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THERMAL SIMULATION OF ELECTRONIC SYSTEMS WITH NON-UNIFORM INLET VELOCITIES

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ABSTRACT

The inlet flow conditions to microelectronics systems are generally assumed to be steady and invariant with respect to position when using a thermal simulation tool for predicting circuit board temperatures. However, inlet flow conditions can often be distorted as a result of flow obstructions, an abrupt turning of the cooling fluid at the intake to the system or due to the proximity of a cooling fan to the system inlet.

The latter problem of simulating the effects of fan induced swirling velocities is addressed in this paper. An analytical-numerical simulation routine, META, is used to predict wall temperatures along a flat rectangular duct with heated rectangular modules attached to one surface. The rotationally dominated inlet velocity, resulting from an axial fan located at the entrance to the duct, is approximated using a simple linear velocity variation across the principal flow direction.

Thermal simulations are compared to published experimental data obtained over a range of channel Reynolds numbers from 600 - 1800.

NOMENCLATURE

Bi	Biot number
H	channel height, m
k_f	fluid thermal conductivity, $W/(m \cdot K)$
N	number of elements in flow direction
Pr	Prandtl number
q_w	wall heat flux, W/m^2
Q	heat flow rate, W
Re	Reynolds number
Re_H	channel Reynolds number
T_w	wall temperature, K
T_0	inlet temperature, K
$\bar{u}$	bulk velocity, m/s
W	board width, m
x, y	planar Cartesian coordinates, m

Greek Symbols

ξ	dummy variable in the flow direction, m
χ	dimensionless position $\equiv \xi/x$
Λ	velocity profile factor

INTRODUCTION

The ever increasing trend towards miniaturization and increased power density levels in microelectronic components and printed circuit boards has introduced important reliability concerns at the systems level where thousands of watts of power must be dissipated over a relatively small control volume. A typical system will consist of a cabinet, which surrounds one or more shelves containing multiple rows of printed circuit boards. Circuit boards are separated by air spaces that form vertical channels which can be used for distributing a cooling fluid, such as ambient air, through the system. Air can be drawn into the cabinet, either through buoyancy effects due to air density differences across the cabinet or by means of an external source, such as a cooling fan. Air which enters the system, usually at room temperature, is heated as it sweeps the circuit boards, resulting in a higher sink temperature for down stream components. It is imperative to understand the heat distribution in microelectronic systems in order to insure that all components operate within their recommended temperature window for reliable, uninterrupted operation.

Many studies have been performed to simulate microelectronic systems in controlled laboratory settings. For these studies, machined metallic cubes are mounted along one wall of a flat rectangular duct, reminiscent of the channel formed by two circuit boards in a typical microelectronic system. Sparrow et al., 1982 performed forced convection experiments using brass and naphthalene modules attached to an aluminum substrate, in order to study the effect that missing modules and flow barriers had on heat transfer. Chang et al., 1992 used a numerical approach to study heat transfer and fluid flow around heat generating blocks of various sizes attached to a low conductivity substrate. A collection of experimental studies by Wirtz et al. (Wirtz and Dykshoorn, 1984, Lehmann and Wirtz, 1984 and Lehmann, 1984) examined the effect of module size and spacing and channel dimensions for heat generating aluminum rectangular blocks attached to low conductivity substrates formed from plexiglass and balsa wood. Common to all of these studies was the attention given to preserving uniform inlet velocities and temperatures. Unfortunately, actual microelectronic systems do not always benefit from these same ideal conditions, where velocities can vary across the inlet to the system due to turning effects, flow obstructions or the proximity of cooling fans to the system inlet.

The following investigation will examine the effect of these non-uniform inlet conditions on the surface temperature of board modules within the system. An analytical/numerical model (META) developed by the Microelectronics Heat Transfer Laboratory (Culham et al., 1991) is used to simulate the thermal performance of populated circuit boards where inlet velocities are varied across the width of the circuit boards. Thermal simulations obtained using META will be compared to experimental data obtained by Kim and Lee, 1991a and 1991b.

