A Novel Concept of Dummy Heat Sources for Heat Transfer Enhancement in a Vertical Channel

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Abstract: This paper reports the results of forced convection in a vertical rectangular channel with seven discrete heat sources and dummy (no heat input) heat sources mounted on a wall. The position of seven heat sources is fixed in an optimal arrangement. The heat transfer effects are investigated numerically and experimentally by arranging six and eight dummies in different configurations. The heat flux of 1500 W/m^2 and the air velocities of 0.6, 1, and 1.4 m/s are considered. Bakelite, FR4 and the copper clad boards are used as substrate board materials. The Reynolds numbers and the Nusselt numbers are respectively: $5360 \leq Re \leq 12500$ and $1.5 \leq Nu \leq 3$. The governing equations are solved using the finite element method in conjugate heat transfer module of COMSOL Multiphysics®. The dummy heat sources have a significant influence on the flow and heat transfer for low Reynolds numbers. In general, cases which show the lowest temperature distributions on the modules are those with copper clad board and six dummy heat sources.

Keywords: Forced convection, discrete heat sources, optimal arrangement, conjugate heat transfer, thermal management of electronics

1. Introduction

The most typical electronic packaging involves forced air cooling of single chip components on a printed circuit board (PCB). The heat generated in the components can be directly dissipated from their exposed surface by convection or indirectly first through the board by conduction and then by convection to the air stream. Choi et al. [1] numerically investigated the coupled conduction and forced convection transport from substrate-mounted modules in a channel to identify the effects of the substrate conductivity. The authors found that recirculating cells as well as stream wise conduction through the substrate play an important role in predicting convective heat transfer from the printed circuit board (PCB) and modules and in determining the temperature distributions in the PCB, modules, and fluid.. Recent advances in computing speed, and improved packaging density in modern digital computers, can be attributed to convection heat transfer research resulting in novel thermal designs for handling high heat dissipation rates. Davalath and Bayazitoglu [2] studied forced convection cooling across rectangular blocks by considering conjugate heat transfer for two-dimensional, developing flow over an array of rectangular blocks, representing finite heat sources on parallel plates. Mc Entire and Webb [3] performed forced convection experiments on two dimensional discrete protruding and flush-mounted heat source array to measure convective heat transfer. The authors revealed that the local heat transfer measurements have significant variation in the wall temperature across the heater faces, which is strongly affected by the flow structure. Hajmohammadi et al. [4] determined the optimal configuration of heat sources in an array on a substrate cooled by laminar forced convection to minimize the 'hot spot' temperature of the plate or to maximize the rate of heat transfer. Study showed that the reduction in 'hot spot' temperature and enhancement in heat transfer with optimum adiabatic spacing. da Silva et al.[5] studied optimal distribution of discrete heat sources on a plate with laminar forced convection and presented two different approaches such as large number of small heat sources, and small number of large heat sources with finite length. The authors showed that the best possible design are the highly complex configuration characterized by large numbers of optimally placed heat sources but is not the one in which the wall is heated uniformly. Tye-Gingras et al. [6] numerically optimize the triggering time of several discrete heaters attached to a channel in forced convection. Study showed that the activation phase lag and position of each heater can significantly decrease the overall thermal resistance. Desrayaud et al. [7] investigated

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numerically natural convection air-cooling of a substrate-mounted protruding heat source in a stack of parallel boards. They demonstrated that stream wise conduction through the substrate is an important cooling mechanism and must be accounted for. Muftuoglu and Bilgen [8] determined optimum positions of discrete heaters by maximizing the conductance and studied heat transfer and volume flow rate with discrete heaters at their optimum positions. They found that the global conductance is as an increasing function of the Rayleigh number, the heater size and the number of heaters and best thermal performance is obtained by positioning the discrete heaters closer to the bottom and closer to each other at the beginning of fluid flow. Hadim [9] Forced Convection in a fully Porous and partially porous channel with localized heat sources and indicated that as the Darcy number decreases, a significant increase in heat transfer is obtained, especially at the leading edge of each heat source. Chen et al. [10] have numerically investigated for enhanced heat transfer from multiple discrete heated sources in a horizontal channel by metal-foam porous layer. They obtained significant cooling augmentation of heaters by using metal foam porous layers. Kim and Anand [11] have numerically studied laminar developing flow and heat transfer between a series of parallel plates with five surface mounted heat sources including the substrate conduction in the analysis. They found that thermal entry length decreased with an increase in substrate conductivity. Rau and Garimella [13] investigated direct cooling of electronic components using dielectric liquid HFE-7100 and obtained local heat transfer distributions at high spatial resolution under two-phase transport conditions in confined and submerged impingement from arrays of miniature jets.

