THERMAL ANALYSIS IN A CHANNEL BETWEEN VERTICAL ELECTRONIC CIRCUIT BOARDS COOLED BY FORCED CONVECTION

XU JIAYU, NOBUYOSHI NAKAGAWA
AND KUNIO KATO

*Department of Biological & Chemical Engineering, Gunma University,
Kiryu, 376*

Key Words: Forced Convection, Heat Transfer, Electronic Circuit Board, Finite-Difference Analysis, Temperature Distribution

A heat transfer analysis of a convective air cooled electronic circuit board (ECB) in a vertical electronic circuit boards channel was carried out.

To validate the model and the algorithms used in the computations, experimental tests were performed on dummy ECBs. Temperature distribution on an axial ECB and axial temperature distribution of air were measured with thermocouples. The temperature measurements were in good agreement with the computed value.

As a working example, the temperature distributions of air flow and the vertical circuit boards with evenly arranged electronic components cooled by forced convective heat transfer were simulated, and typical results were reported and discussed. It is clear that, such a computer program for the thermal analysis of electronic circuit boards is effective and enables the user to choose any arbitrary arrangement of components on the ECB as well as different values of all design parameters. The program can serve as a powerful tool either for checking the performance of an existing ECB or for sensitivity analysis in designing the packaging of new electronic circuit boards.

Introduction

The remarkable developments in miniaturization of electronic circuits have resulted in a steady increase of heat dissipation rates in electronic circuits and modules, while allowable peak temperature levels remained essentially unchanged. Hence the role of heat transfer analysis has become increasingly important in the design of electronic packaging for reliable operation, particularly in view of the large variety of geometries encountered in cooling of electronic equipment.

A good thermal design has to assure that for a given electronic package including a certain number of electronic circuit boards (ECBs) of different shapes and sizes, usually the boards are vertically arranged in a shelf, and a given power dissipated from each ECB, the temperature of any component never exceeds its permitted limit. In certain cases, structural and operational constraints are imposed on the system, such as weight limitations, limited flow rates of cooling air and high inlet air temperature.

In various electronic instruments such as computers and telephone exchange units, natural convective cooling is employed and there have been many experimental or theoretical studies of natural convective heat transfer between heated vertical

parallel plates (Aung, 1972, Churchill, 1976, A. Bar-Cohen and W. M. Rohsenow, 1984). The heat transfer in a channel between vertical electronic circuit boards by natural air convection was investigated (Kato *et al.*, 1991). In some cases, however, forced convective cooling is used, a finite-difference analysis of air forced convective cooling printed circuit boards in an enclosed or a horizontal parallel plate channel was carried out (U. Navon *et al.*, 1994).

The electronic system analyzed in this paper is a system which consists of several vertical parallel ECBs. The heat dissipated from the components is removed by forced air.

In view of obtaining an optimal thermal design required in some cases, and a conventional thermal design based on overall average values of temperature differences and heat transfer coefficients is inadequate, rather, the complete temperature field of the ECB must be determined by numerical analysis. Obviously, a convenient way to perform such an analysis is by resorting to available software. For this as well as other reasons, it was decided to carry out a numerical thermal analysis of the ECBs and, in addition, performed a set of experiments in order to validate the theoretical model by comparison with measured data.

1. The Physical Situation

The physical situation chosen for analysis in the

* Received on March 7, 1996. Correspondence concerning this article should be addressed to K. Kato.

present study is shown in **Fig. 1(A)**. It is a two-dimensional idealization of a typical electronic circuit board array. **Fig. 1(B)** shows the geometrical details of the board. The heat generation within the component is assumed to be uniform. The thickness of the boards on which the components are mounted is assumed to be small and the section temperature profile of it can be assumed uniform. The temperature field of air can be assumed to be uniform across the channel.

An accurate determination of the three-dimensional temperature distribution within each component and its substrate is quite complicated because it depends on a large number of factors, e.g., the exact structure of each component, the type of mounting and the type of laminated ECB employed. These difficulties may be overcome, with some loss of accuracy, by assuming that each component acts as a uniform heat source. The temperature of the adjacent substrate is equal to the component temperature while the regions surrounding the component act essentially as a fin around the thermal source. In other words, the thermal resistance of the ECB in a direction to its surface is normally much smaller than the surface thermal resistance to the air flow. This is a good approximation for any electronic component equipped with a large number of copper lead wires that reduce substantially the thermal resistance of the ECB.