THERMAL MODEL

META (MicroElectronics Thermal Analyzer) combines an analytical boundary layer solution with a finite volume solid body model through an iterative procedure that uses the local heat transfer coefficient as the coupling condition along common interfaces. The fluid-side solution is based on the integral form of the boundary layer equations for flow over flat plates with arbitrary, flux specified boundary conditions. The solid-side solution takes advantage of the negligible temperature gradient across the thickness of a circuit board (i.e. Bi< 0.1), thereby allowing a two-dimensional finite volume solution to be used for calculating board temperatures. The discretized mesh used in the finite volume analysis has a uniform cell size based on the dimensions the smallest heat dissipating component on the surface of the circuit board. Once converged, the two solutions give a unique temperature profile at the fluid-solid interface which simultaneously satisfies the energy equation in both the fluid and the solid domains.

In addition to forced convection heat transfer from flat surfaces, META accounts for heat losses from raised bodies of arbitrary height attributed to forced, natural or mixed convection, channel flow conditions plus radiative heat losses between a circuit board and the surrounding circuit boards or walls of the system enclosure.

The effects of non-uniform inlet velocities can be accounted for in META by introducing an inlet velocity, $u_0(y)$, which allows for a velocity variation across the principal flow direction, at the inlet to the system.

As shown by Sparrow and Lin (1965), the integral form of the energy equation for steady, laminar flow with an arbitrarily specified heat flux at the wall can be solved such that

$$T_w(x)-T_0 = \frac{x}{3 \times 0.454 \cdot \mathrm{Re}(x)^{1/2}\mathrm{Pr}^{1/3} \cdot k_f} \int_0^1 q_w(\chi)\,[1-\chi]^{-2/3}\,d\chi \quad (1)$$

The integral in Eq. 1 can be easily solved if the distributed heat flux, $q_w(\chi)$, is approximated as a series of step changes, q_{w_i}. The linear behavior of the energy equation allows the individual wall temperature solutions for each step change in heat flux to be superimposed to obtain an overall temperature distribution along the full length of the plate. The accuracy of the calculated wall temperature solution is improved by increasing the number of step changes (N) to approximate the original heat flux distribution, $q_w(\chi)$.

$$T_w(x) - T_0 = \left\{\frac{x}{0.454 \cdot \mathrm{Re}(x)^{1/2}\mathrm{Pr}^{1/3} \cdot k_f}\right\} \times \sum_{i=1}^{N} \left[q_{w_i}\left\{[1-\chi_{(2i-1)}]^{1/3} - [1-\chi_{(2i)}]^{1/3}\right\}\right] \quad (2)$$

Equation 2 was obtained using a boundary integral solution for flow over an isolated flat plate where the velocity and temperature profiles within the boundary layers are assumed to have the form of a cubic parabola. However, for cases where the wall temperature in the entrance region of a flat rectangular duct is to be calculated, a core velocity based on conservation of mass over the region formed between the two developing boundary layers (Sparrow, 1954) can be used to calculate a representative velocity which can be used to determine the Reynolds number in Eq. 2.

$$u_c(x) = u_0 \left\{ \frac{1}{1 - \dfrac{3.48x}{H\sqrt{\mathrm{Re}(x)}}} \right\} \quad (3)$$

In cases where the inlet velocity, u_0, is uniformly specified i.e, independent of y, Eq. 3 is used for calculating the local wall temperatures, however, if the inlet velocity varies with respect to y, then u_c must be adjusted to allow for non-uniform inlet conditions.

For simplicity the non-uniform velocity at the inlet will be limited to linear variations across the width the channel. Therefore, Eq. 3 can be rewritten to reflect the y-dependence of the inlet velocity.

$$u_c(x,y) = u_0(y) \left\{ \frac{1}{1 - \dfrac{3.48x}{H\sqrt{\mathrm{Re}(x,y)}}} \right\} \quad (4)$$

where the implicit form of Eq. 4 requires that the Reynolds number used in the calculation of $u_c(x,y)$ be obtained from the previous iteration in the META simulation routine.

