EXPERIMENTAL EVALUATION OF THE CONVECTIVE AND THE CONJUGATE COOLING OF A PROTRUDING HEATER IN A DUCT

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Abstract. *Experiments were performed to investigate the conjugate forced convection-conduction cooling of a protruding heater mounted on the lower (substrate) plate of a rectangular duct. The heater was an Aluminum rectangular block heated by means of electric power dissipation in an embedded resistance. Airflow was forced in the duct with the hydraulic diameter Reynolds number (Re_D) in the range from 2,000 to 6,000. Effects of the substrate plate thermal conductivity on the heater conjugate cooling were obtained from measurements with two distinct plates: a Plexiglas plate and an Aluminum plate. The results were expressed in terms of the adiabatic Nusselt number (Nu_{ad}) and the conjugate coefficient (g^+_{11}), both as functions of Re_D. The direct convective heat loss from the heater surfaces to the airflow was described by the adiabatic Nusselt number. The conjugate cooling, encompassing the heater direct convective loss to the airflow and the conduction loss to the substrate plate was described by a single conjugate coefficient g^+_{11}.*

Keywords: *conjugate heat transfer, adiabatic Nusselt number, conjugate coefficient, experimental investigation*

1. INTRODUCTION

Electronic components assembled on circuit boards dissipate electric power by ohmic effect during their operation. Due to the decreasing size of the components, the thermal design of a circuit board must be performed carefully, since they operate reliably only within a limited range of temperatures. At temperatures above a maximum usually specified by the manufacturers, the components life cycle may be drastically reduced (Kraus and Bar-Cohen, 1983). This may cause electronic equipment failure due to overheating of a component on a circuit plate. Circuit boards are usually assembled with small spacing, constituting channels convectively cooled by forced airflow between the plates. The thermal control of board components with high heat flux sometimes is attained with the help of a finned heat sink attached to the component. Due however to miniaturization and compactness of electronic equipment, many times nowadays there is not enough room for heat sinks and then one alternative to cool these components is to make use of the circuit boards as thermal conductors (Nakayama, 1997). In this case, the components on a circuit board are cooled by a conjugate forced convection-conduction mechanism, comprising forced convection from the components surfaces in contact with the airflow and conduction through their contact with the board (substrate plate). The present experimental investigation was performed considering a discrete heater assembled on a conductive substrate plate and cooled by forced airflow, as indicated in Fig. 1.

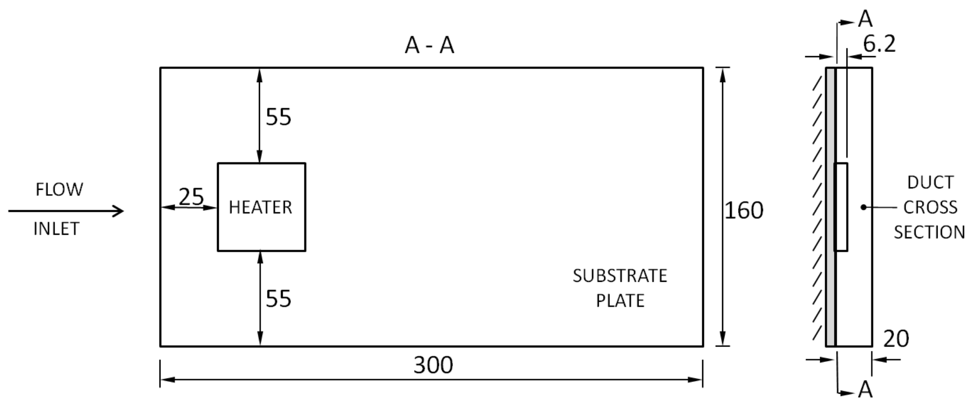

Figure 1. Heater in the duct

The rate of convective heat transfer (q_{cv}) from the heater was described by the adiabatic heat transfer coefficient (h_{ad}), since it is invariant with the power dissipation in the heater. This concept was developed by Moffat and co-authors (Moffat et al., 1985, Moffat and Anderson, 1990, Moffat, 1998 and 2004). The reference temperature associated to this coefficient is the heater adiabatic surface temperature (T_{ad}). It is defined (Moffat, 1998) as the heater equilibrium temperature when its power is turned off, while it does not exchange heat by conduction and radiation and all the surrounding temperatures remain the same. It is expressed by