In spite of the large variety of studies dealing with natural and forced convection air cooling, forced convection in liquid cooling, forced convection cooling of discrete heat sources using porous media available in published literature, investigations on comprehensive experimental and numerical laminar natural and forced convection air cooling of electronics packages have been rarely reported.

Hence an extensive experimental and numerical study has been performed with air cooling of discrete heat sources. The novel concept of dummy heat sources along with the heat generating discrete heat sources is demonstrated in the present study. The effect of substrate board thermal conductivity on heat dissipation has also been studied in detail to provide certain guidelines for electronic cooling.

2. Substrate board and Heat Sources

The substrate board materials used in the present study are bakelite, FR4 and Copper clad board. The heat sources are made of aluminum. The thermal conductivity of bakelite is 1.4 W/m K. Fig. 1 shows bakelite substrate board mounted with seven heat sources in an optimal arrangement. The similar procedure as in the cited references is adopted to define non-dimensional parameter λ to decide the particular configuration [14, 15].

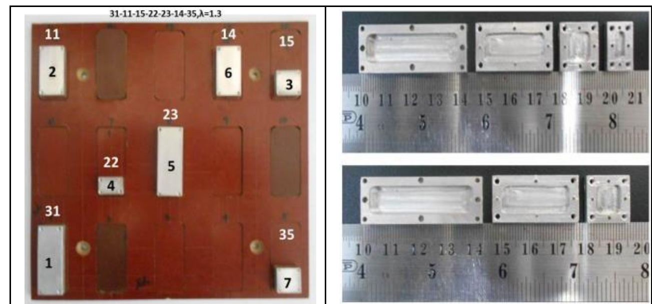

Figure 1. Substrate board and heat source details.

FR4 is a material composed of fiberglass cloth with an epoxy resin binder that is flame resistant. FR4 stands for flame retardant and denotes safety in case of fires. The thermal conductivity of FR4 is 0.3 W/m K. FR4 is used in the manufacture of printed circuit boards and are strongly anisotropic with the conductivity normal to the boards being much smaller than tangential to it. FR4 is a backbone upon which rigid printed circuit boards (PCB) are formed. A thin layer about 35 micron of copper foil is laminated to one or both sides of an FR4 glass epoxy panel to form copper clad board. A copper clad board is a layered composite consisting of copper foil (k = 390 W/m K) and a glass-reinforced polymer or FR4. The in-plane and through plane thermal conductivity of copper clad board is calculated using Eq. 1 and 2,

following reference [16,17]. The in-plane thermal conductivity of copper clad board is high whereas the through-plane thermal conductivity is comparatively very low. Hence the in-plane thermal conductivity is considered as an effective thermal conductivity and is calculated using Eq. 1.

$$k_p = \frac{\sum_{i=1}^{N}(k_i t_i)}{\sum_{i=1}^{N} t_i} \quad (1)$$

and $$k_s = \frac{\sum_{i=1}^{N} t_i}{\sum_{i=1}^{N}(\frac{k_i}{t_i})} \quad (2)$$

The effective thermal conductivity of the board can be tailored to specific need and it depends on the thickness and the number of layers of FR4 and copper foil. The thickness of copper foil usually is 35µm and the thickness of FR4 ranges between 254µm to 76 mm.

The copper clad board is single sided or two sided copper, usually a total of nine layers of copper and FR4 forms thickness of about 1.6 mm to 3.4 mm. The in-plane thermal conductivity is about 40.5 W/m k and the through plane thermal conductivity is about 0.28 W/m k for two sided copper layer. Hence the internal heat conducted to the board surface is due to in-plane conductivity and subsequent convection to the ambient air [18,19].

