COBEM 99
XV Congresso Brasileiro de Engenharia Mecânica
15th Brazilian Congress of Mechanical Engineering
22 - 26 de Novembro de 1999 / November 22 - 26, 1999 Águas de Lindóia, São Paulo,

THE INFLUENCE OF THE DISTANCE BETWEEN PLATES IN AN ARRAY OF VERTICAL PRINTED CIRCUIT BOARDS COOLED BY NATURAL CONVECTION: NUMERICAL AND EXPERIMENTAL ANALYSIS

Ana Cristina Avelar
Universidade Estadual de Campinas, Departamento de Energia
Cx. P. 6122 – 13083-970 – Campinas, SP, Brasil
Marcelo Moreira Ganzarolli
Universidade Estadual de Campinas, Departamento de Energia
Cx. P. 6122 – 13083-970 – Campinas, SP, Brasil

Abstract. *Arrays of vertical channels formed by parallel plates are frequently encountered in natural convection cooling of electronic equipment. In such applications, for an allowable maximum temperature rise, the heat dissipated from a channel decreases with decreasing spacing, but the total number of plates increases. Hence, there is an optimum spacing between plates that maximizes the heat dissipation per unit volume. In the present study, numerical and experimental investigations of an optimum spacing were carried out in an array of fiber glass plates with resistors discretely distributed on their surfaces. Uniform heating tests with variation of power and distances among plates were performed. In the numerical analysis, energy balances for the components were formulated and the heat diffusion equation in two-dimensional Cartesian coordinates system and steady state conditions was numerically solved. The concept of "buoyancy induced forced convection" was applied to the resistors, where forced convection heat transfer coefficients were employed. Good agreement between numerical and experimental results was observed.*

***Key Words**: Natural convection, Optimum spacing, Printed circuit boards, Vertical channels*

1. INTRODUTION

Because of its simplicity, low cost and realiability natural convective cooling is an attractive heat transfer mechanism. In particular, many studies have been done on natural convection from vertical parallel plates due to its conection with the cooling of electronic equipament.

Levy *et al*. (1975) determined experimentally an optimum spacing between two plates, which minimizes the plate temperature for a given rate of convective heat transfer. The experiments were performed in air using parallel arrays of four and six vertical aluminium plates. By using composite relationships Bar-Cohen and Rohsenow (1984) developed correlations for natural convective heat transfer in heated channels covering the two asymptotes, one for a fully developed channel regime and other for a single plate boundary

layer regime. An optimum spacing correlation was obtained by maximizing the total heat dissipation through the channel. Both uniform heat flux and uniform surface temperature boundary conditions were considered. For these cases, symmetric and asymmetric heating were analysed. Anand *et al*. (1992) determined numerically the optimum plate spacing for single channels asymmetrically heated. The two-dimensional flow equations for air were solved employing the boundary layer approximations.

Marrone *et al*. (1997) analysed the problem of optimizing the plate separation of an open, parallel-plate vertical channel, symetrically heated, cooled by natural convection. An elliptic model and the finite-diference discretization technique were applied. They used a computational domain that comprises the actual physical domain between the channel, as well as two relatively large domains placed upstream of the channel entrance and dowstream of the channel exit. Bar-Cohen & Geisler (1997) rederived composite relations for analytical modeling of heat transfer from arrays of vertically-oriented populated Printed Circuit Boards, PCBs. They considered both isothermal and isoflux, single-sided, conditions. It was employed a pressure loss correlation which accounts for the effect of chip packages on the fluid dynamics for laminar fully developed flow in a narrow channel. The relations presented can be used in optimizing the spacing between PCBs to maximize the total heat transfer from a given base area or volume.

The majority of works about vertical channels cooled by free convection and optimum spacing deals with smooth surfaces, although in practical situations the channels are formed by plates with protusions. This study is aimed at analysing the heat transfer in an array of seven PCBs with resistors densely distributed on their surfaces. An optimum spacing is analysed theoretically and experimentally. This work is a continuation of the study by Avelar & Ganzarolli (1998), in which an optimum plate spacing was verified experimentally and analytically. The advantage of the numerical analysis is the possibility of simulating no-uniform heating of the plates. In the numerical analysis the program developed by Avelar (1997) was adapted for the plates analysed in the present study.