In this study, $u_0(y)$ is limited to linear variations of the form

$$u_0(y) = -\left(\frac{u_0(0) - u_0(W)}{W}\right) y + u_0(0) \quad (5)$$

The extent of the velocity variation at the inlet can be expressed in terms of the mean bulk velocity, $\bar{u}$, by using a velocity profile factor, (Λ), defined as

$$\Lambda = \frac{\Delta u_0}{\bar{u}} = \frac{u_0(0) - u_0(W)}{\bar{u}} \quad (6)$$

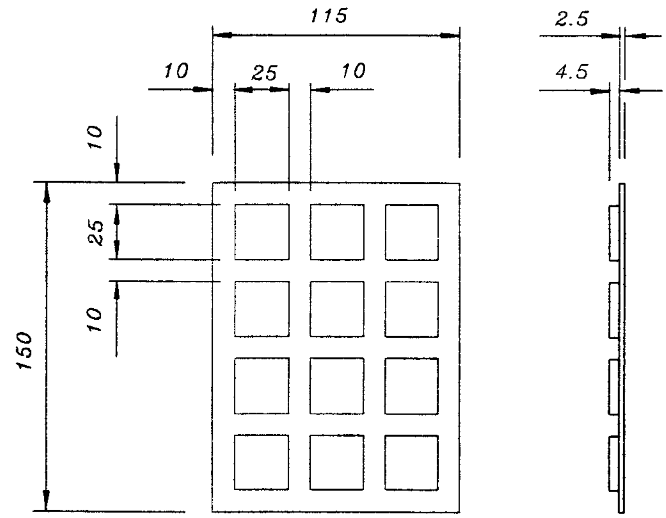
where the range of Λ is

$$-2 \leq \Lambda \leq 2 \quad (7)$$

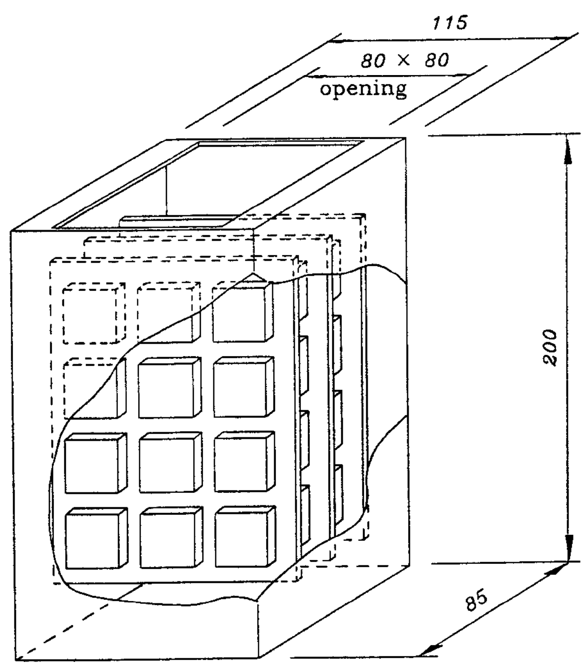
with $\Lambda = 0$ based on $\Delta u_0 = 0$, providing uniform inlet conditions and $\Lambda = -2$ and 2 being the case where the $u_0(0) = 0$ and $u_0(W) = 0$, respectively.

EXPERIMENTAL COMPARISONS

Lee and Kim, 1991a and 1991b, presented experimental data for a system level prototype with non-uniform swirling velocities at the inlet, resulting from fan induced flow. Their test apparatus, as shown in Fig. 1, consisted of three plexiglass boards, each containing arrays of twelve rectangular aluminum blocks. The boards were equally spaced in an open ended enclosure where the channel height between the top of the aluminum modules and the back surface of the adjacent board was maintained at 12 mm. The modules were laid out in a rectangular array, with each aluminum module being 25 mm × 25 mm × 4.5 mm. Only the modules on the center board were heated, using film heaters sandwiched between the aluminum blocks and the plexiglass board. An inlet opening of 80 mm × 80 mm containing an axial fan, used to introduce a swirling flow, was located 25 mm from



(all dimensions are in mm)



(all dimensions are in mm)

Figure 1: Experimental Test Apparatus

the leading edge of the boards. Channel Reynolds numbers were varied between 600 and 1800. Johnston, 1976 suggests that it is difficult to attain fully developed turbulent flow below a channel Reynolds number of 3000 and that developing lengths are of the order of 100 × H. Therefore, given the channel dimensions and the range of Reynolds numbers used by Lee and Kim, the air flow to the channels for the thermal simulations was considered to be non-uniform, laminar, developing flow with an ambient air temperature of 20 °C.