$$q_{cv} = h_{ad} A_s (T_s - T_{ad}) \tag{1}$$

In Eq. (1) A_s is the heater surface area in contact with the airflow and T_s is the heater surface temperature. For a substrate plate with low thermal conductivity, most of the electric power dissipated in the heater is removed by direct forced convection to the airflow (q_{cv}) and the heater temperature may be conveniently predicted by Eq. (1). As the substrate plate thermal conductivity increases, it may become an important conductive path for the heater cooling. In this case the heater power dissipation will occur by a conjugate forced convection-conduction mechanism and Eq. (1) alone will not be enough to predict T_s because in this case q_{cv} is only an unknown fraction of the power dissipation. There are several investigations related to this conjugate cooling, as reviewed by Nakayama, 1997. Discrete protruding heaters mounted on a conductive substrate plate cooled by channel flow were considered in the works of Davalath and Bayazitoglu, 1987, Kim and Anand, 1994 and 1995. They reported results for both the convective and the total heat transfer from the heaters respectively in terms of a Nusselt number and a global thermal resistance. These parameters were based either on the fluid inlet temperature in the channel or its mixed mean temperature just upstream of the heater. Their results depend on the power dissipation in the heaters and they were usually reported under the condition of uniform heating of several heaters. An invariant descriptor of this conjugate cooling was developed in the Doctoral Thesis of Alves (2010) in the form of dimensionless conjugate coefficients g^+_{ij} relating the temperature of a heater mounted on a conductive substrate plate to the power dissipation in all the heaters. Considering N heaters mounted on a substrate plate cooled by channel flow, it was shown that their temperature increase above the flow inlet temperature could be related to the power dissipation in each heater (q_j) by a square matrix of conjugate coefficients, as indicated in Eq. (2).

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

The channel flow rate is represented by $\dot{m}$ and the fluid specific heat is indicated by c_p. The i^{th} heater temperature increase above the fluid inlet temperature (T_0) is represented by $\Delta T_i = (T_s - T_0)_i$. Eq. (2) indicates that this temperature increase is related to the power dissipation in all the heaters (q_j) by a square matrix of dimensionless conjugate coefficients g^+_{ij}. In the work of Alves (2010), these coefficients were developed and obtained considering a numerical two-dimensional analysis of three heaters on a conductive substrate plate cooled by laminar airflow.

The present experimental investigation was undertaken to evaluate both the convective and the conjugate cooling of a single heater assembled on a substrate plate, as indicated in Fig. 1. These results were described respectively by the adiabatic Nusselt number Nu_{ad} and by a single conjugate coefficient g^+_{11}. Both were dependent on the channel flow Reynolds number Re_D, but they are invariant with the power dissipation in the heater.

2. EXPERIMENTAL APPARATUS AND PROCEDURE

The experiments were performed with a single protruding heater assembled on the lower horizontal wall (substrate plate) of a rectangular duct. The rectangular duct had a cross section of (160x20) mm and a length of 300 mm, with very smooth lateral and top surfaces, made of Plexiglas. Two distinct plates with identical dimensions were used as the duct lower wall - where the heater was mounted - one was made of Plexiglas and the other of polished Aluminum. The duct was assembled to a plenum box in the laboratory and airflow was forced through the duct in suction mode by a fan located downstream. The open airflow circuit is indicated in Fig. 2, where air from the laboratory was forced into the rectangular duct and then through the plenum box, where it was measured by a nozzle and channeled to a pipe with flow control valves. Downstream, the fan located outside the laboratory discharged the airflow outdoors.