2. Convective Heat Transfer to Air

The air flows in the channels formed by vertical electronic circuit boards are shown schematically in **Fig. 1(A)** and one of the electronic boards is shown in **Fig. 1(B)**.

In the absence of entry length sections, the flow is simultaneously developing thermally and hydrodynamically. The values of Reynolds number are varied from 1500 to 7000 for both sides of the ECBs. The flow in the component-side channel is characterized by three-dimensional effects caused by separation downstream of the components and bypass around them. Hence the accurate determination of local heat transfer either at the component surface or at the ECB section exposed to the flow is very complicated. An efficient thermal analysis package intended for use by a designer of the cooling system must, therefore, include simplifying assumptions concerning the flow and heat transfer regimes. These assumptions are as follows:

- The parallel plate channel flow on both sides of the ECB is assumed to be simultaneously developing.
- On the wall, which contains components, local Nu may be calculated from the empirical equation (1), we proposed before (N. Nakagawa *et al.*,

1996). At the rear surface of the ECBs, local Nu' may be calculated from the empirical equation (2), we proposed before (N. Nakagawa *et al.*, 1996).

$$Nu = 0.00043 Re^{1.55} D_e^{0.26} Pr^{1/3} / Y + 0.025 Re^{1.08} D_e^{-0.59} Pr^{1/3} \quad (1)$$

$$Nu' = 0.017 (Re \cdot Pr)^{1.02} (D_e / Y)^{0.39} \quad (2)$$

where $Nu = hD_e/\lambda$, is the local Nusselt Number. h is the local heat transfer coefficient. D_e is the total hydraulic diameter,

$$D_e = a \times D_{e1} + (1 - a) \times D_{e2} \quad (3)$$

here a is the area ratio of components to total plate,

$$D_{e1} = 4S' / (2D + 2W + 2xd) \quad (4)$$

$$D_{e2} = 4S / (2D + 2W) \quad (5)$$

S is the section area of a channel without components, D is the distance between vertical plates, W is the width of the plate, x is the number of components of a row, d is the height of the component, S' is the section area of a channel with components. Y is the distance from the bottom of the boards.

c) Natural convection and radiation effects are neglected.

d) No heat is lost through the edges of the ECBs.

3. Finite-Difference Formulation

The steady state energy equation for an ECB of thickness δ , with prescribed discrete heat sources q [$W m^{-2}$] distributed at the surface, is

$$\frac{\partial^2 T}{\partial X^2} + \frac{\partial^2 T}{\partial Y^2} + \frac{q - h(T - T_g) - h'(T - T_g)}{\delta \lambda} = 0 \quad (6)$$

where λ is the effective conductivity of the ECB in the X - or Y -direction. This equation and the boundary conditions can be transformed into finite-difference forms in terms of finite-difference energy balances for an area element of the ECB and the adjoining air flows. The grid in the X - Y plane of the ECB is set up in such a way that each component includes an integer number of area elements. This procedure may require stretching or shrinking of some components, thus introducing a certain error into the model. However, if the grid is reasonably fine, the error will not be too large. Two finite-difference energy balances are obtained for each nodal element (i, j). The equation for the ECB is

$$\begin{aligned}
& \beta(T_{i,j} - T_{i-1,j}) + \frac{1}{\beta}(T_{i,j} - T_{i,j-1}) \\
& + \beta(T_{i,j} - T_{i+1,j}) + \frac{1}{\beta}(T_{i,j} - T_{i,j+1}) \\
& = -\frac{1}{\lambda\delta} [h_{i,j}(T_{i,j} - T_{g,i,j}) + h'_{i,j}(T_{i,j} - T_{g'i,j}) \\
& - q](\Delta X \Delta Y)
\end{aligned} \quad (7)$$

here β is the ratio $\Delta Y/\Delta X$.

For the air flow in y-direction on either side of the ECB, the energy balance is

$$\begin{aligned}
\dot{m}c_p(T_{g,i,j} - T_{g'i,j-1}) &= (h_{i,j} + h'_{i,j})A_{i,j} \\
[T_{i,j} - \frac{1}{2}(T_{g,i,j} + T_{g'i,j-1})]
\end{aligned} \quad (8)$$

where $\dot{m}$ is the mass flow rate and T_g is the air temperature in the channel. $h_{i,j}$, the heat transfer coefficient, is computed by equation (1), $h'_{i,j}$, is computed by equation (2), $A_{i,j}$ is the nodal heat transfer area. The inlet air temperature in all channels is known of course. For a grid consisting of $m \times n$ nodes, a system of $2mn$ linear algebraic equations for the unknown nodal temperatures is obtained.