Lee and Kim did not quote actual channel inlet velocities, necessary for simulating the thermal performance of the system described above. Since it is the intent of this paper to demonstrate the capability of META to use an arbitrarily specified inlet velocity profile to predict circuit board temperatures, inlet velocities are estimated using system dimensions and mass flow rates estimated from channel Reynolds numbers specified by Lee and Kim.

Thermal simulations of Lee and Kim's apparatus were obtained using META with a uniform sized mesh base on 25 elements per module (5 mm × 5 mm per element) for a total of 690 elements.

Case 1: Uniform Inlet Conditions

Two examples of flat rectangular ducts that have uniform inlet velocities where one wall of the duct is populated with an array of aluminum cubes are taken from Wirtz and Dykshoorn, 1984 and Lee and Kim, 1991b. Each case presents a slightly different geometric configuration and heat distribution pattern.

Wirtz and Dykshoorn, 1984 measured the temperature of aluminum blocks in the thermal wake of a single heated element (3.68 W) located in the center of the second row from the leading edge of the board. The cubes were mounted on a composite wall consisting of 1.6 mm balsa wood and 12.5 mm plexiglass. A suction fan was used to introduce a uniform inlet velocity of 1.96 m/s. Temperatures were recorded for downstream modules using calibrated thermistors. The empirical data is presented as the bracketed values in Fig. 2. Thermal simulations using forced convection on the heated channel wall and natural convection on the back of the channel wall provided good agreement with experimental data. Temperatures, as calculated at the mid-point of the blocks using META, are presented as the non-bracketed values in Fig. 2. A uniform sized mesh base on four elements per module (12.7 mm × 12.7 mm per element) for a total of 800 elements was used for the META simulations.

Although the comparison with Wirtz and Dykshoorn is useful for assessing the ability of META to predict module temperatures in the flow direction, the use of a single heated element does provide much insight into the ability of META to predict temperatures across the principal flow direction. For this purpose, a study by Lee and Kim, 1991b in which the inlet conditions were maintained uniform by using a suction type blower to draw air through the test section, will be presented. The test section used for this experiment was identical to the test section used for the swirling inlet conditions, described above. The temperatures of the twelve modules on the center board were recorded using thermocouples attached to the underside of the modules. The total power dissipated was 24 watts, however, the actual resistive network used to distribute the power did not compensate

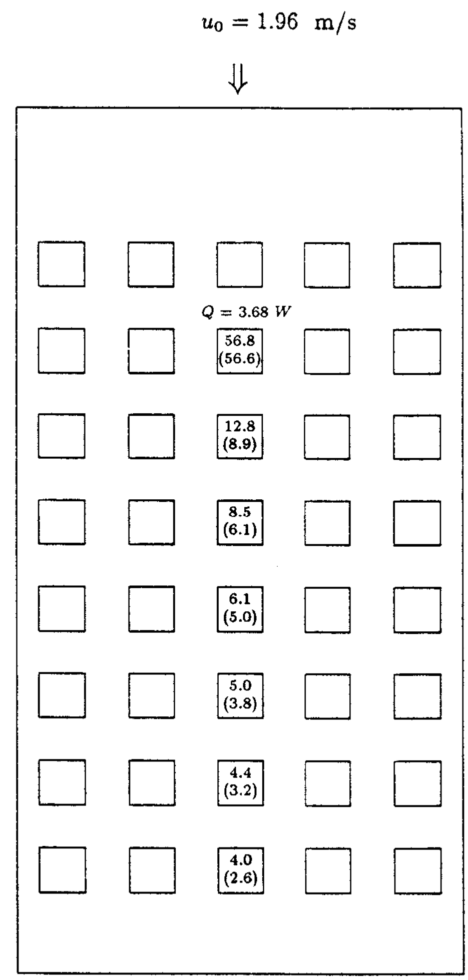


Figure 2: Thermal Wake Temperature Rise (°C) for Aluminum Modules in Channel Flow non-bracketed values - META simulation (·) - experimental data - Wirtz and Dykshoorn, 1984

for temperature sensitivity of the resistors, resulting in a somewhat non-uniform distribution of power to the individual heaters. The experimental data for this test is shown as the bracketed values in Fig. 3a. Despite uniform inlet conditions, a slight asymmetry is shown in the data for several reasons including the experimental error associated with using thermocouples and an inability to provide exactly 2 watts of power to each heater due to line losses associated with the power distribution network.