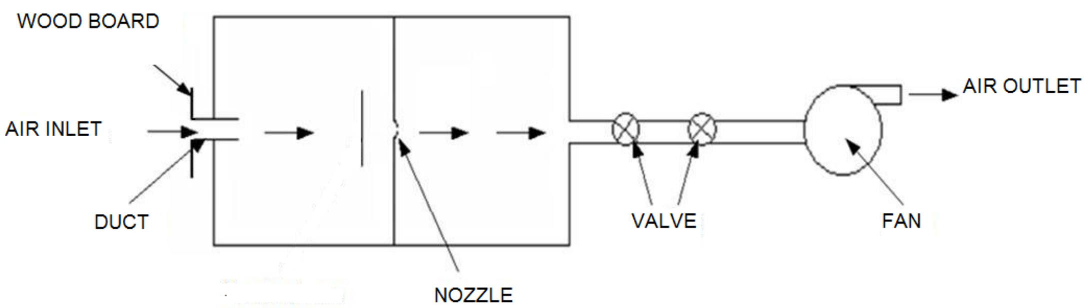

Figure 2. Diagram of the open airflow circuit.

The heater was a rectangular block made of Aluminum with a square base (50x50) mm and a height of 6.2 mm. It was made from two Aluminum pieces enveloping a 0.254 mm diameter Teflon coated Chromel resistance wiring with an electric resistance equal to 8 ohms. The heater was mounted on the lower duct wall with its upstream edge 25 mm from the duct entrance. Its height was 31% of the duct height and its lateral faces were 55 mm from the duct lateral walls. The outside surfaces of the duct were thermally insulated from the surroundings by a 100 mm thick layer of polyurethane foam. The flow was measured by a calibrated long radius nozzle with an internal diameter of 17 mm, located inside the plenum box, as indicated in Fig. 2. The pressure drop across the nozzle was measured by an inclined manometer filled with laboratory grade ethanol (relative density 0.7876) and with a 0.01 inches divisions scale. The temperatures were obtained from type K (Chromel-Alumel) Teflon coated thermocouple wires (Omega Eng., USA) with either 0.127 mm or 0.254 mm diameter. The thermocouples were positioned in the heater, at the inlet airflow and also distributed on the substrate plate and on the upper duct wall. The thermocouples were selected by a thermocouple switch (Omega Eng. OSW5-20, USA) and read by a digital electronic temperature indicator (Omega Eng. DP41-TC, USA), with 0.1°C resolution. The electric power dissipation in the heaters wiring was obtained from a DC power supply (HP 6296A, USA).

The experimental data collected in each test comprised the readings from the thermocouples, the manometer height for the flow measurement, and the electric power dissipation, which were obtained under steady state conditions, after a few hours of operation of the apparatus. In any test, with fixed power dissipation and airflow rate, steady state conditions were assumed when the temperature readings from all the thermocouples were within 0.1°C during an interval of 30 minutes.

The tests with the Plexiglas substrate plate served two purposes: to evaluate Nu_{ad} from the direct convective heat transfer from the heater to the airflow, and to evaluate the conjugate coefficient g^{+}_{11} from the forced convection-conduction heater losses. In these tests, care was taken to have a reduced conductive thermal contact at the heater-plate interface, in order to enhance the direct convective loss from the heater to the airflow. For this purpose, the four lower corners of the heater were coated with small triangle pieces of 0.25 mm thick electric insulation tape, so that the heater edges did not touch the substrate plate. The gap between the heater edges and the plate was filled with a thin film of silicone rubber to avoid any flow underneath the heater. In addition, the Plexiglas substrate wall under the heater was reduced to one half (1.2 mm) of its original thickness (2.4 mm). The cavity thus formed had a stagnant air layer and it was covered with an Aluminum foil to reduce thermal losses by radiation from the heater lower base. All the heater surfaces were polished to a mirror like reflective surface in order to reduce its emissivity and the thermal losses by radiation. The convective heat transfer from the heater to the airflow (q_{cv}) was obtained by an energy balance on the heater, subtracting the other heater thermal losses from the total power dissipated in the heater, as in Eq. (3).

