CONJUGATE COOLING OF A PROTRUDING HEATER IN LAMINAR CHANNEL FLOW

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Abstract. *The conjugate forced convection-conduction heat transfer from a two-dimensional protruding heater mounted to a finite thickness wall of a parallel plates channel was investigated numerically, using the control volumes method together with the SIMPLE algorithm. A uniform heat generation in the protruding block was transferred either directly to an airflow in the channel by convection or to the substrate wall under the heater by conduction. The considered airflow was steady, laminar, hydrodynamically and thermally developing, with constant properties. This problem is related to the cooling of electronic components assembled on printed circuit boards (PCB). The board acts as a substrate in contact with the heater, providing a path for conductive heat spreading through the wall and then back to the airflow by convection at the substrate interface. The substrate convective heat transfer upstream of the 2-D heated block preheats the airflow before it reaches the heater surfaces. Due to this thermal wake effect, it is convenient to treat the direct convection from the heater surfaces to the airflow by the adiabatic heat transfer coefficient. The conjugate problem was solved numerically within the solution domain comprising both the fluid and solid regions. The heater was considered with a high thermal conductivity and the results were obtained to show the effects of parameters such as the channel flow Reynolds number; the heater height in the channel; the substrate thickness; and the substrate to fluid thermal conductivities ratio. The results indicated that within typical ranges for such problems, there was a heat transfer enhancement from the heater at the expense of an increase of its average adiabatic surface temperature.*

Keywords*: conjugate heat transfer, adiabatic heat transfer coefficient, laminar channel flow, protruding heater, numerical analysis.*

1. INTRODUCTION

The purpose of the present work was to perform a numerical analysis of the conjugate forced convection and conduction cooling of a two-dimensional (2D) protruding heater in laminar channel flow. The heater, with a rectangular cross section, is mounted with perfect thermal contact on the lower wall of a horizontal channel, as indicated in Fig. 1. The channel wall is conductive, with thickness t and thermal conductivity k_s. A uniform heat generation per unit depth of this figure is considered in the protruding heater. It is transferred either directly by convection at the heater surfaces in contact with the channel flow or by conduction at its interface with the conductive wall. This conjugate cooling problem was investigated considering air as the cooling fluid and for a thermal conductivities ratio of the solid heater to the air equal to 500. The channel wall thickness was 1 mm and its thermal conductivity was investigated in the range from 10 to 80 relative to that of the air. At the channel entrance, the airflow has uniform velocity and temperature profiles. The heater height was either 15% or 30% of the channel height and its length in the flow direction was equal to the channel height. The rate of heat generation per unit depth was considered the same for both heater heights. This problem is related to the cooling of electronic components assembled on a printed circuit board (PCB). The board wall acts as a heat spreader and it is used as a cooling path to the mounted components.

The conductive flow from the heater to the board occurs along both the upstream and the downstream directions, causing two opposite effects on the heater cooling. First, the conductive flow in the upstream direction preheats the airflow and is responsible for a thermal wake impinging on the heater surface. This effect corresponds to an upstream heating condition, reduces the direct convective transfer from the heater and contributes to a higher heater temperature. Due to this effect, it is convenient to treat the direct convection from the heater surfaces to the airflow by the adiabatic heat transfer coefficient. The opposite effect is due to heat spreading on the board by conduction - it increases the effective heat transfer area to the airflow and thus contributes to decrease the heater temperature. If this effect prevails over the first one, the conductive board will enhance the heater cooling, as compared to the case of an adiabatic board.

The convective heat transfer was characterized by the adiabatic heat transfer coefficient h_{ad}, due to its independence on the thermal boundary conditions. This concept was introduced by Arzivu and Moffat (1982) from experiments in electronics cooling and extended by subsequent works of Arzivu *et al*. (1985), Anderson and Moffat (1992a, 1992b), Moffat (1998) and Moffat (2004). They showed that the adiabatic heat transfer coefficient is an invariant descriptor of the convective heat transfer. It is independent of the thermal boundary conditions, being a function only of the geometry and flow characteristics. A review of this concept was also presented by Alves and Altemani (2007). Using h_{ad}, the convective heat transfer still depends on the heater size and position in the channel.

A comprehensive review of the work on conjugate forced convection and conduction heat transfer from electronic components mounted on circuit boards was presented by Nakayama (1997). Experimental and numerical works considering distinct heat sources geometries and their arrangement on the circuit board have been addressed in the literature. Here, the emphasis will be directed to numerical investigations of protruding heaters in laminar channel flow.

Davalath and Bayazitoglu (1987) presented a pioneering numerical investigation of the conjugate forced convection and conduction cooling of three heated blocks mounted to the lower conductive wall of a horizontal channel. They performed a two-dimensional analysis considering both a conducting plate and an insulated plate for laminar flow in the channel. The conductive wall had a thickness relative to the channel height equal to 0.10 and both heater and wall had a thermal conductivity relative to the air equal to 10. The Reynolds number was based on the channel height and the associated average fluid velocity and its effect was investigated in the range from 100 to 1500. At the channel entrance the flow was assumed with fully developed velocity profile and the temperature was uniform. The length and the height of the heated blocks relative to the channel height were respectively equal to 0.5 and 0.25 and the convective heat transfer coefficient was based on the fluid inlet temperature in the channel. The channel entrance length extended 3.5 channel heights upstream of the first heater, while the corresponding length downstream the third heater was equal to 9.5 channel heights. They considered a uniform heat generation in each block and presented the local Nusselt number variation along their surfaces. Simulations for three values of the Prandtl number (0.1, 0.7 and 2.0) were performed and its effect was included in a correlation for the average *Nu* number of each heated block, for the insulating and for the conductive plates.

