plate were used evaluate both the adiabatic convective coefficient h_{ad} and the conjugate coefficients g^+_{ni}. With the aluminum substrate plate, only the conjugate coefficients were evaluated, since the convective and conductive components of the heater conjugate heat loss have the same order of magnitude in this case. The experimental tests were performed with the heaters maintained under steady state either at 40ºC or at 50ºC in order to verify the invariant nature of h_{ad} and g^+_{ni}. Uncertainties of the final results were propagated from the measured values according to the procedure described by Kline and McClintock (1953).

3.1 Acrylic subsbrate plate

A total of 23 experiments were run with the acrylic substrate plate. The thermal contact between each of the four heaters and substrate plate was minimized, in order to reduce the conductive heater heat transfer to the substrate plate and emphasize its convection to the airflow. The convective heat transfer q_{cv} was evaluated according to Eq. (6) and then used to obtain the adiabatic convective coefficient h_{ad} using Eq. (1) and the adiabatic Nusselt number Nu_{ad} from Eq. (2). Considering just one pair of active heaters (1a and 1b or 2a and 2b) in each test, the adiabatic temperature was the flow inlet temperature in the duct.

The Nu_{ad} values obtained for each pair of heaters are shown in Fig. 3, as a function of the Reynolds number Re_D. These results show that the adiabatic Nusselt number is invariant to the convective heat transfer from the heaters as a single correlation was obtained for both temperatures used on each pair of heaters. Convective heat transfer was responsible for transferring 87.5% to 93% of the heaters power dissipation, indicating that Eq. (1) may be used to accurately predict the heaters' equilibrium temperature in this situation. Values obtained for $Nu_{ad,1}$ are larger than those for $Nu_{ad,2}$ for smaller Reynolds numbers, but as the flow rate increases, the results for both protruding heaters approach each other.

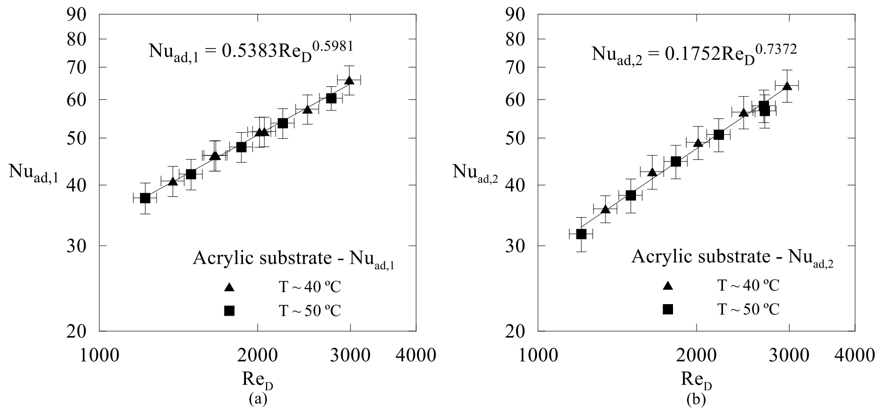

Figure 3. Adiabatic Nusselt numbers for the acrylic substrate. (a) Nu_{ad} for Heaters 1a and 1b. (b) Nu_{ad} for the heaters 2a and 2b

These experimental measurements were also used to evaluate the conjugate coefficients. The conjugate heat transfer q_{cj} was obtained from Eq. (7) and then used to evaluate the conjugate coefficients g^+_{ni} using Eq. (4). The results are shown in Fig 4 as a function of the Reynolds number. These results also indicate that the g^+_{ni} coefficients are invariant to the conjugate heat loss from the heaters. The conjugate forced convection-conduction mechanism ranged from 96% to 98% of the total power dissipated in the heaters mounted on the acrylic substrate. This suggests that Eq. (4) may be used to predict the heaters' final temperature more accurately than Eq. (1).

Values obtained for the g^+_{12} coefficient are comparatively small with relatively large uncertainty, indicating that the power dissipation in the downstream heater has a negligible effect on the temperature of the upstream one when using the acrylic plate. The results for the g^+_{11} and g^+_{22} coefficients, which indicate the self-heating effect of each heater, are within less than 10% of each other, indicating very similar conjugate heat transfer rates for the two pairs of heaters.