$$q_{cv} = q_{dp} - (q_{cc} + q_{rd} + q_{sf} + q_w) \tag{3}$$

In Eq. (3), q_{dp} indicates the total dissipated power in the heater, and the remaining terms in parenthesis represent the heater thermal losses. Thus, q_{cc} represents the conduction losses from the heater lower base through the stagnant air layer in the Plexiglas cavity under the heater and q_{rd} indicates all the radiation losses from the heater surfaces. The heater conduction losses to the plate through the silicone film around its four lower edges are indicated by q_{sf} and the losses through the thermocouple and power wires connected to the heater are represented by q_w. The details of the thermal losses evaluation may be found in Loiola, 2010. For each experimental test with the Plexiglas plate, the value of q_{cv} obtained from Eq. (3) was replaced in Eq. (1) to obtain h_{ad} and it was presented by the adiabatic Nusselt number defined by Eq. (4).

$$Nu_{ad} = \frac{h_{ad}\, L_h}{k} \tag{4}$$

In Eq. (4), L_h is the edge of the heater square base and k indicates the fluid (air) thermal conductivity, evaluated at the film temperature between the heater and the incoming airflow. All the results were expressed as functions of the duct flow Reynolds number, based on the airflow mass flow rate and on its perimeter p_w, $\mathrm{Re}_D = 4\dot{m}/(\mu\, p_w)$.

The conjugate coefficient g^{+}_{11} for the heater on the Plexiglas substrate was obtained from the conjugate forced convection-conduction heat transfer rate q_{cj} from the heater to the airflow. It was obtained from a distinct energy balance for the heater, as follows,

$$q_{cj} = q_{dp} - (q_{rd} + q_w + q_{ti}) \tag{5}$$

As indicated by Eq. (5), the heater thermal losses by radiation (q_{rd}), by the heater wires (q_w) and by the thermal insulation layer underneath the substrate plate (q_{ti}), were subtracted from the power dissipation in the heater to obtain q_{cj}. The dimensionless conjugate coefficient g^{+}_{11} was then obtained from Eq.(2), in the form,

$$\dot{m} c_p (T_{s,1} - T_0) = g^+_{11} q_{cj} \tag{6}$$

The experimental tests with the Aluminum substrate plate served to enhance the conductive contribution to the conjugate heater cooling. The heater was tightened to the substrate plate by means of two screws and their interface was filled with a thermal paste. Due to the relatively large conduction from the heater to the Aluminum plate, the adiabatic Nusselt number was not evaluated from these tests – otherwise, its uncertainty would be too large. Thus, q_{cj} was obtained from an energy balance identical to Eq. (5) and g^+_{11} was obtained from Eq. (6). Since the thermal conductance from the heater to the substrate plate is larger for the Aluminum substrate than for the Plexiglas substrate, it may be expected that under the same flow and power conditions, the heater temperature will be smaller for the Aluminum substrate. From Eq. (2), this implies smaller conjugate coefficients for the Aluminum substrate.

The uncertainties of the experimental results were evaluated according to the procedure described by Kline and McClintock (1953). Uncertainties were associated to each measured quantity and the propagation of these quantities on the reported results was then evaluated according to this method. Thus, the uncertainties for the Reynolds number Re_D and for the Nusselt number Nu_{ad} were evaluated to be about 6%, while those for the conjugate coefficient g^+_{11} were evaluated around 4% for the Plexiglas substrate plate and about 6% for the tests with the Aluminum substrate plate.

3. RESULTS AND DISCUSSION

The experimental tests were run for a range of Re_D from 2,000 to 6,000, corresponding to average airflow velocities in the duct from 1 m/s to 3 m/s. In each test the heater temperature was adjusted by the electric power dissipation in its internal resistance. The three thermocouples inserted in the heater always indicated (within the 0.1°C meter resolution) a uniform heater temperature in each test. For distinct tests the heater temperature was kept within a small temperature range for any tested airflow rate by control of the heater electric power dissipation in each test - this power dissipation increased with the airflow rate. Eight tests were run with the Plexiglas substrate, with the heater electric power dissipation ranging from 0.95 W to 1.80 W. The heater temperature in all these eight tests was within (40±1)°C. Twelve tests were run with the Aluminum substrate, eight of which with the heater temperature in the range (40.5±0.5)°C and four tests with this temperature within (48.0±0.3)°C. For the first eight tests, the heater electric power dissipation varied from 5.56 W to 9.72 W, while for the four tests with the heater at a higher temperature, the power dissipation was in the range from 8.12 W to 13.66 W.