4. Numerical Results

The computations were performed for a 25.0 cm by 20.0 cm or 40.0 cm ECB which held 81 or 162 rectangular components measuring $6 \times 21 \times 4$ mm³ each. The schematic layout of the components is shown in Fig. 1(B) (C) (D). The widths of the channels are 1.9 cm or 1.5 cm. The substrate thickness, δ , was 4 mm and copper laminates 0.14 mm thick yielded an effective conductivity, λ , of 2.4 W (m K)^{-1} . A 126×101 node network was used in the computation.

The discrete temperature field of the ECB and the air flow were obtained for the values of the parameter listed above. The results of one run are represented graphically in Fig. 2(A) (B) (C). This run was made for a total dissipation rate of 700 W m^{-2} , air inlet temperature of 20.0°C and Re value of 5000. The temperature rise in the flow direction is shown in Fig. 2(A) for the ECB and in Fig. 2(B) for the air flow in the channels, respectively. Fig. 2(C) shows the temperature distribution of flow air and ECB at different positions of $X=0$, $X=12.5$ cm and $X=25.0$ cm, respectively.

5. Experimental Examination

A schematic diagram of the experimental

Fig. 1 Arrangement of electronic boards (A) and details of boards (B), (C), (D)

apparatus is shown in Fig. 3. In order to assure the accuracy and reproducibility of results, the dissipation rates of the components were simulated by 16-pin network-resistance devices on the writing plate made of glass-epoxy, the dissipation rates, air channel geometry and all air flow parameters were the same as those used in the computation and described in section 4. Properties of the plates are shown in Table 1.

Five of these electronic boards were vertically placed at a certain distance as shown in Fig. 3. A distributor plate was attached at the bottom of the shelf. The shelf was walled by acrylic resin boards to prevent flow disturbance by ambient gas. The ECBs were heated by a certain electric power level supplied to the circuit. The validity of the theoretical model was tested by a series of experiments, that is, the axial temperature distribution measured at the surface of the ECB and air flow in the channel were compared with the theoretical prediction.

Temperatures of the writing plate were measured by copper-constantan thermocouples placed at the points shown in Fig. 1(B) (C) (D). Axial temperatures of air flow in the channel and the ambient temperature were measured with thermocouples. The ambient temperature was measured at the bottom of the shelf.

Electric powers of 700 W m^{-2} and 350 W m^{-2} were supplied to the circuits, and the boards were heated. After the gas temperature in the channel became steady, temperature measurements were

Table 1 Properties of writing plates

Material	Thermal conductivity	Specific heat	Size
glass-epoxy	0.59 [W m ⁻¹ K ⁻¹]	0.80 [J g ⁻¹ K ⁻¹]	0.2 m × 0.25 m × 0.004 m 0.4 m × 0.25 m × 0.004 m

Fig. 2 Computed temperature distribution. Temperature of ECB (A), temperature of flowing air (B) and air flow at $X=0$, $X=12.5$ cm and $X=25.0$ cm

Fig. 3 Experimental apparatus

made. The differences between the measured temperature and the constant air inlet temperature were recorded. The results are shown in **Fig. 4(A) (B) (C)**. It can be seen that the results of the present model are in good agreement with the experiments.

6. Simulation Results

As a working example, the calculations of temperature distributions of air flow and the vertical circuit boards with evenly arranged electronic components cooled by forced convective heat transfer as shown in **Fig. 1(A) (B)** were performed. Effects of the power input, q , the distance between the boards, D , and the Reynolds number, Re , on temperature distributions of ECB and air flow in the channel were examined.

Figure 5 shows the effect of power supplies, q , on temperature distributions with height Y . It was found that the temperature of the plate and air increased with q increased. It was observed that the temperature difference between the plate and the component around one component was different from neighbouring ones and increased with increasing q . This may be due to the error introduced into the model because of the stretching or shrinking of some components in the computing process. We also noticed that the air temperature was slightly higher around components.