2. EXPERIMENTAL PROCEDURE

A schematic view of the plates used is shown in Fig. 1.

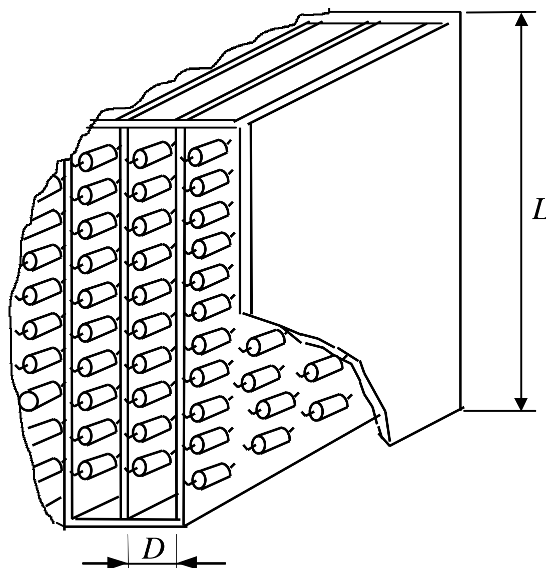

Figure 1 – Experimental apparatus.

An array of seven fiber glass plates was accommodated in an metallic structure that is used in telecommunications devices and that allows the variation of the distance among plates. Each plate was 365mm heigth (*L)* and 340mm width (*W*), with 1,5mm thickness and had 171 resistor distributed on its surface. Power was supplied to the plates by regulated D.C. sources. Experiments were performed varying the spacing between plates. The distances D = 10, 15, 20, 30, 35, 40 and 45mm were investigated. The plates were uniformly heated with power ranging from 20 to 45W. The total generation rate was set as the same for each plate. The lateral edges of the plates were isolated. Resistors temperature measurements were made using thermocouples 36 AWG type T. Care was taken to embbeding the thermocouples in the resistors surfaces. A very small role was drilled in the resistors surfaces, which was covered with a thin layer of thermal paste, and the thermocouples were fixed with epoxy adhesive.

3. NUMERICAL ANALYSIS

3.1 Energy balance in the component

The heat flow path from the component to the cooling air is shown in Fig. 2. Part of the heat dissipated is transferred by convection to the air flow. Another part is transferred by radiation to the surroundings. The rest of the heat is conducted to the printed circuit board, from which it is transferred by convection and radiation from its two faces.

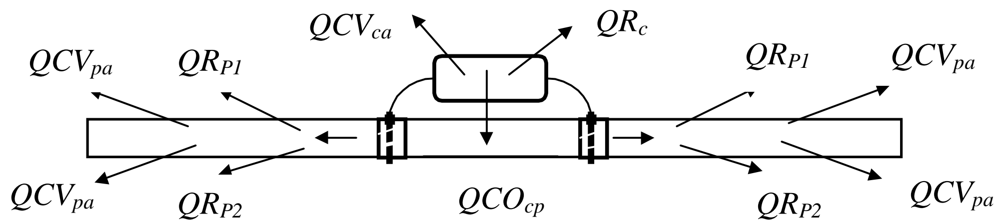

Figure 2 – Heat Flow paths from components and from printed circuit board.

- QCO_{cp} - heat transferred from the component to the printed circuit board through the lead wires;
- QCV_{ca} - heat transferred by convection from the component to the air;
- QCV_{pa} - heat transferred by convection from the plate to the air;
- QR_c - heat transferred by radiation from the component to the surfaces arround it;
- QR_{p_l} - heat transferred by radiation from the plate to the surfaces arround it.