The uniform inlet condition example was simulated using META with all dimensions, thermophysical properties and surface emissivities as stated in Kim and Lee's papers, through personal communications with Kim and through actual measurement of the test section, as provided by Kim. A laminar flow model was used with an inlet velocity of 7.3 m/s.

The non-bracketed terms in Fig. 3a show the temperatures calculated using META with the uniform inlet velocity. The agreement between META and the experimental data is within 1 - 2 °C for rows 2 - 4, with a larger discrepancy of 4 - 5 °C for the first row along the leading edge of the board. The lack of symmetry in the experimental data might indicate that some non-uniformity still exists in the velocity profile across the channel which could account for some of the difference between experimental and predicted results.

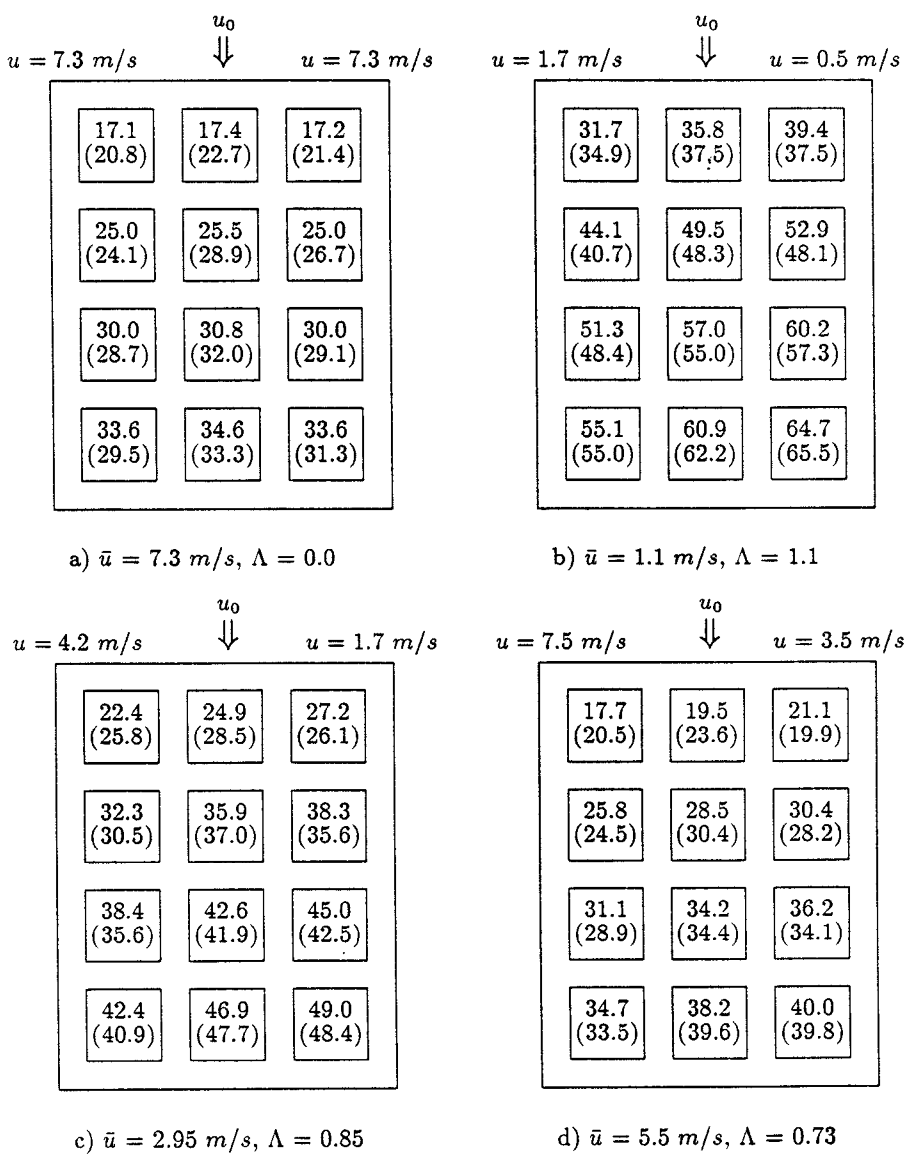


Figure 3: Module Temperatures Rises (°C) for Various Inlet Velocity Profiles
( ) - experimental data - Kim and Lee, 1991a and 1991b
non-bracketed values - META simulation

Case 2: Non-Uniform Velocity Inlet Conditions

All conditions as described in the uniform flow case of Kim and Lee were maintained with the exception that the forced flow was initiated using a single axial fan at the inlet to the test section. The axial fan introduced a rotationally dominated velocity which could be prescribed as a linear velocity variation across the leading edge of the heated board.