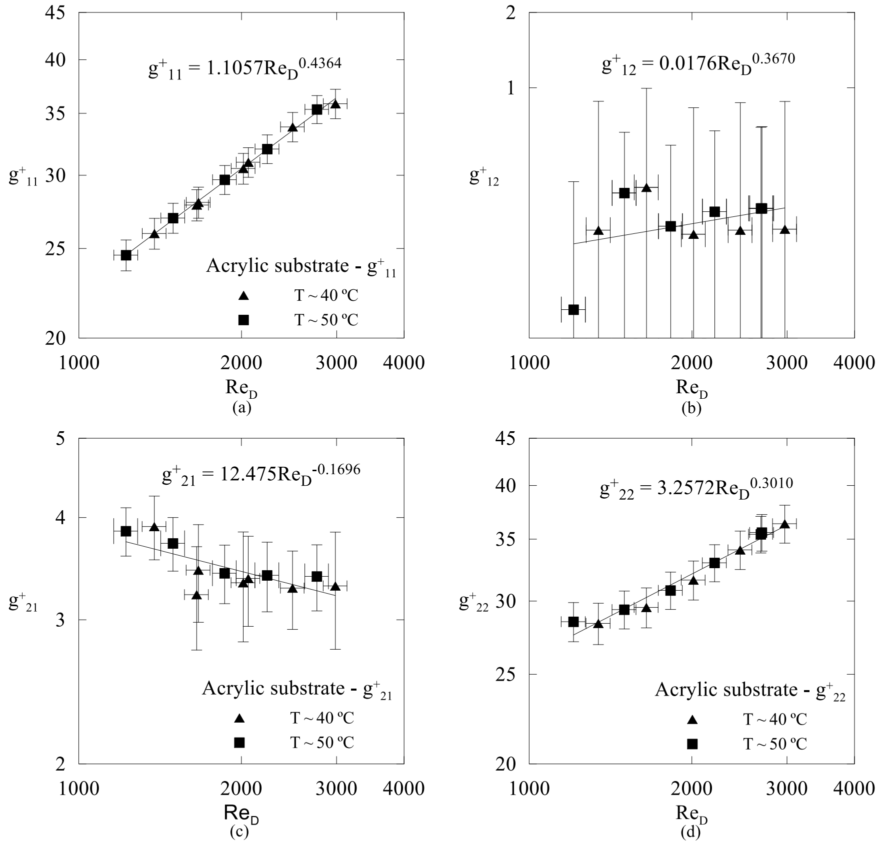

Figure 4. Conjugate coefficients for the acrylic substrate plate (a) g^+_{11} (b) g^+_{12} (c) g^+_{21} (d) g^+_{22}

3.2 Aluminum subsbrate plate

For the 23 tests with the aluminum substrate, the thermal contact between the heaters and the substrate plate was increased by using thermal grease at their interface. Combined with the higher thermal conductivity of aluminum, this meant that heat transfer by conduction to the substrate plate was much larger than with the acrylic substrate. For this reason, only the conjugate coefficients g^+_{ni} were evaluated from tests with the aluminum substrate. Evaluating Nu_{ad} would likely result in large experimental uncertainties. Results obtained for the conjugate coefficients for this substrate plate are shown in Fig. 5. Conjugate heat transfer was responsible for 90% to 95% of total power dissipation, this fraction increasing with the Reynolds number. This smaller participation, when compared to the acrylic substrate plate, occurs because the aluminum substrate attains higher temperatures and thus larger radiation losses in Eq. (7), resulting in overall higher thermal losses. However, conjugate heat transfer is still large enough so that Eq. (4) is able to accurately predict the heaters' equilibrium temperature.