The results obtained with the heater on the Plexiglas substrate were the adiabatic Nusselt number Nu_{ad} defined by Eq. (4) and the conjugate coefficient g^+_{11}, expressed by Eq. (6), both as functions of Re_D. The experimental results for Nu_{ad} are presented in Fig. 3 (a), indicating its increase with Re_D, due to larger airflow rate in the duct. They were fitted by the power law correlation defined by Eq. (7).

$$Nu_{ad} = 0.22\, Re_D^{0.67} \tag{7}$$

Although Re_D was in the range of low Reynolds turbulent flow and the heater position was near the duct entrance, the exponent 0.67 is only slightly lower than the typical value 0.8 associated to fully developed duct turbulent flow. The results obtained in this case from the energy balance in Eq. (3) showed that the fraction of the heater power dissipation transferred directly by convection from the heater surfaces to the airflow increased with Re_D from 84% to 91%. This indicates that the heater temperature may be predicted reasonably well from the results of Nu_{ad} and Eqs. (1) and (4), since most of the heater cooling occurs by forced convection. The experimental data points for g^+_{11} are presented in Fig. 3 (b), together with a fitted power law correlation given by

$$g^+_{11} = 2.33\, Re_D^{0.37} \tag{8}$$

From the energy balance indicated by Eq. (5), the obtained results showed that the conjugate forced convection-conduction heater losses ranged from 94% to 97% in the investigated Re_D range. Thus, the heater temperature may be better predicted with Eq. (6) and the correlation for g^+_{11} than with Eqs. (4) and (1) and the correlation for Nu_{ad}. All the data points of Nu_{ad} and g^+_{11} in Figs. 3 (a) and 3 (b) were within 3% of their respective correlations.

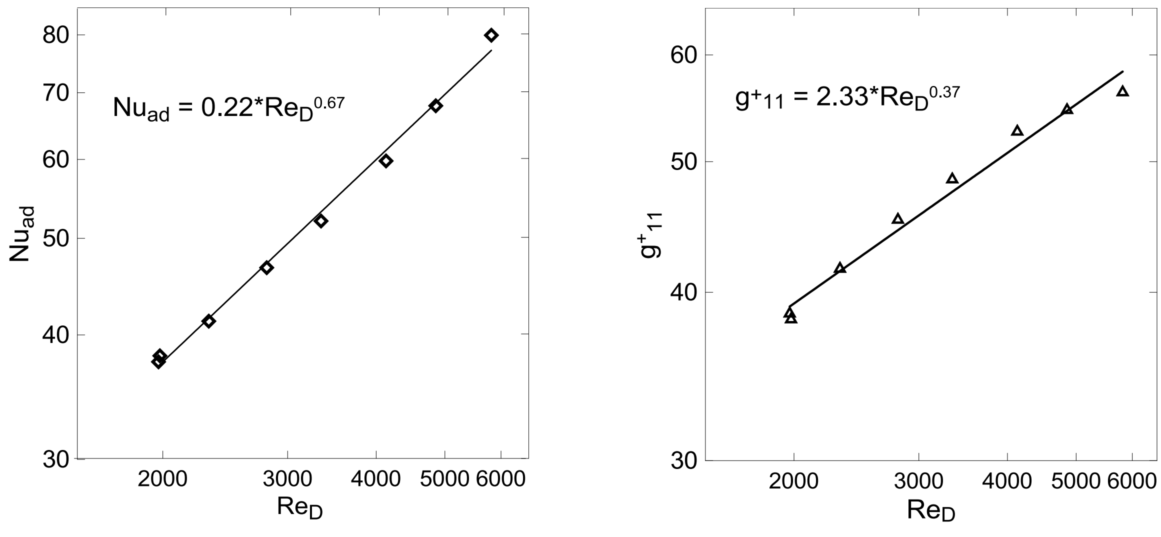

Figure 3. The adiabatic Nusselt number and the conjugate coefficient for the Plexiglas substrate.