Experiments were conducted at three separate fan speeds resulting in channel Reynolds numbers of 596.4, 1192.8 and 1789.2. As shown in Fig. 3b, 3c and 3d, the swirl introduced by the inlet fan resulted in a consistent temperature variation over the board with higher temperatures obtained when moving from left to right across the surface of the board.

Inlet velocities were not precisely stated in the Kim and Lee papers for the non-uniform conditions. This is likely due to the difficulty in measuring velocities in a rotationally dominated flow. As a result flow velocities used in the simulations are estimated values that would be required to obtain good agreement with empirical data while still adhering to the physical constraints associated with the test section used in the experimental testing. The linear velocity variations across the channel inlet used for the simulations were as shown in Table 1.

Table 1: Inlet Velocity Profile for Non-Uniform Flow Examples

Re_H [Kim and Lee, 1991a and 1991b]	$u_0(0)$ m/s	$u_0(W)$ m/s	$\bar{u}$ m/s
596.8	1.7	0.5	1.1
1192.8	4.2	1.7	2.95
1789.2	7.5	3.5	5.5

DISCUSSION

In their papers, Kim and Lee, 1991a and 1991b, attribute the asymmetry in the temperature profiles, shown in the experimental data, to turbulent effects introduced by the swirling motion of the inlet fan. It is the purpose of this paper to suggest an alternate scenario, where the flow is assumed to remain laminar for channel Reynolds numbers less than 3000 and the asymmetry in temperature can be accounted for by skewing the inlet velocity profile as it enters the channel, to approximate the forced flow introduced by an axial fan.

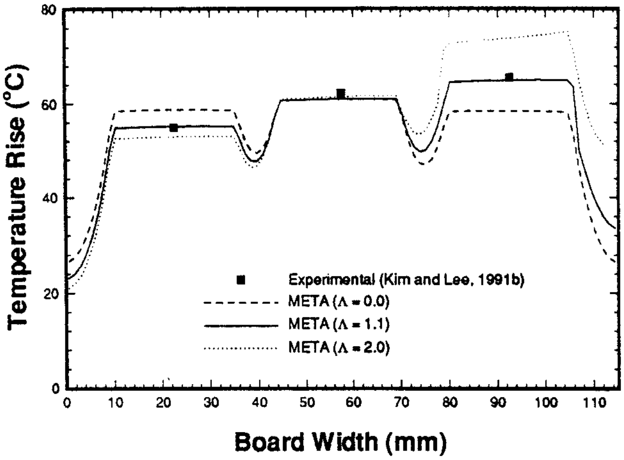
The comparisons between experimental and simulated data presented in Figs. 3b, 3c and 3d indicate that inlet velocity profiles can be selected that provide very good agreement. In general the agreement in rows 2 - 4 are much better than in the row along the leading edge of the board. The overall root mean square of the differences between experimental and simulated temperatures for the three cases presented is approximately 2.0 °C. This difference could be significantly improved by optimizing the shape of the inlet velocity profile, using something other than a linear representation. It is estimated that an optimize velocity profile would reduce the root mean square differences to less than 1.0 °C.

The non-uniformity in the inlet velocity profile is measured as a function of the bulk velocity by using the velocity profile factor (Λ), define in Eq. 6. The uniform flow example, shown in Fig 3a, has a $\Lambda = 0$. The other three non-uniform velocity examples shown in Figs. 3b, 3c and 3d have velocity profile factors ranging between 1.1 and 0.73.