Avelar (1997) analysed the influence of the radiative heat transfer in an similar problem, an array of PCBs with 25 resistor discretely distributed on 200x164mm, and verified that this mechanism is not so significant for small distance between plates, when the analysis presented is more appropriate. Thus neglecting the radiation effects, an energy balance for the component represented in Fig. 2 shows that the heat dissipated by the component is:

$$QC = QCV_{ca} + QCO_{cp} \qquad (1)$$

where the heat transferred by convection to the air (QCV_{ca}) is:

$$QCV_{ca} = h_c A_c (T_c - T_a) \qquad (2)$$

and the heat conducted to the printed circuit board (QCO_{cp}) is given by:

$$QCO_{cp} = \left(\frac{T_c - T_p}{R_{cp}} \right) \quad (3)$$

where R_{cp} represent the thermal resistance between the component and the printed circuit board. This parameter was determined based on experimental data.

3.2 Energy balance in the the printed circuit board

The steady state energy equation for the printed circuit board considered, when the heat transfer through the plate is neglected, is:

$$\varepsilon k_x \frac{\partial^2 T}{\partial x^2} + \varepsilon k_y \frac{\partial^2 T}{\partial y^2} + S = 0 \quad (4)$$

where, ε is the plate thickness and S is a source containing the heat conducted from the component to the plate and the heat transferred by convection to the air. The thermal conductivity was estimated taking into account the copper distribuition in the rear side of the plate, represented in Fig. 3.

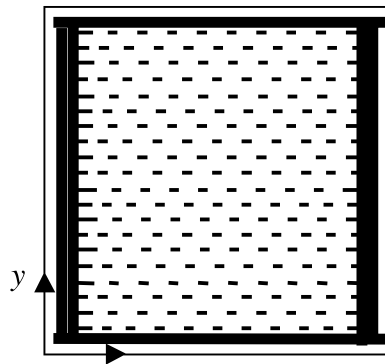

Figure 3 – Rear side of the printed circuit board.

In the y direction the conductivity was assumed equal to the fiber glass conductivity. In the x direction the conductivity was evaluated by an equivalent thermal circuit represented in Fig. 4.

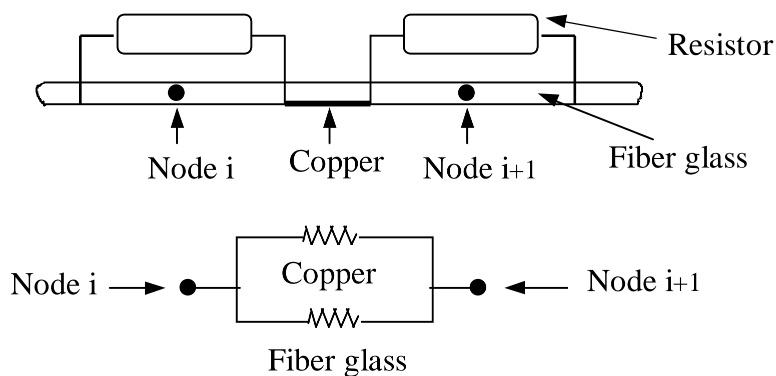

Figure 4 – Equivalent thermal circuit.

The conduction resistances of the copper and fiber glass are in parallel.

The printed circuit board was divided in eleven columns and twenty rows and Eq. (4) was integrated using the finite volume formulation.

3.3 Energy balance for the air flow in the channel

As represented in Fig. 5, the channel was divided in volume elements coincident with the plate divisions, but the grid points were taken at the volume elements interfaces. An energy balance for the channel air flow shows that the air entalphy variation is equal to the heat transferred by convection from the plate to the air, plus the heat transferred by convection from the conponent to the air.

$$\dot{m}_a c_p \left(T_{a_{i,j+1}} - T_{a_{i,j}}\right) = QCV_{ca} + 2QCV_{pa} \quad (5)$$

where $Ta_{i,j}$ represents the average mixture temperature in the cross section considered.