Kim and Anand (1995) studied the conjugate forced convection and conduction cooling of an array of heated blocks mounted on a wall of a stack of parallel plates to a laminar flow. They considered only the condition of a periodic fully developed laminar flow and heat transfer from an array of blocks with uniform heat dissipation in each block. This condition is attained as the number of heated blocks in the flow direction is increased. The Reynolds number was based on the channel flow average velocity and hydraulic diameter, with simulations performed in the range from 100 to 2000. The calculation domain length comprised a module with one heated block and the spacing between two consecutive blocks. A Nusselt number was based on the fluid inlet bulk temperature upstream of each module and the results were presented for both local and average values. The simulations were performed for five ratios of the heated block length to the channel height in the range from 0.67 to 2 and four values of the heated blocks height relative to the channel height in the range from 0.2 to 0.67. The heated blocks thermal conductivity was 500 times that of the air, while the substrate wall relative thermal conductivity was varied in the range from 0.1 to 50. The results were obtained considering the air as the cooling fluid and the average Nusselt number was expressed as a correlation of the Reynolds number, the relative substrate wall thermal conductivity and the geometric parameters. Since the Nusselt number describes only the convective heat transfer from the heated blocks, an overall thermal resistance was also defined and the results included also correlations for the adiabatic and the conductive substrate wall.

Kim and Anand (1994) considered a stack of parallel plates with protruding heaters and imposed repeated boundary conditions in the cross flow direction, in order to analyze just one channel. A uniform velocity and temperature profiles were considered at the channel inlet, under laminar flow conditions. Five protruding heaters with the same heat generation were mounted on a channel wall and one-half of a sixth heater was used as a thermal outlet boundary condition. The flow conditions and geometry were analogous to those presented by Kim and Anand (1995). The channel wall length upstream the first heater leading edge was equal to four heater widths from the channel entrance and it was considered adiabatic. Thus, upstream the first heater wall conduction effects were not considered in this work. An overall thermal resistance was defined as the temperature difference between the maximum temperature inside a block and a reference temperature, with respect to the rate of heat generation in the block. Two reference temperatures were employed: the channel inlet temperature and the inlet mixed mean temperature for a module comprising each heater. The results indicated that for the lowest Reynolds number (Re=100) the periodically fully developed flow and heat transfer conditions were attained within just a few rows. For higher Reynolds, these conditions may require a much larger number of rows. It was also verified that the substrate wall thermal conductivity contributes to decrease the number of rows needed to attain the fully developed conditions.

Young and Vafai (1997) performed a numerical investigation of the laminar forced convective cooling of a single heated obstacle mounted upon an insulated channel wall. The basic obstacle geometry comprised two heights (0.125 and 0.25 channel height) and three widths (0.125, 0.25 and 0.5 channel height), while its thermal conductivity relative to the air varied from 1 to 6600. The obstacle leading edge was located two channel heights downstream the channel entrance, and the relative channel length downstream of the obstacle trailing edge was equal to 8. At the channel entrance the temperature was uniform and the air velocity profile was parabolic. The Reynolds number was based on the channel hydraulic diameter and the average air velocity at the channel entrance. They presented streamlines and temperature profiles for the flow in the channel and indicated a long recirculation zone which increases with the Reynolds number downstream the obstacle. A local Nusselt number based on the inlet flow temperature was evaluated along the heater perimeter in contact with the channel flow. The averages of the local *Nu* over each of the three heater surfaces and the average *Nu* for the heater were also presented, indicating a general trend of decrease of the heater average *Nu* as the heater size (height and length) increased. Two distinct thermal boundary conditions were tested. One was a uniform heat flux at the heater base and the other considered a uniform heat generation within the heater. The local and the average *Nu* were about the same for both conditions, but the maximum heater temperature was lower for the volumetric heating.

The present considered a single heater mounted on a conductive channel wall. The investigation considered the effects of a laminar airflow rate, the solid substrate plate thermal conductivity and the heater height in the channel, for a uniform volumetric heat generation within the heater. The single heater, with a length equal to the channel height, was positioned in the channel with its leading edge centered on the substrate plate. The thermal effects of wall conduction upstream the heater were treated by means of the adiabatic heat transfer coefficient.

2. ANALISYS

2.1. Problem formulation and heat transfer parameters

The solution domain of the present investigation, comprising solid and fluid regions, is indicated in Fig. 1. A single two-dimensional heater is mounted on a conductive substrate in a horizontal channel. The heater length L_b was equal to the channel height and two heater heights were considered in the analysis, equal to 0.15 or 0.30 the channel height H_b.

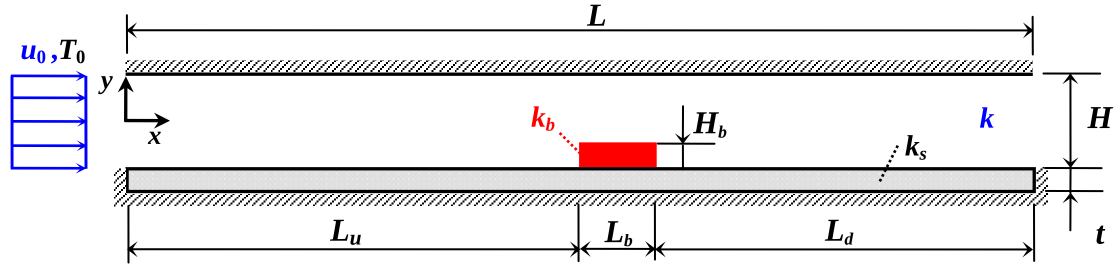


Figure 1. Domain of the conjugate heat transfer problem.

At the channel entrance, the airflow is laminar with constant properties and a uniform velocity and temperature profiles. The airflow velocity u_0 and the channel hydraulic diameter defined the Reynolds number

$$\mathrm{Re} = \frac{\rho u_0 2H}{\mu} \tag{1}$$

The simultaneous development of the velocity and temperature profiles in the solution domain was obtained in laminar flow from the numerical solution of the conservation equations of mass, momentum and energy. For the fluid region these equations were

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{2}$$

$$\rho\left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y}\right) = -\frac{\partial p}{\partial x} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) \tag{3}$$

$$\rho\left(u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y}\right) = -\frac{\partial p}{\partial y} + \mu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right) \tag{4}$$

$$\rho c_p\left(u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y}\right) = k\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \tag{5}$$

In the solid regions, comprising the substrate plate and the heater, due to the absence of flow, only the energy equation had to be solved, including the uniform heat generation within the heater.