From the tests with the heater on the Aluminum substrate, only the conjugate coefficient g^+_{11} was evaluated. The relatively large conduction loss from the heater to the substrate would cause a large uncertainty on the heater convective heat transfer evaluation from these tests. Since the only change relative to the previous tests was the material of the substrate plate, the correlation obtained for Nu_{ad} from the tests with the Plexiglas plate was assumed to be the same for the Aluminum substrate plate. Using the energy balance indicated by Eq. (5), the new experimental data indicated that the conjugate heater cooling increased with Re_D from 86% to 93% of the heater power dissipation. These fractions are smaller than the corresponding values for the Plexiglas substrate because the Aluminum plate operates at a higher temperature and consequently its thermal losses are larger. The conduction heat transfer from the heater to the Aluminum substrate plate was estimated subtracting the convective transfer rate from the conjugate heater cooling rates. The results indicated that the conduction contribution to the heater cooling ranged from 67% to 71% of the heater power dissipation as Re_D increased in the investigated range. In this case, knowing the heater power dissipation, its temperature would be reasonably well predicted from the conjugate coefficient. On the other hand, a prediction based on Nu_{ad} would indicate a heater temperature increase about three times larger than the correct value. The experimental results for g^+_{11} were obtained from two sets of data with the heater maintained under steady conditions either at 40°C or at 48°C. As indicated in Fig. 4, all the data were fitted into a single correlation,

$$g^+_{11} = 0.34\ Re_D^{0.43} \tag{9}$$

The data points in Fig. 4 are within 2% of this correlation for g^+_{11}. The values of g^+_{11} presented in Fig. 4 for the Aluminum substrate are about a quarter of those presented in Fig. 3 (b) for the Plexiglas substrate at the same Re_D due to the larger global thermal conductance between the heater and the airflow.

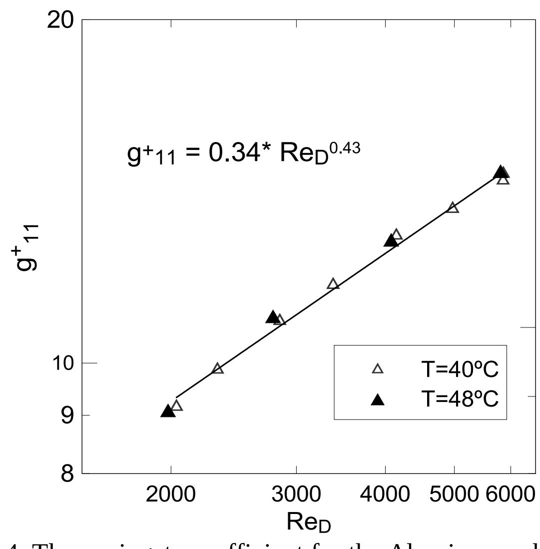

Figure 4. The conjugate coefficient for the Aluminum substrate.

4. CONCLUSIONS

The reported experimental investigation furnished the characteristics of the conjugate forced convection-conduction cooling of a protruding heater mounted on a conductive substrate plate in a duct. The tests with a Plexiglas substrate plate indicated that the direct convection from the heater to the airflow varied from 84% to 91% of the heater power dissipation. Thus, the adiabatic Nusselt number was evaluated from these tests, since it is an invariant descriptor of the convective heat transfer and it is very convenient when the heater is cooled mainly by forced convection.

When the heater cooling is dominated by the conjugate forced convection-conduction mechanism, as was the case with the Aluminum substrate, it is much more convenient to describe this process by means of the conjugate coefficient g^+_{ij} in matrix form. In the tests with the Aluminum plate, the heater direct convection losses were evaluated in the range from 18% to 22% of the heater power dissipation, while the conjugate losses ratio ranged from 86% to 93%. In the present investigation, with a single heater on the substrate plate, a single coefficient was evaluated, ie, g^+_{11}. This coefficient was obtained with distinct heater temperatures, showing its invariant characteristic and convenience to describe the conjugate cooling.

The conjugate coefficient for the Aluminum plate was nearly a quarter of that for the Plexiglas plate, an expected trend, since the heater conduction losses were higher than those for the Plexiglas substrate. Thus, for the same heater power dissipation, its predicted temperature will be smaller for the Aluminum substrate plate.

5. ACKNOWLEDGEMENTS

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