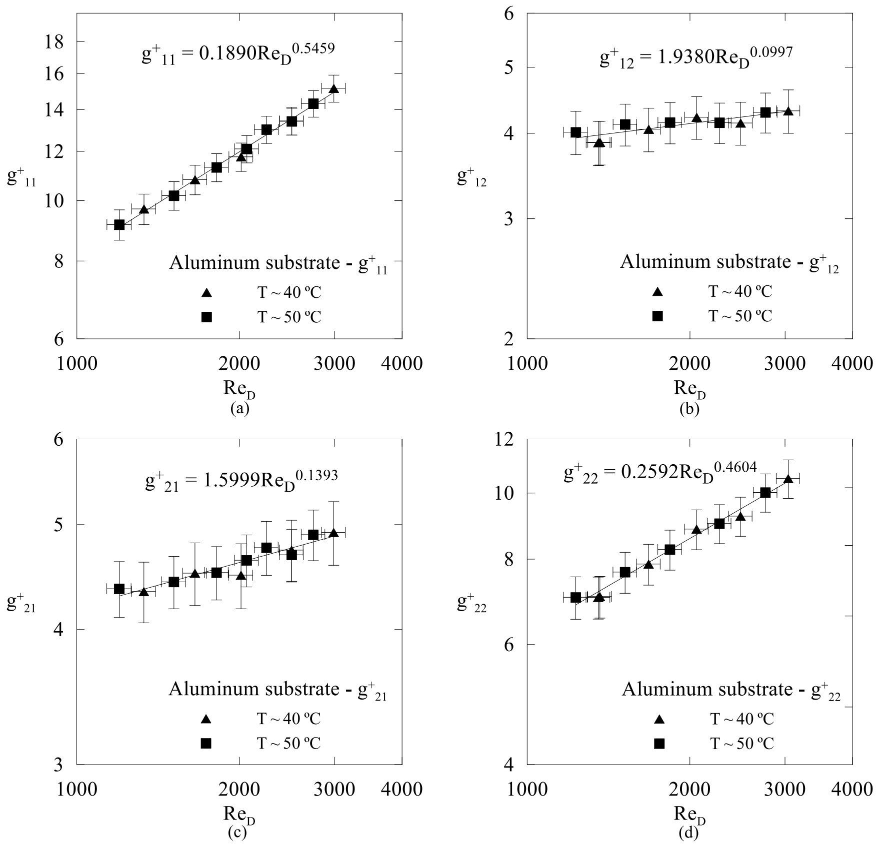

Figure 5. Conjugate coefficients for the aluminum substrate plate (a) g^+_{11} (b) g^+_{12} (c) g^+_{21} (d) g^+_{22}

With the aluminum substrate plate, the g^+_{22} coefficient is noticeably smaller than g^+_{11}, indicating that for the same airflow rate Heater 2a presents a higher conjugate heat transfer rate than Heater 1a. This may be explained by the conductive fraction of conjugate heat transfer. As shown in Fig. 1, Heater 1a is closer to the edge of the substrate plate, while Heater 2a is closer to its middle, thus being able to dissipate more power by conduction and more heat overall. This comparison shows that Eq. (1) would not be able to predict the equilibrium temperature of the heaters accurately, since much of the conjugate heat transfer occurs by conduction.

The g^+_{11} and g^+_{22} coefficients obtained for the aluminum substrate plate are also smaller than the corresponding ones obtained from experiments with the acrylic one, indicating more intense conjugate heat transfer while using the aluminum substrate. This is also explained by the conduction portion of the conjugate heat transfer, as the aluminum thermal conductivity is considerably larger. On the other hand, the g^+_{12} and g^+_{21} coefficients, which indicate the effect of the conjugate heat transfer of one heater on the temperature of the other one, are considerably larger than those obtained for the acrylic substrate. This indicates that the conduction through the aluminum substrate implies a larger influence of one heater on the temperature of its neighbor.

4. CONCLUSIONS

Experimental results were obtained for two invariant descriptors of the cooling of heaters mounted on a substrate plate with forced airflow in a duct with a top inlet. One was the adiabatic convective coefficient for the heaters mounted on a low thermal conductivity substrate plate. The other invariant descriptor as a matrix of dimensionless conjugate coefficients, used to describe the conjugate forced convection-conduction of the heaters. Two distinct substrate plates were used, in order to investigate the effect of conduction on the conjugate heat transfer, one made of acrylic and the other, of aluminum. Experimental tests were performed for Reynolds numbers Re_D in the range from 1200 to 3000, under steady state conditions. Tests were performed for two different heater temperatures, in order to verify that the coefficients are indeed invariant to the power dissipation. Two pairs of heaters were mounted on each substrate plate.

Both descriptors were evaluated for the acrylic substrate plate. Convective heat transfer was responsible for about 90% of total power dissipation in the heaters on substrate plate, indicating that the adiabatic convective coefficient h_{ad} may be used to predict the temperature of the heaters accurately. Considering, however, the conjugate forced convection-conduction from the heaters, this fraction increased to about 97%, indicating that the the conjugate coefficients g^+_{ni} give an even better temperature prediction. Correlations were obtained for h_{ad} and all four g^+_{ni} coefficients as functions of the Reynolds number. Since a single correlation was obtained for each descriptor for distinct heater temperatures, they were shown to be invariant to the conjugate heat transfer from the heaters.

Results from the aluminum substrate plate were used to evaluate only the conjugate coefficients, due to the higher fraction of heat transfer by conduction. In this case, the fraction conjugate heat transfer ranged from 90% to 95% of the power dissipation in the heaters, indicating that the g^+_{ni} would also be adequate in predicting the heaters temperatures. The adiabatic convective coefficient would not be able to accurately predict the heaters final temperatures when using this substrate plate. A single correlation was also obtained for each one of the g^+_{ni} coefficients, indicating that, for this substrate plate, they are invariant to the conjugate heat loss the heaters.

5. ACKNOWLEDGEMENTS

The support of CAPES in the form of a Master's Program Scholarship to the first author is gratefully acknowledged.

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