The flow boundary conditions were a uniform distribution at the channel inlet and zero-velocity at all the solid interfaces. The outflow boundary was treated with a negligible diffusion for the u-component of velocity and a zero velocity for the v-component. The thermal boundary conditions were a uniform flow temperature T_0 at the channel inlet and an adiabatic condition at the upper and lower channel surfaces, and at both ends of the substrate plate, as indicated in Fig. 1. At the outflow boundary, the energy equation was also treated by the assumption of negligible diffusion. The source of energy was a rate of heat generation equal to q in the protruding heater, independent of its height. It was provided by a uniform volumetric heat generation q''' in the heater, so that per unit depth normal to Fig. 1, $q= q'''(H_bL_b)$. A perfect conductive thermal contact was considered at the interface of the heater with the substrate plate.

The cooling fluid was always air, with constant properties taken from a table (Kays and Crawford, 1993) at 27 °C for the simulations performed. The heater thermal conductivity was considered equal to 500 times that of the air, while the conductivity of the substrate plate was varied parametrically in the range from 10 to 80 that of the air. Additional data were obtained considering an adiabatic substrate plate, in order to obtain the adiabatic heat transfer coefficient.

The effects of the Reynolds number defined by Eq. (1) were investigated for five values in the laminar regime, in the range from 630 to 1890. These values correspond, for a channel height H = 0.01 m, to average airflow velocities in the range from 0.5 m/s to 1.5 m/s in the channel.

Along the heater perimeter in contact with the airflow, a local convective heat transfer coefficient and Nusselt number were defined by

$$h_0(\xi) = \frac{q_f''(\xi)}{T_i(\xi) - T_0} \tag{6a}$$

$$Nu_0(\xi) = \frac{h_0(\xi)\, L_b}{k} \tag{6b}$$

The temperature $T_i(\xi)$ is that along the heater interface with the airflow and T_0 is the inlet airflow temperature in the channel. The local heat flux $q_f''(\xi)$ was evaluated from the numerical solution of the temperature field in the calculation domain, invoking the temperature and heat flux continuity at the heater-airflow interface. The simulations were performed for both the conductive and the adiabatic substrate plate. Those with the adiabatic substrate provided the local adiabatic convective heat transfer coefficient $h_{ad}(\xi)$ and the corresponding local adiabatic Nusselt number $Nu_{ad}(\xi)$. In this case the adiabatic heater surface temperature $T_{ad}(\xi)$ is uniform, equal to the inlet flow temperature and all the generated heat is transferred to the airflow by convection.

The rate of direct convective heat transfer q_f from the heater to the airflow and the heater average surface temperature $\overline{T}_b$ were employed to define an average convective heat transfer coefficient and Nusselt number, as follows.

$$\overline{h}_0 = \frac{q_f}{A_b(\overline{T}_b - T_0)} \tag{7a}$$

$$\overline{Nu}_0 = \frac{\overline{h}_0\, L_b}{k} \tag{7b}$$

Considering the adiabatic substrate, the values obtained from Eq. (7) were respectively the adiabatic average heat transfer coefficient $\overline{h}_{ad}$ and the adiabatic Nusselt number $\overline{Nu}_{ad}$ for the heater.

For a conductive substrate, the heater average surface temperature rise above the inlet flow temperature $\left(\overline{T}_b - T_0\right)$ is due to two effects. The direct convective heat transfer rate q_f from the heater to the airflow is responsible for the heater temperature rise above its adiabatic temperature, $\left(\overline{T}_b - \overline{T}_{ad}\right)$. The heat transfer rate q_u conducted through the substrate wall upstream of the heater returns by convection to the airflow. This effect gives rise to a thermal wake responsible for the heater adiabatic temperature rise above the inlet flow temperature, $\left(\overline{T}_{ad} - T_0\right)$. These two effects may be added as follows.

$$\left(\overline{T}_b - T_0\right) = \left(\overline{T}_b - \overline{T}_{ad}\right) + \left(\overline{T}_{ad} - T_0\right) \tag{8}$$

Due to imperfect fluid mixing in the channel, the adiabatic temperature rise $\left(\overline{T}_{ad} - T_0\right)$ is always greater than the corresponding fluid mixed mean temperature rise ($T_{m,u}$–T_0). The ratio of these two temperature differences defined the coefficient g_u^*. An energy balance in the flow region upstream the heater related ($T_{m,u}$–T_0) to q_u and the channel airflow rate $\dot{m}$:

$$\left(\overline{T}_{ad} - T_0\right) = \left(T_{m,u} - T_0\right) g_u^* = \frac{q_u}{\dot{m} c_p} g_u^* \tag{9}$$

Similarly, the heater average temperature rise $\left(\overline{T}_b - \overline{T}_{ad}\right)$ was related by an influence coefficient g_h^* to the corresponding fluid mixed mean temperature rise $(\Delta T_m)_b$, which in turn was expressed in terms of q_f and the flow rate $\dot{m}$.

$$\left(\overline{T}_b - \overline{T}_{ad}\right) = \left(\Delta T_m\right)_b\, g_h^* = \frac{q_f}{\dot{m} c_p} g_h^* \tag{10}$$

Substituting Eq. (9) and (10) into Eq. (8),

$$\left(\overline{T}_b - T_0\right) = \frac{q_f}{\dot{m} c_p} g_h^* + \frac{q_u}{\dot{m} c_p} g_u^* \tag{11}$$

Defining a dimensionless heater average temperature $\overline{\theta}_b = (\overline{T}_b - T_0)\,k/q$ and the Peclet number $Pe = RePr$, Eq. (11) may be expressed in the dimensionless form

$$\overline{\theta}_b = \frac{2}{Pe}\left(\frac{q_f}{q_h} g_h^* + \frac{q_u}{q_h} g_u^*\right) \tag{12}$$

The fractions (q_f/q) and (q_u/q) and the upstream coefficient g_u^* depend on the conduction through the substrate and they were obtained for each tested condition. On the other hand, the convective coefficient g_h^* depends on the channel geometry and the flow conditions, so that it was obtained from numerical tests considering an adiabatic substrate wall.

