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A matrix to evaluate the conjugate cooling of a heaters' array

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ABSTRACT

Circuit boards are usually cooled by forced airflow and the heat transfer from the discrete electronic components may be enhanced by conductive boards. The conjugate forced convection-conduction cooling of an array of N heaters mounted on a conductive substrate was described by means of dimensionless conjugate coefficients g^{+}_{kj} grouped in a N-square matrix G^{+}. Experiments were performed with one or two rows of protruding heaters mounted on the conductive lower wall of a rectangular duct, made of either Aluminum or Plexiglas. One end of the duct was closed and the heaters were cooled by two impinging airflows exiting from square holes at the duct upper wall. For each substrate plate, the conjugate coefficients were obtained from tests with a single active heater at a time and expressed as functions of a Reynolds number in the range from 2000 to 7000. Additional tests with two or three active heaters at a time were then performed and the measured temperature increase of each heater above the flow inlet temperature was predicted quite well using the previously obtained coefficients matrix G^{+}. These results showed that the conjugate coefficients are invariant with the conjugate cooling rates from the heaters. The conjugate coefficients were also used to predict the allowable conjugate cooling rates for an array of heaters, in order to keep their temperature within reliable limits. Numerical simulations were performed for a duct configuration similar to that of the experiments, but with flush mounted heaters. The numerical results were thus distinct from those of the experiments with protruding heaters, but they showed similar trends of change with the Reynolds number and they indicated a perspective of the effects of the heaters height.

1. Introduction

Electronic components mounted on circuit boards are usually cooled by forced airflow due to its availability, ease of handling and high dielectric strength. On the other hand, the air thermal properties, typical of gases, and the trends of miniaturization and increasing performance of electronic equipment, demand an increasing need for a careful thermal design in order to keep the components' temperatures within their reliable limits [1]. Heat transfer enhancement techniques, such as finned heat sinks mounted on top of critical components [2–4], jet flow impingement [5,6] and conductive circuit boards [7,8] are often employed for this purpose. The available space in electronics cooling is often restricted, so that sometimes heat sinks may not be an option. In this case, impinging flow and conductive boards may be combined to enhance the heat transfer from a component on a circuit board. When an electronic component is cooled mainly by convection, its temperature may be conveniently predicted by the adiabatic heat transfer coefficient, as described by Moffat [9,10], because it is an invariant descriptor of the convective heat transfer rate from the component. When a conductive circuit board is employed, the electronic components are cooled mostly by conjugate forced convection-conduction, so that the adiabatic heat transfer coefficient alone may not allow a reliable prediction of their temperature.

Davalath and Bayazitoglu [11] performed a pioneering two-dimensional numerical investigation of the cooling of three protruding heaters mounted on the lower wall (substrate plate) of a parallel plate channel with forced laminar flow. The substrate plate was assumed either insulated or thermally conductive. Their results indicated that the heaters local *Nu* was noticeably larger at their top surfaces than at their side surfaces, while the average *Nu* for each heater decreased downstream in the channel. They also indicated a substantial temperature decrease around the heaters when the adiabatic substrate plate was replaced by a conductive substrate. Kim and Anand [12] investigated numerically the cooling of five

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and a half uniformly heated protruding blocks mounted on the lower conductive wall of a channel by laminar forced flow. An overall thermal resistance was employed to characterize the conjugate forced convection-conduction cooling of the blocks. It decreased as the channel flow Re or the substrate thermal conductivity increased, as expected. Compared to the results for an adiabatic substrate, the heaters thermal resistances and temperature decreased markedly even for a small substrate thermal conductivity relative to that of the fluid. Kim and Anand [13] also investigated numerically the conjugate cooling of 2D protruding heaters mounted on a conductive plate between a series of parallel plates. The results were presented in terms of a Nusselt number for the convective heat transfer and a global thermal resistance to include the conductive contribution. In these investigations, the heaters' cooling was described either by a Nusselt number or by a thermal resistance. Both were defined considering the heaters' temperature rise above either the channel flow inlet temperature or the local mean flow temperature just upstream of each heater. Thus, they depend on the power distribution in the heaters' array and the presented results are valid only for uniform heating. Young and Vafai [14] investigated the cooling of a single protruding heater in a parallel plate channel with adiabatic walls. They showed that the shape and material of the obstacle has a significant effect on the flow and heat transfer and they presented numerical results for the local and the average Nusselt numbers for the obstacles under laminar flow conditions in the channel. Young and Vafai [15] considered several protruding heaters mounted on a parallel plate channel with adiabatic walls, cooled by a laminar flow. Their results indicated that the heaters' average Nusselt number tended to a periodically developed value which was approached by the ninth heater in a row. Zeng and Vafai [16] considered the convective cooling of an array of 2D protruding heaters mounted in a channel and presented two general correlations for the Nusselt number. They performed extensive numerical simulations taking into account distinct geometric configurations of the channel and the heaters, under laminar flow conditions. Two recent works by Tavakkoli et al. [17,18], presented a comprehensive thermal analysis and optimization of a 3D integrated circuit (IC). The 3D ICs present superior electronic performance when compared to conventional 2D technology. They performed numerical simulations of a 3D IC model composed of a substrate, thermal interface materials, dies, device layers, a heat spreader and a heat sink packaged on top of each other. The chips in their 3D IC model were composed of three 0.1 mm thick silicon dies with a 0,002 mm thick device layer on top of each die, separated from each other by layers of thermal interface material 0.015 mm thick. They might be integrated by through-silicon vias (TSVs), so that the entire system behaved as an integrated device. In spite of the many variables associated to their model, the authors were able to contribute with distinct key features to the understanding of a 3D IC thermal behavior and optimization.