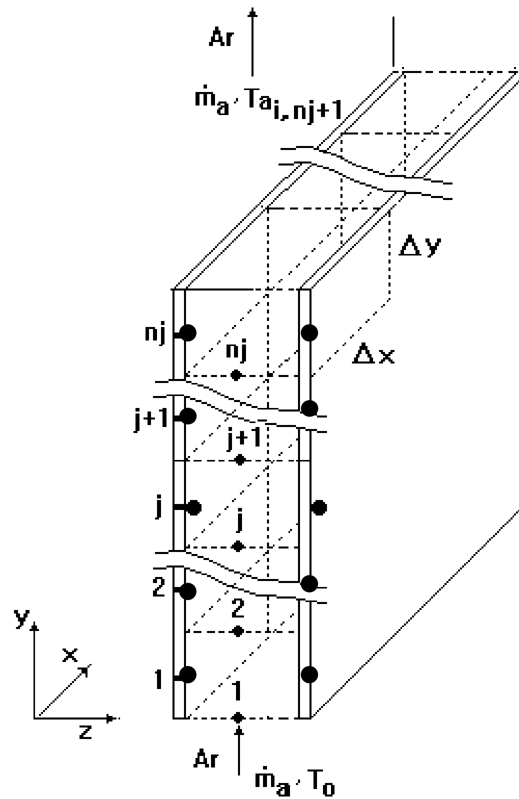

Figure 5 – Air channel flow.

3.4 Heat transfer coefficient in the components

Moffat & Ortega (1986) investigated the naturally-induced convective heat transfer from an array of cubical elements that were deployed in an in line array on one wall of a vertical channel. They concluded that when the channel walls are closely spaced and there is good fluid mixing at any streamwise location, the heat transfer is driven primarily by the globally induced channel flow, rather than by local buoyancy mechanisms.

Based on these conclusions, the heat transfer coefficients in the components surfaces was determined by using an empirical correlation for the Nusselt number proposed for forced convection situations by Witzman *et al.* (1990):

$$Nu_a = 0{,}465 Re_a^{0{,}60} \tag{6}$$

where Nu_a *e* Re_a are based on the characteristc length of the component, a, defined as the square root of the component top surface area.

The heat transfer coefficient is based on the element adiabatic temperature, which is defined as the temperature that an element achieves when the convective heat transfer from the element to the fluid goes to zero. In the absence of conduction and radiation from other elements, it is the temperature achieved when no-power is applied to the element and the rest of the system is activated. When the air temperature is uniform in a channel cross section, this temperature can be represented by the air average mixture temperature.

3.5 Heat transfer coefficient in the plate

The heat transfer coefficient in the plate was calculated by using a correlation to the Nusselt Number that was obtained from the study of Fujii *et al.* (1994). They investigated the natural convection heat transfer to the air from an array of vertical parallel plates, each of which uniformly heated on one side. According to Fujii *et al.* (1994), the Nusselt number in the plate is:

$$Nu = \frac{\phi}{6{,}93}\left[1 - exp\left(-5{,}72\phi^{-0{,}66}\right)\right] \tag{7}$$

$$\phi = \frac{Gr * Pr/H}{\left(Gr * Pr/L\right)^{1/2}} \tag{8}$$

where

$$Gr* = \frac{\beta g q'' d_{ef}{}^{4}}{k_{ar}\nu^{2}}, \quad \beta = \frac{1}{T_o}, \quad Nu = \frac{h d_{ef}}{k}$$

$$H = \frac{L}{d_{ef}}, \qquad Y = \frac{y}{d_{ef}}, \qquad q'' = \frac{Q}{2WL},$$

The distance d_{ef} is defined as the spacing between two plates minus the resistor height and Q is the energy dissipated in the PCB.

In this case the heat transfer coefficient was based on the air temperature in the entrance region.

4. RESULTS AND DISCUSSION

A comparison between experimental and numerical values of the resistors temperature at the channel exit are shown in the Figs. 6 and 7. The results are presented for Q = 30W and 40W in terms of d_{ef} and temperature excess (difference between the temperature of the ambient fluid and the resistor temperature).