2.2. Numerical solutions

The stated conservation equations were solved simultaneously within the conjugate domain indicated in Fig. 1, comprising the solid substrate and the flow regions. The control volumes method (Patankar, 1980) was employed with the SIMPLE (Semi-Implicit Method for Pressure Linked Equations) algorithm in order to obtain the fluid flow. The solution for the solid regions was obtained considering a very large viscosity, so that the resulting velocities were negligible in the substrate plate and the heater. At the solid fluid interfaces, use was made of the harmonic mean for the evaluation of the diffusion coefficients at neighboring control volumes, in order to handle the abrupt changes of the solid and fluid properties. The mentioned boundary conditions for the flow and for the temperature distributions were imposed at the four boundaries of the domain described by Fig.1.

The results were obtained with a non-uniform two-dimensional numerical grid deployed on the solution domain, comprising 348 control volumes in the flow direction and 34 control volumes in the transversal direction. In the flow direction, the grid was most refined over the heater, which contained 128 uniformly distributed control volumes. Along the upstream length L_u, 170 control volumes were uniformly deployed, and 50 others were distributed along the downstream length L_d. Along the *y*-direction, the grid consisted of 34 non-uniform control volumes. The grid was finer around the solid fluid interfaces due to larger gradients of the dependent variables near these regions.

Several grids were tested before the final distribution was selected to obtain the numerical results. The results for distinct grids were obtained under the conditions of Re = 1260, initially with the adiabatic substrate. The number of grid points uniformly was increased until further grid refinement practically did not change the obtained results. The Richardson extrapolation technique (De Vahl Davis, 1983) was employed for a particular case, indicating that the numerical error associated with the grid spacing was quadratic. In the simulations with a conductive substrate, it was verified that uniform grids with 4 control volumes per mm in the solid regions were enough to obtain results independent of further grid refinement.

The iterative solution process was interrupted when the absolute changes of the heater average temperature $\overline{\theta}_b$, the average Nusselt number $\overline{Nu}_0$, and the heat flow ratio (q_s/q) between two consecutive iterations were smaller than 10^{-5}. The numerical results were obtained in a microcomputer (Pentium D processor 2.8 GHz and 1GB RAM), in about 10 minutes for a typical solution considering a conductive substrate.

3. RESULTS

The numerical results were obtained considering a channel length L = 0.2 m and the plates spacing H = 0.01 m. The heater length was L_b = 0.01 m, with its upstream edge at L_u = 0.1 m from the channel entrance. Two heater heights were considered, equal to 1.5 mm and 3 mm, corresponding respectively to 15% and 30% of the channel height. A single value was adopted for the substrate thickness, t = 1 mm. The effect of the substrate conductivity was investigated for five values of (k_s/k) in the range from 10 to 80, while the heater conductivity was chosen with (k_b/k) = 500. Five values of the channel flow Reynolds number were considered in the range from about 600 to 1900 (average air velocities from 0.5 m/s to 1.5 m/s), always in the laminar regime. The air properties were obtained from tabulated values at 300 K.

The simulations for an adiabatic substrate plate $(k_s/k = 0)$ are very important because they constitute the easiest way to get the adiabatic heat transfer coefficient. In this case the heater adiabatic surface temperature becomes the inlet channel flow temperature, $\overline{T}_{ad} = T_0$, and all the heat is transferred directly by convection to the airflow, $q_f = q$. The heater interface temperature with the airflow was obtained by interpolation from the numerical temperature distributions in the control volumes adjacent to the heater surface. The average values, obtained by integration over the heater surface, for the case of an adiabatic substrate are presented in dimensionless form in Fig. 2, for the two heater heights considered in this work. They indicate a decrease of the heater temperature with the Reynolds number and are represented within 1% by the following correlation:

$$\overline{\theta}_{b,ad} = 0.752\,\mathrm{Re}^{-0.421}\left(\frac{H_b}{H}\right)^{-0.250} \tag{13}$$

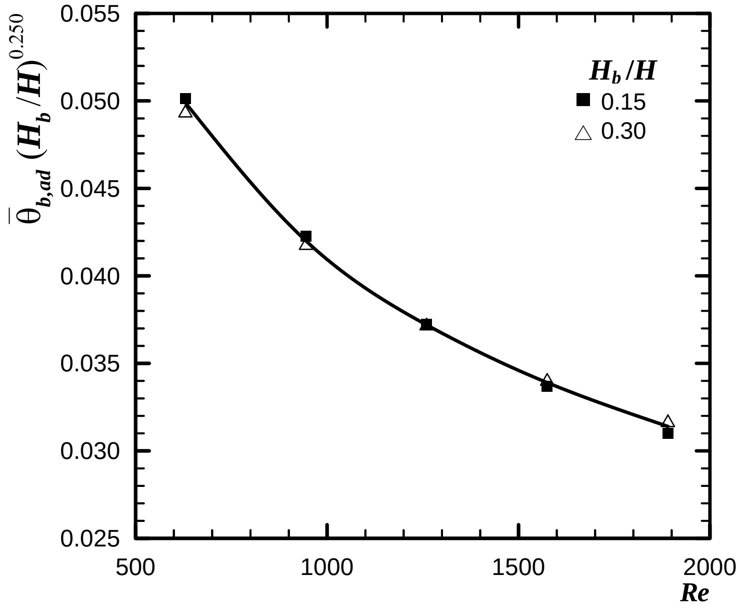


Figure 2. Heater dimensionless average adiabatic surface temperature.