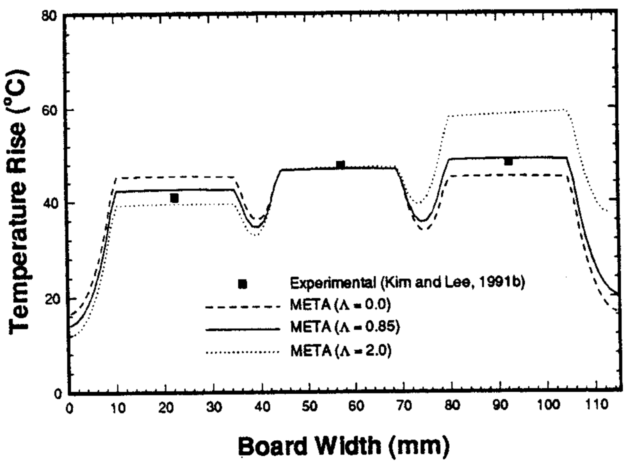
Figure 4 shows the effect that varying the shape of the velocity profile at the channel inlet has on the resulting temperature at the mid point of the last row on the circuit board. Three plots are presented for each of the mean velocities used in Figs. 3b - 3d, first a uniform inlet velocity ($\Lambda = 0$) where the temperature profile is symmetric about the centerline of the board. The maximum temperature occurs at the centerline, where the thermal resistance to conduction within the board is greatest. The second case shows the temperature profile for the actual conditions used in Fig. 3, while the third case is the limiting case ($\Lambda = 2.0$), where the inlet velocity at one edge is 0 m/s and the other edge has a velocity equal to twice the mean bulk velocity. All cases reveal a near isothermal condition over the aluminum packages due to the high thermal conductivity of the modules.

The low velocity example, with a mean velocity of 1.1 m/s, is shown in Fig. 4a. The temperature of the center module remains unchanged for $0 \leq \Lambda \leq 2$ since a linear velocity profile at the channel inlet insures that the centerline of the board always sees the mean bulk velocity. However, an 18 °C temperature difference occurs over the right hand module when Λ is varied between 0 and 2. Temperature changes of this order are very important when designing systems of circuit boards which might have a maximum to minimum operating temperature window of 50 °C. It can also be seen in Fig. 4a that the predicted module temperatures obtained using META are in excellent agreement with the empirical data.

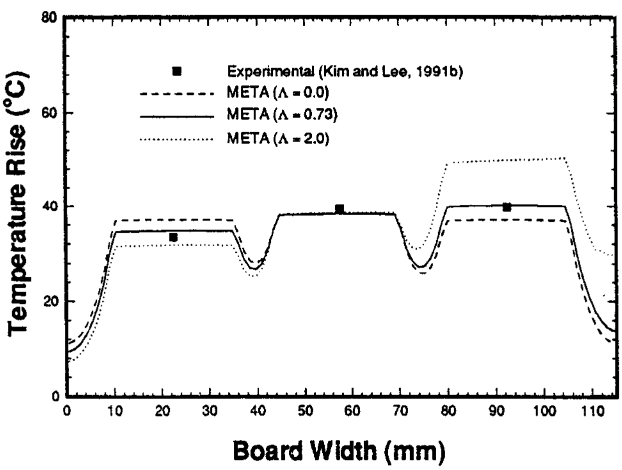
Figures 4b and 4c show similar trends for the cases where the mean velocity was 2.95 and 5.5 m/s, respectively. A



a) $\overline{u} = 1.1\ m/s$



b) $\overline{u} = 2.95\ m/s$



c) $\overline{u} = 5.5\ m/s$

Figure 4: Temperature Variations for Modules in the Fourth Row With Various Inlet Velocity Profiles

temperature difference of 14 °C is obtained over the right hand module as Λ is varied between 0 and 2 when the mean velocity is 2.95 m/s while a 13 °C change occurs over the same element when the mean velocity is 5.5 m/s.

In the examples shown in Fig. 4, it can be concluded that all the modules, including the last row, see the effects of non-uniformities at the channel inlet. Abnormalities in the flow introduced at the inlet appear to propagate along the length of the channel, hence the excellent agreement between the simulated and the experimental data in the last row of the board.

CONCLUSIONS

A linear velocity distribution, across the principal flow direction, can be used with good success to simulate the effects of swirling fan induced flow. Further studies on flow distributions will allow a more specific form of the velocity distribution to be used, such as a power law representation or a quadratic representation which might provide better agreement when predicting wall temperatures with rotationally dominated inlet velocities.

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