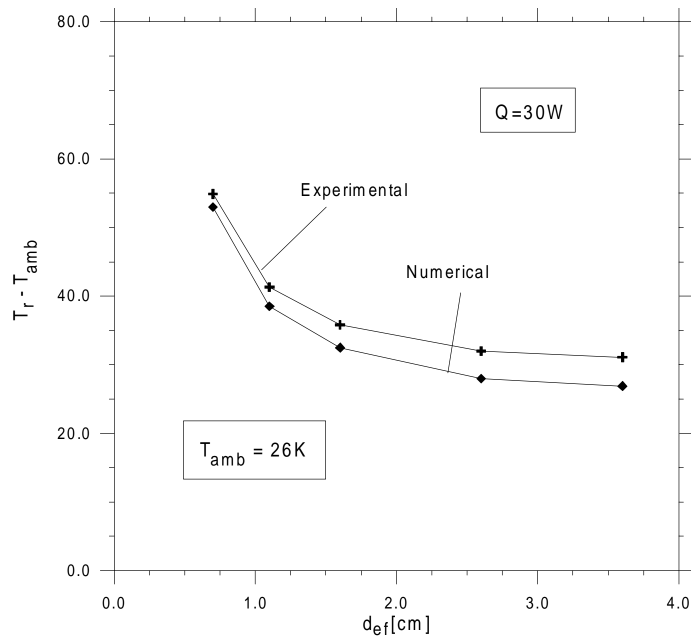

Figure 6 – Experimental and numerical results of resistors temperature excess for Q = 30W.

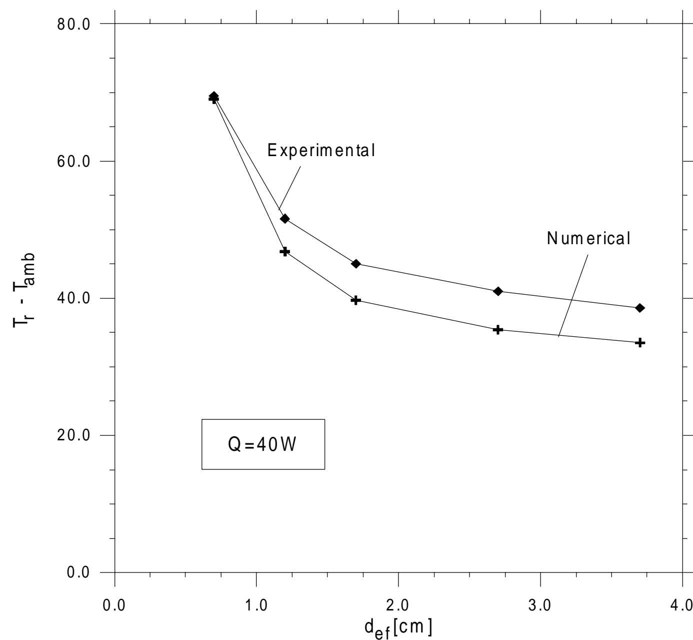

Figure 7 – Experimental and numerical results of resistors temperature excess for Q = 40W.

It is noted that the numerical analysis underpredicts the temperature distribution and it is more appropriate for small distance between plates, when there is a good fluid mixing into the channel. The most significative difference between numerical and measured values was verified for the largest aspect ratio and it is around about 10%.

Figure 8 shows the numerical simulations of the power as a function of the distance between plates power for the given temperature differences 40K and 50K and the corresponding two points of optimum spacing numerically found.

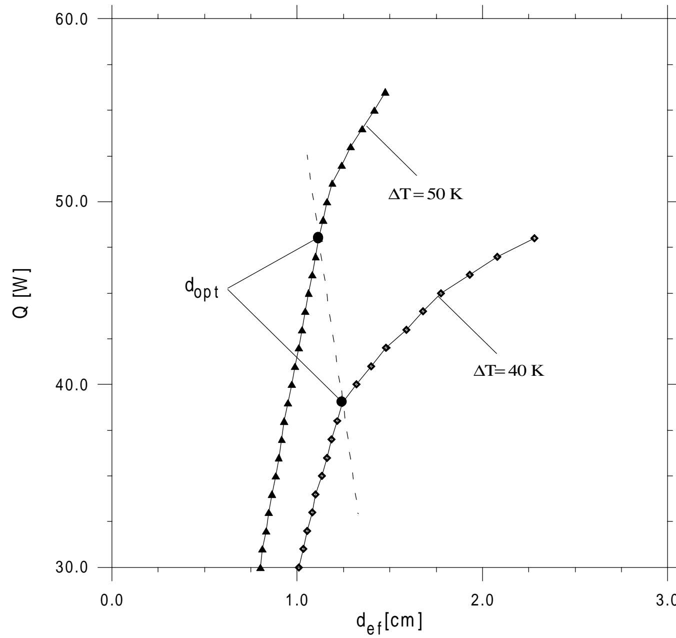

Figure 8 – Variation of distance between plates with the power: numerical results.