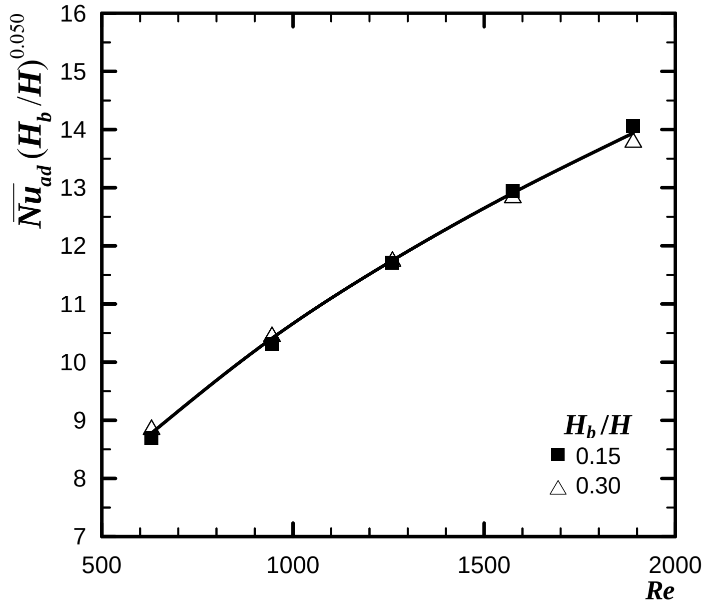
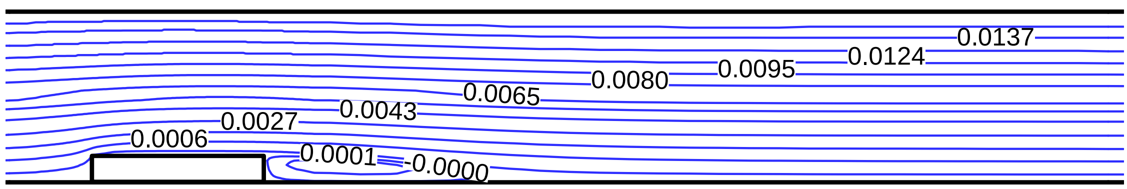


Figure 3. Average adiabatic Nusselt number on the heater surface.

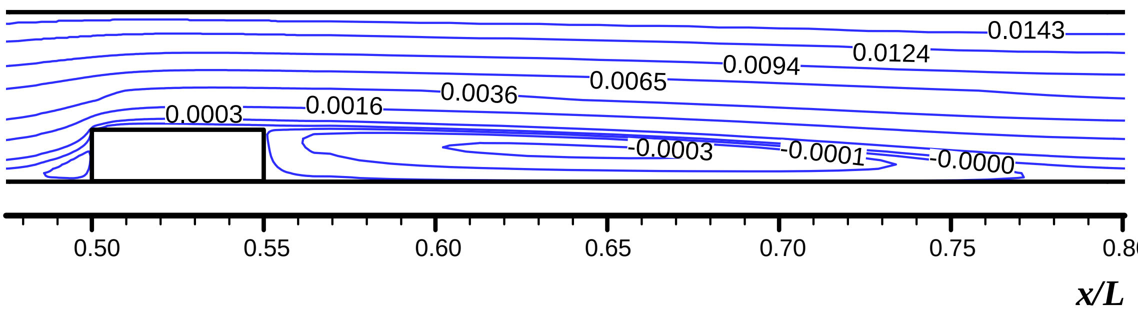
The heater average adiabatic Nusselt number $\overline{Nu}_{ad}$ was obtained from $\overline{\theta}_{b,ad}$, according to Eq. (7). The numerical results of $\overline{Nu}_{ad}$ increase with the Reynolds number, as indicated in Fig. 3 and they are independent of the substrate conductivity. They were correlated within 1 % for both heater heights, by

$$\overline{Nu}_{ad} = 0.582\,\mathrm{Re}^{0.421}\left(\frac{H_b}{H}\right)^{-0.050} \tag{14}$$

For the adiabatic substrate, typical streamlines and isotherms are presented in Figs. 4 and 5. Their shapes are similar to those obtained by Young and Vafai (1997) and they explain the local adiabatic Nusselt number distributions presented in Fig. 6 for both heater heights. Both $Nu_{ad}(\xi)$ distributions are quite similar and the local values increase with the Reynolds number.

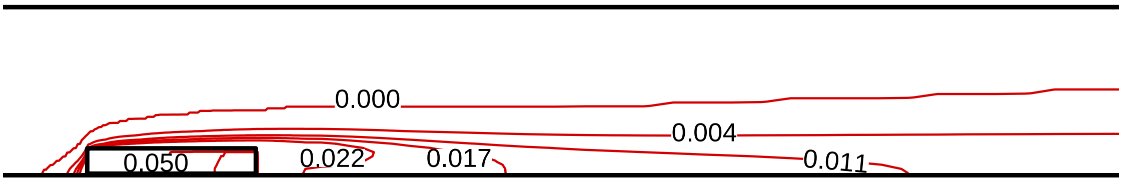


(a) Re = 1890 and H_b/H = 0.15

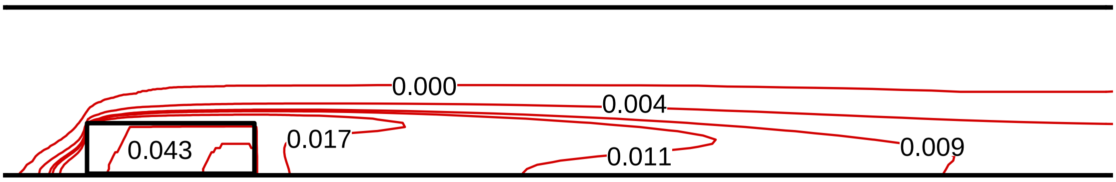


(b) Re = 1890 and H_b/H = 0.30

Figure 4. Streamlines.



(a) Re = 1890 and $H_b/H = 0.15$



(b) Re = 1890 and $H_b/H = 0.30$

Figure 5. Isotherms for adiabatic substrate.