Alves and Altemani [19] performed numerical simulations of the conjugate forced convection-conduction cooling of an array of 2D protruding heaters mounted on the lower conductive wall of a channel, as indicated in Fig. 1. The numerical results indicated that the temperature increase of any heater above the flow inlet temperature was related to the conjugate cooling rates of all the N heaters of the array by means of a square matrix G^+ of dimensionless coefficients g^+_{kj}, as in Eq. (1).

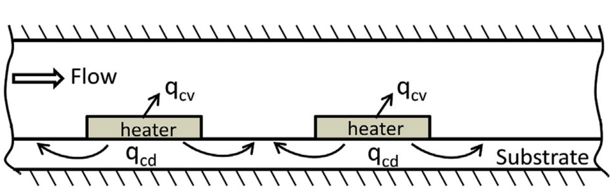

Fig. 1. Conjugate cooling from protruding heaters on a conductive substrate plate.

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

In Eq. (1), $(\dot{m}c_p)$ is the channel flow thermal capacity, while q_j and ΔT_j indicate, for the jth heater of the array, respectively the conjugate forced convection-conduction cooling rate and the temperature increase above the channel flow inlet temperature T_0. The lower surface of the substrate plate indicated in Fig. 1 was adiabatic, so that the conduction loss from any heater to the substrate plate eventually returned to the flow by convection. They also performed numerical simulations of similar problems and compared their results with those previously presented in the literature. One comparison was with the convective cooling of a single 2D protruding heater mounted on an adiabatic substrate in a parallel plate channel. The results indicated isotherms and streamlines almost identical with those presented in the original work of Young and Vafai [14]. The simulations also reproduced recirculation flow lengths downstream of the protruding heater within 1% of the data presented by Zebib and Wo [20] and by Leung et al. [21]. Simulations were also performed for the configuration with three 2D protruding heaters with the same configuration studied originally by Davalath and Bayazitoglu [11], considering a conductive substrate. The results obtained for the local Nusselt number along the heaters were distinct from those of the original work, but agreed with the results reported previously by Young and Vafai [15], Kang et al. [22], Kim et al. [23], and Huang et al. [24]. The results for the average Nusselt number were about 30% larger than those of Davalath and Bayazitoglu [11], but this discrepancy was also noted by Young and Vafai [14] and justified in terms of the reduced numerical grid employed in the original work.

The most convenient procedure to obtain the conjugate coefficients, either by numerical simulations or from experiments, is to perform tests with a single active heater of the array at a time. Considering for example that only the jth heater of the array is active, the temperature increase of the kth heater above the flow inlet temperature T_0 may be expressed, from Eq. (1), as

$$(T_k - T_0) = \left(\frac{q_j}{\dot{m}c_p}\right) g^+_{kj} \tag{2}$$

The term in parenthesis on the right side of Eq. (2) represents the flow mixed mean temperature rise due to the conjugate cooling rate from the jth heater. Due to the discrete heating in the duct and the conduction through the substrate plate, the resulting flow temperature rise is not uniform in the duct cross section. The conjugate coefficient g^+_{kj} represents the ratio of the actual kth heater temperature rise and the mixed mean flow temperature rise due to the active jth heater conjugate cooling rate. The conjugate coefficients are very convenient for two main reasons. First, the temperature increase of any heater of the array is relative to the flow inlet temperature in the duct. Second, they are invariant with the conjugate cooling rate from any heater of the array (under the restrictions of the same geometry, fluid and substrate plate properties, and flow rate). Conjugate coefficients were also obtained from experiments by Loiola and Altemani [25,26], considering either a single or two protruding heaters mounted on the lower conductive wall of a rectangular duct cooled by parallel forced