In order to construct Fig 8 the program was executed for several powers and distances between plates. For fixed values of temperature excess, 40K and 50K, it was identified the power and distance that yielded the specified values. Then each specified power was divided by the equivalent distance. It was verified initially that the result increases, and after a determined distance, d_{opt}, it decreases. It was numerically found d_{opt} equal 2,015cm and Q=48W for 50K and d_{opt} equal 2,145cm and Q=39W for 40K.

Figure 9 presents the optimum spacing experimentally found for the temperature differences of 40K and 50K.

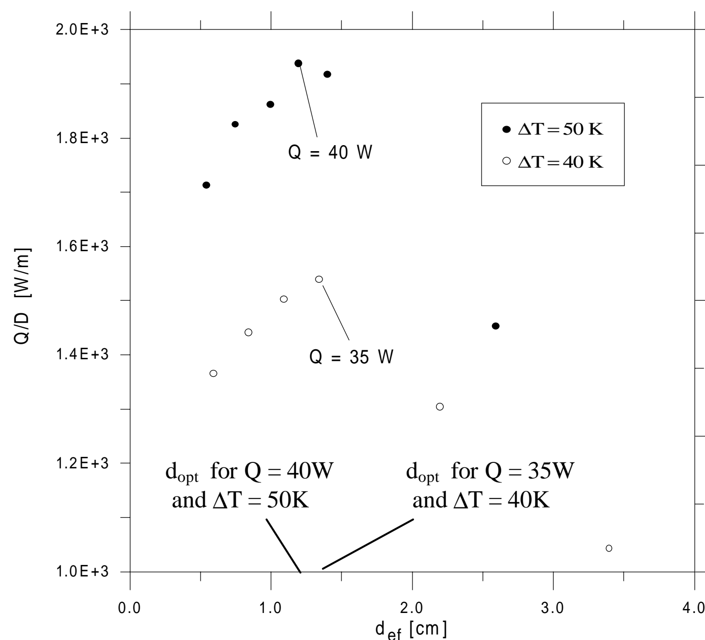

Figure 9 – Experimental results of optimum spacing.

The values presented in Fig. 9 were estimated from the experimental results of resistors temperature excess at the channel exit shown in Fig. 10.

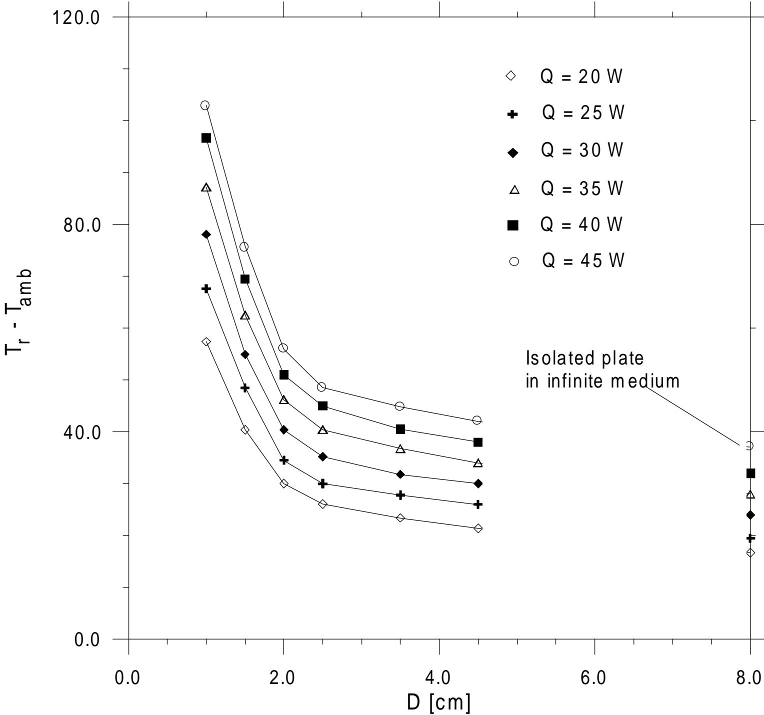

Figure 10 – Measured values of resistors temperature excess at the channel exit.