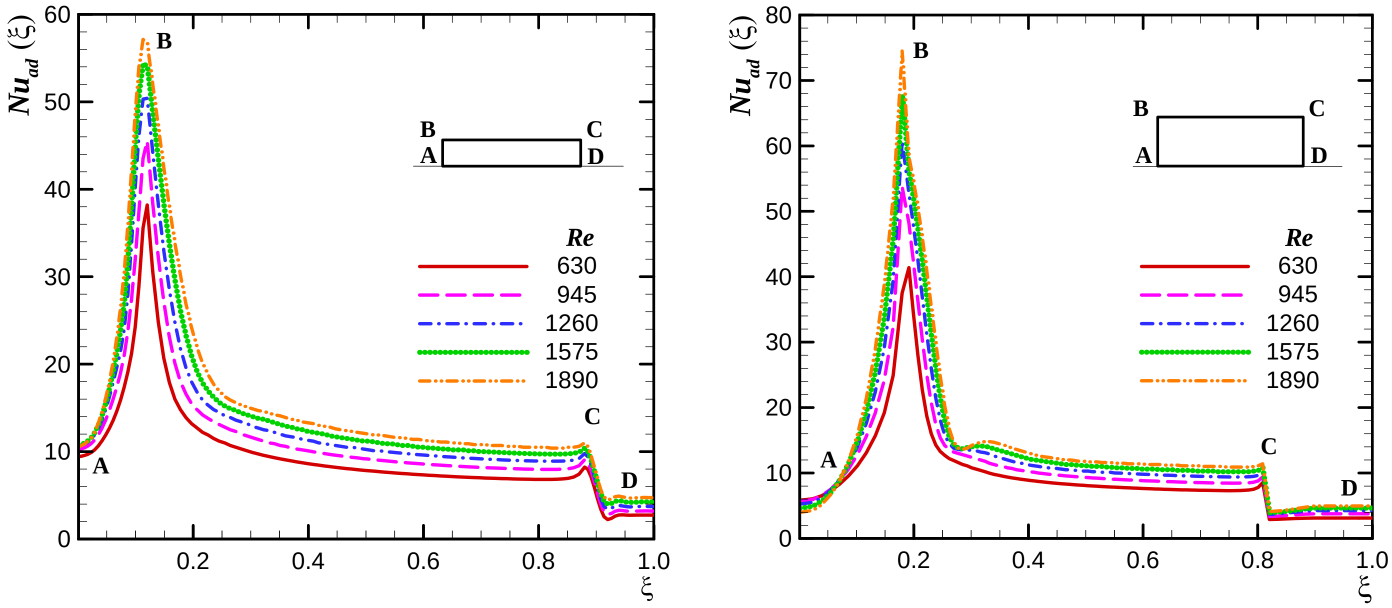


Figure 6. Local adiabatic Nusselt number distributions on the heater surface.

The influence coefficient g_h^*, defined by Eq. (10), was obtained from the simulations for the adiabatic substrate, but it is also valid for a conductive substrate. The results are presented in Fig. 7 and they indicate an increase of g_h^* with *Re*, mainly due to larger mass flow rates. This coefficient was correlated within 1 % for both heater heights by

$$g_h^* = 0.266\,\mathrm{Re}^{0.579}\left(\frac{H_b}{H}\right)^{-0.250} \tag{15}$$

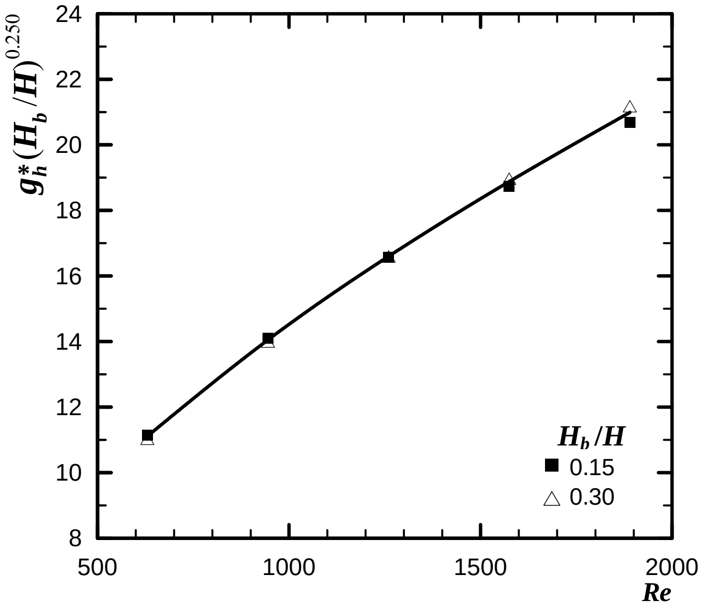


Figure 7. The coefficient g_h^* and the associated correlation..

For a conductive substrate, the heater cooling occurs simultaneously by forced convection directly to the airflow and by conduction to the substrate plate. The substrate conduction was investigated considering five values of the thermal conductivities ratio (k_s/k) in the range from 10 to 80. Within this range, the fraction (q_s/q) of the total heat generation which is conducted from the heater to the substrate plate was evaluated, with the results shown in Fig. 8. It always increased with the substrate conductivity, but decreased with the Reynolds number, due to the associated increase of the direct convective transfer with the airflow rate. The fraction (q_f/q) transferred directly by convection from the heater to the airflow is just the complement of (q_s/q) to one. The numerical data of (q_s/q) for all the tests were within 10% of a correlation of the form indicated by Eq. (16). The coefficients of this equation are presented in Tab. 1.They were obtained by the least squares method not only for (q_s/q), but also for (q_u/q), g_u^* and $\overline{Nu}_0$.

$$Variable = A\mathrm{Re}^B \left(\frac{k_s}{k}\right)^C \left(\frac{H_b}{H}\right)^D \tag{16}$$

Table 1. Coefficients of Eq. (16) for several variables.

Variable	***A***	***B***	***C***	***D***
q_s/q	0.161	-0.238	0.486	-0.339
q_u/q	0.130	-0.288	0.486	-0.435
g_u*	0.341	0.562	-0.139	-0.150
Nu_0	0.464	0.479	-0.069	0.016

The rate of heat conduction from the heater to the substrate plate through their interface is directed upstream and downstream of the heater. The portion conducted in the upstream direction is represented in dimensionless form by the fraction (q_u/q). The numerical results indicated in Fig. 9 presented a trend to decrease with the airflow, a behavior similar to that observed for (q_s/q). A correlation of the form indicated by Eq. (16) was obtained for (q_u/q).

The heat conducted by the substrate plate upstream of the heater returns to the airflow, causing a thermal wake impinging on the heater surface. Due to this effect, the heater mean adiabatic surface temperature increases above the inlet flow temperature T_0. This temperature increase was described in dimensionless form by the coefficient g_u^* defined by Eq. (9). The results for this coefficient are indicated in Fig. 10, showing its increase with the airflow rate. A fitting to all the numerical data of g_u^* was obtained in the form of Eq. (16), as indicated in this figure, correlating the results within 6%. The heater mean adiabatic surface temperature rise above T_0, as evaluated by Eq. (9), increased with the thermal conductivities ratio (k_s/k) and decreased with the Reynolds number and the relative height (H_b/H) of the heater in the channel. This result is shown in Fig. 11, in dimensionless form, Θ, indicating the fraction of the total heater mean surface temperature rise which is due to substrate conduction upstream of the heater.