Figure 6 shows the effect of the distance between boards, D , on temperature distributions with height Y . It is found that the temperature of the ECB increased greatly with increasing D and the tem-

Fig. 4 Comparison of temperature between calculated values and experimental values for $q=700 \text{ W m}^{-2}$, $L=20.0 \text{ cm}$ (A) $q=700 \text{ W m}^{-2}$, $L=40.0 \text{ cm}$ (B) and $q=350 \text{ W m}^{-2}$, $L=40.0 \text{ cm}$ (C)

perature of air flow increased slightly with increasing D .

Figure 7 shows the effect of the Reynolds number, Re , on temperature distributions with height Y . It was found that the temperature increased with Re decreased.

Fig. 5 Effect of power supplies, q , on temperature distributions with height Y

Fig. 6 Effect of distance between boards, D , on temperature distributions with height Y

Fig. 7 Effect of Reynolds number, Re , on temperature distributions with height Y

From the simulation results mentioned above, we know that the program presented here can be used to design electronic packages optimally.

Conclusion

(1) The model proposed in the present paper offers a good tool for the thermal analysis of an ECB. It can be useful either in the design of electronic packaging or for evaluating the thermal performance of existing systems.

(2) The empirical equations (1) and (2) we proposed before, are suitable to evaluate the local Nusselt Number in this finite-difference analysis.

Nomenclature

a	= area ratio of components to total plate	[—]
$A_{i,j}$	= nodal heat transfer area	[cm ²]
C_p	= specific heat capacity of air at constant pressure	[J kg ⁻¹ K ⁻¹]
d	= height of component	[cm]
D	= distance between vertical boards	[cm]
D_e	= total hydraulic diameter of channel	[cm]
D_{e1}	= hydraulic diameter of channel with components	[cm]
D_{e2}	= hydraulic diameter of channel without components	[cm]
$h_{i,j}$	= local heat transfer coefficient of channel with components	[W m ⁻² K ⁻¹]
$h'_{i,j}$	= local heat transfer coefficient of channel without components	[W m ⁻² K ⁻¹]
L	= height of board	[cm]
$\dot{m}$	= mass flow rate of air	[kg s ⁻¹]
m	= number of nodes in X direction	[—]
n	= number of nodes in Y direction	[—]
Nu	= local Nusselt number of surface with components	[—]
Nu'	= local Nusselt number of surface without components	[—]

Pr	= Prandtl number of air	[—]
q	= discrete heat source (rated power dissipation of a component section)	[W m ⁻²]
Re	= Reynolds number ($Re = UD_e/\nu$)	[—]
S	= section area of a channel without components	[cm ²]
S'	= section area of a channel with components	[cm ²]
T	= temperature of ECB	[K]
T_o	= inlet temperature of air	[K]
T_g	= temperature of air	[K]
U	= gas velocity in channel	[m s ⁻¹]
W	= width of the board	[cm]
x	= number of components of a row	[—]
X	= horizontal coordinate	[cm]
Y	= vertical coordinate (distance from the bottom of the board)	[cm]
β	= ratio of $\Delta Y/\Delta X$	[—]
δ	= substrate thickness	[cm]
λ	= thermal conductivity of plate	[W m ⁻¹ K ⁻¹]
ν	= kinetic viscosity of air	[m ² s ⁻¹]

Literature Cited

- Aung, W., L. S. Fletcher and V. Sernas; "Developing Laminar Free Convection Between Vertical Flat Plates with Asymmetric Heating," *Int. J. Heat and Mass Transfer*, **15**, 2293-2299 (1972)
- Bar-Cohen A. and W. M. Rohsenow; "Thermally Optimum Spacing of Vertical, Natural Convection Cooled, Parallel Plates," *J. Heat Transfer*, **106**, 116-123 (1984)
- Churchill, S. W.; "A Comprehensive Correlating Equation for Forced Convection from Flat Plates," *AIChE Journal*, **22**, 264-268 (1976)
- Kato, K., T. Takarada, H. Miyazaki, H. Sato and N. Nakagawa; "Heat Transfer in a Channel Between Vertical Electronic Circuit Boards Cooled by Natural Air Convection," *J. Chem. Eng. Japan*, **24**, 568-574 (1991)
- Nakagawa, N., Xu Jiayu, K. Shibui and K. Kato; "Forced Convective Heat Transfer from a Channel Between Vertical Electronic Circuit Boards," *J. Chem. Eng. Japan*, **29**, 959-965 (1996).
- Navon U., S. Kreitenberger and C. Gutfinger; "Thermal Analysis of a Printed Circuit Board Cooled by Forced Convection," *Heat Transfer*, **2**, 415-419 (1994)