In Fig. 9 it was adopted a procedure similar to that used to make Fig. 8. In all the constant power curves presented in Fig. 10, for the fixed values of temperature excess, 40K and 50K, it was identified the spacing between plates yielding the specified temperature difference. Finally, each power was divided by the equivalent distance. Table 1 shows the numerical and experimental results of power and optimum spacing equivalent to the specified values of temperature excess.

Table 1 – A comparison between numerical and experimental results.

	Numerical results		Experimental results	
ΔT [K]	Q[W]	d_{opt} [cm]	Q [W]	d_{opt} [cm]
40	39	2,145	35	2,04
50	48	2,015	40	2,00

It can be noticed a good agreement between the numerical and experimental results of optimum spacing. Although the energy dissipated in the PCBs are overestimated, the difference between numerical and measured values is approximately 15%. This fact occurs because the numerical analysis underpredicts the resistor temperature, as it can be noted in Figs. 6 and 7, thus in order to yield satisfactory values of d_{opt}, for fixed temperature values, the power is overestimated.

5. CONCLUSIONS

Uniform heating tests with variation of power and distance between plates were carried out in an array of seven fiber glass plates with resistors densely distributed on their surfaces, and an optimum spacing between plates was investigated numerically and experimentally.

The numerical analysis proposed underestimates the resistors temperature and it is appropriated for low aspect ratio. The higher difference between numerical and experimental values is approximately 10%. For fixed values of resitors temperature the optimum spacing was satisfactorily predicted, but the power was overestimated in around 15%.

The great advantage of the numerical analysis is that it can be useful to conditions of no-uniform plate heating. Results based purely on correlations are restricted to uniform boundary conditions, isoflux and isothermal.

6. REFERENCES

Anand, N. K., Kim, S. H., Fletcher, L. S., 1996, The effect of plate spacing on free convection between heated parallel plates, Journal of Heat Transfer, vol. 118, pp. 56-64.

Avelar, A C., 1997, Análise teórica e experimental da transferência de calor em placas de circuito impresso formando canais verticais abertos, Dissertação (Mestrado), UNICAMP, Campinas.

Avelar, A. C., Ganzarolli, M. M., 1998, Influência do espaçamento numa série de placas de circuito impresso formando canais verticais, Proceedings of 7th Brazilian Congress of Engineering and Thermal Sciences, 03-06 de novembro de 1998, Rio de Janeiro, vol. 1, pp. 213-217.

Bar-Cohen, A., Rohsenow, W. M., 1984, Thermally optimum spacing of vertical, natural convection cooled, parallel plates, Numerical Heat Transfer, vol.106, pp. 116-123.

Geisler K. J., Bar-Cohen, A., 1997, Package-corrected composite relations for natural convection between asymmetrically-heated PCBs. In: Avances in Electronic Packaging, ASME -EEI .

Levy, E. K., Eichen, P. A, Cintani, W. R. and Shaw, R.R., 1975, Optimum plate spacings for laminar natural convection heat transfer from parallel vertical flat plates: experimental verification. Journal of Heat and Mass Transfer, n.3. pp. 475-476.

Marrone, B., Campo, A. and Manca, O., 1996, Optimum plate separation in vertical parallel-plate channels for natural convective flows: incorporation of large spaces at the channel extremes. International Journal of Heat and Mass Transfer, vol. 40, pp. 993-1000.

Moffat, R. J., Ortega, A.,1986, Buoyancy induced forced convection. Heat Transfer in Electronic Equipment – ASME HTD, v. 57, pp. 135-144.

Witzman, S. Newport, D., Nicoletta, T., 1990, An alternative methodology for computing the temperature rise of an electronic device inside a direct air-cooled electronic cabinet. In: Proceeding of Ninth International Heat Transfer Conference, Israel. v. 2, pp. 313-318.