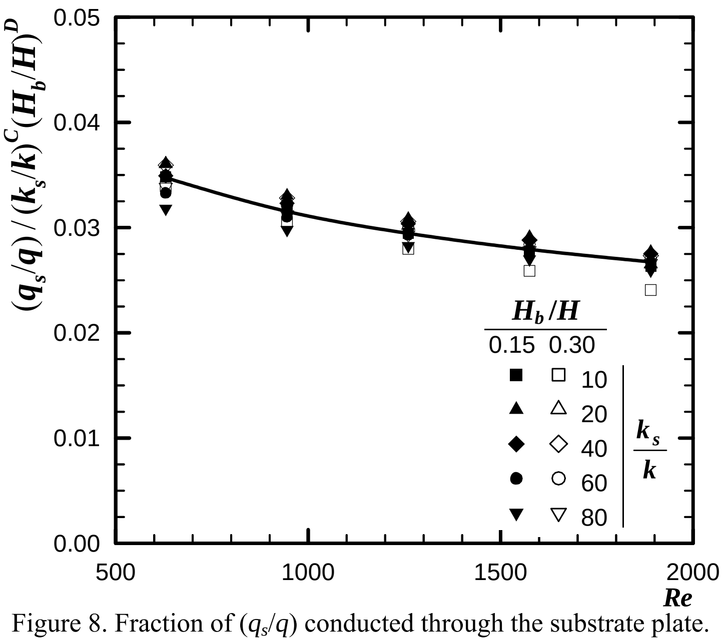


Figure 8. Fraction of (q_s/q) conducted through the substrate plate.

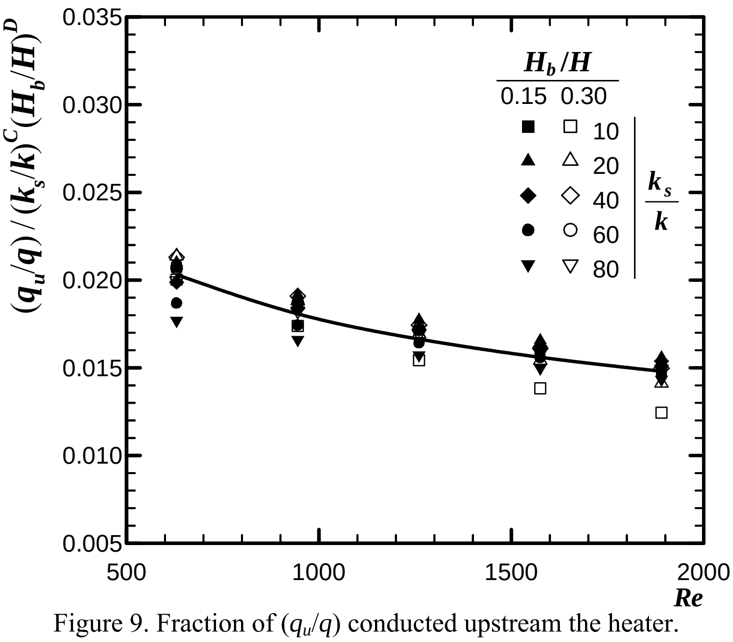


Figure 9. Fraction of (q_u/q) conducted upstream the heater.

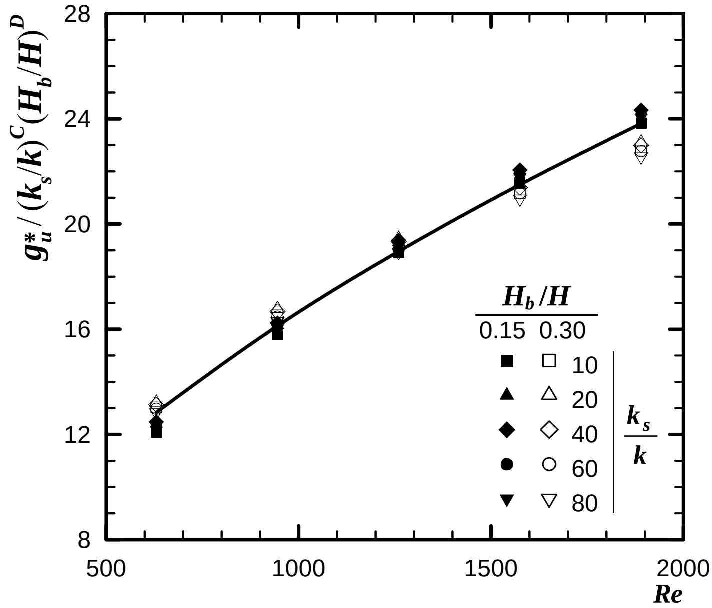


Figure 10. The coefficient g_u^* and the associated correlation.

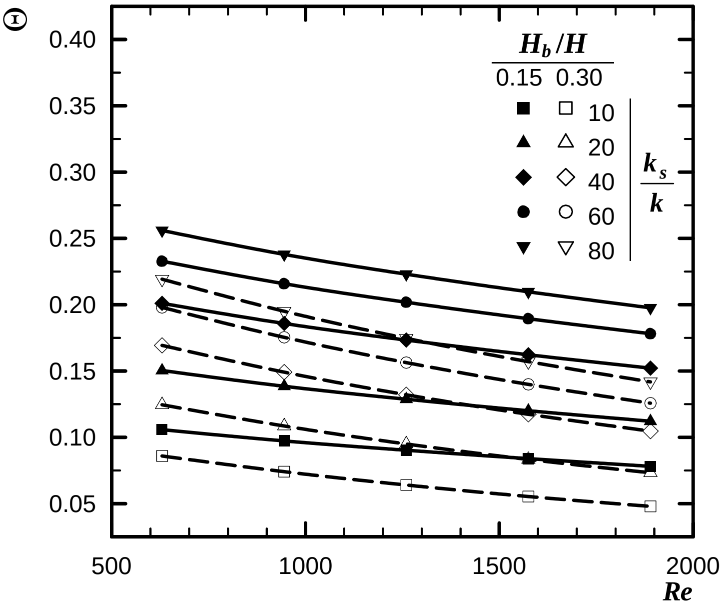


Figure11.The adiabatic temperature rise due to upstream conduction.

The heater mean surface temperature rise above T_0 may be predicted in dimensionless form by the previous coefficients (q_f/q), g_h^*, (q_u/q) and g_u^*, as described by Eq. (12). The obtained results are presented in Fig. 12 and they were correlated to the same parameters of these coefficients, in the form of Eq. (16). The resulting correlation indicates that this temperature decreases as the Reynolds number, the thermal conductivities ratio and the relative heater height in the channel are increased. This prediction is a function of the flow conditions, the heater and fluid properties, and depends on the channel and heater geometry, including the heater position in the channel. It does not depend, however, on the substrate properties and on the rate of heat generation. The independence of the substrate properties is due to the choice made of the heater mean adiabatic surface temperature as the reference temperature in the definition of the convective heat transfer coefficient. Had the uniform inlet flow temperature been chosen as the reference temperature, the resulting average Nusselt number on the heater surface would be $\overline{Nu}_0$. This average Nusselt number would be dependent on the substrate properties, in addition to the previously considered parameters. This average Nusselt number was also evaluated from the numerical simulations and it is presented in Fig. 13, were it is shown its dependence on the ratio (k_s/k), i.e., a substrate property.

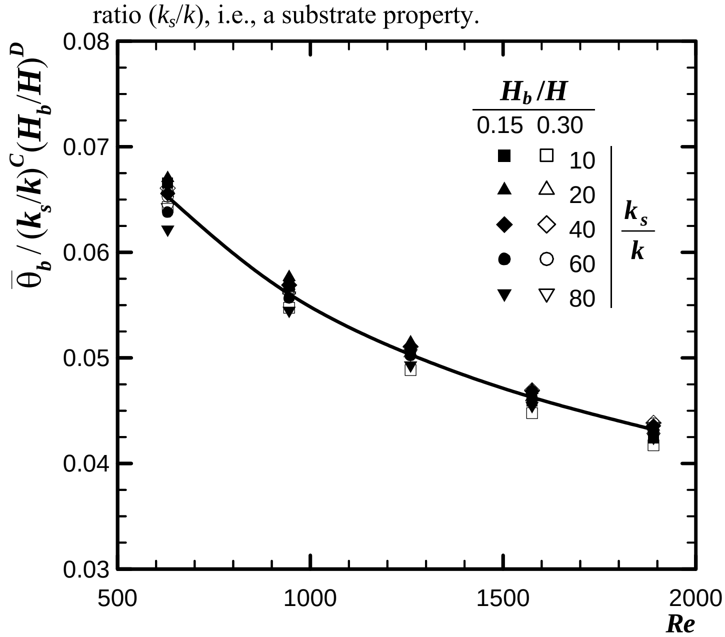


Figure 12. Dimensionless heater mean surface temperature rise above T_0.

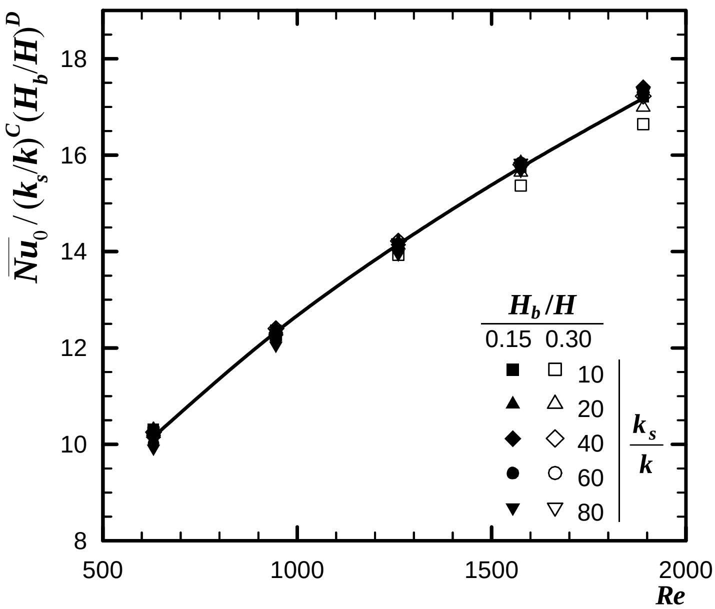


Figure 13. Average Nusselt number based on T_0.

4. CONCLUSIONS

A two-dimensional numerical investigation was performed of the conjugate forced convection and conduction cooling of a protruding heater mounted on a conductive substrate plate of a channel. The flow was in the laminar regime, with a uniform flow and temperature profiles at the channel entrance. Two distinct heater heights were considered in the analysis, with the same rate of heat input per unit depth, uniformly distributed in the heater volume. This problem depends on several parameters including the fluid, substrate and heater thermal properties, the flow and thermal conditions, the channel and heater geometry, and the heater position in the channel. A set of these parameters was selected for the present numerical investigation, as described. The obtained flow distribution was indicated by streamlines surrounding the heater in the channel and giving rise to a large recirculating flow downstream its rear face. Temperature profiles were also sown, indicating relatively large temperature gradients over the heater top surface. The consequence of this was shown in the form of the local Nusselt number distribution over the heater surface. The adiabatic heat transfer coefficient and the corresponding Nusselt number were employed in the analysis to obtain an invariant descriptor of the convective heat transfer, since it does not depend on the rate of heat dissipation or the substrate plate thermal properties. By comparison, the Nusselt number based on the uniform flow inlet temperature depends on the substrate thermal conductivity. The average heater surface temperature rise above the inlet flow temperature was obtained by a simple correlation from the concept of the adiabatic surface temperature and the adiabatic Nusselt number, which is needed to obtain the self-heating coefficient g_h*. The present analysis considers that the heater temperature rise is due to the contribution of two distinct heat transfer mechanisms. One is obvious, due to self-heating, i.e., the heat generation within the heater. The other, which may not be neglected, is due to the thermal wake effect originating from the conductive substrate plate. Next, this topic will be pursued further, with the consideration of additional heaters on the conductive substrate plate.

5. ACKNOWLEDGEMENTS

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