3. Experimental Procedure

Three configurations considered for experiments in the present study to validate the numerical results are 1) seven heat sources without dummy, 2) seven heat sources with six dummies and 3) seven heat sources with eight dummies, with three different substrate materials viz bakelite, FR4 and single sided copper clad board. After arranging the heat sources in a particular configuration on the substrate board, the board is fixed to the wall of the test section. Two K- type (or chromel- alumel) thermocouples from a single heat source are connected in parallel to a data logger channel (Make: Agilent) so as to record the average temperature of each source. The data logger is set to scan once every 5 minutes. Another thermocouple is used to measure the ambient temperature. Each heat source is made from aluminum block embedded with nichrome wire heater and connected to a separate regulated DC power supply.

Experimental setup and the instrumentation used are available in the Heat Transfer and Thermal Power Laboratory at IIT Madras. These have been described in earlier publications such as Hotta et al. [20, 21].

4. Numerical Model

The three-dimensional governing equations of continuity, momentum and energy equations for the fluid flow and the energy equation for the solid regime have been solved by using COMSOL Multiphysics 4.3b and a database of temperature versus configurations was generated. The present model assumes laminar flow and employs the conjugate heat transfer module. The computational domain is shown in Figure 2.

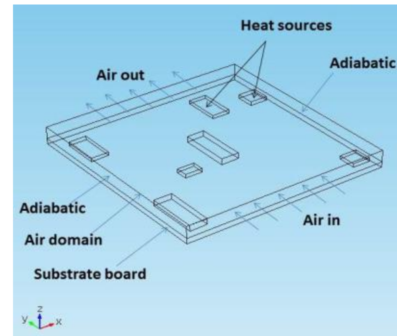

Figure 2. Computational domain

4.1 Governing Equations

Medium under consideration is air. The flow is considered to be incompressible and laminar, with constant properties except for density. Radiation heat transfer, viscous heat dissipation, compressibility effects and contact resistance between the heat source and the substrate are considered to be negligible. To account for conjugate heat transfer, the energy equation is solved for the solid domain. Based on the above assumptions, the governing equations for mass, momentum and energy for a three-dimensional flow in the fluid domain and the energy equation in the solid region are shown in Eq. 3 to 10. Similar procedure for numerical simulation is given in [12].

1. Heat transfer in solids,

$$\rho c_p u \nabla T = \nabla(k \nabla T) + Q \qquad (3)$$

2. Thermal insulation,

$$-n\,(-k \nabla T) = 0 \qquad (4)$$

3. Equations for fluids,

$$\rho(u\,.\nabla)u = \nabla\,.[\,pl + \mu(\nabla u + (\nabla u)^T - \frac{2}{3}\mu\,(\nabla\,.u)l] + F \qquad (5)$$

$$\nabla\,.(\rho\,u) = 0 \qquad (6)$$

$$\rho c_p u \nabla T = \nabla\,.(k\,\nabla\,T) + Q + Q_{vh} + W_p$$

$$Q = \frac{P_{total}}{V} \quad \frac{W}{m^3} \qquad (7)$$

4. Open boundary

$$[-pl + \mu\left(\nabla u + (\nabla\,u)^T - \frac{2}{3}\,\mu\,(\nabla\,.u)l\right]n = -f_0 n$$

$$T = T_0 \quad , \text{if} \quad n\,.\,u < 0 \qquad (8)$$

$$-\nabla T\,.n = 0 \quad if \quad n.u \geq 0 \qquad (9)$$

5. Periodic boundary condition

$$-n_{dst}\,.(k\,\nabla T)_{dst} = n_{src}\,.(k \nabla\,T)_{src} \qquad (10)$$

$$T_{dst} = T_{src}$$

The boundary conditions are as follows,
At wall: u= 0, No slip condition

Initial values: T = T0 $u = u_0 = u_{max}$

The thermos-physical properties, k, ρ, C_p, μ, γ are taken from material
The preprocessor controlled, free tetrahedral, extremely fine mesh is used with stationary study to obtain numerical solution. The present study focuses on obtaining minimum discrete heat sources temperature for maximum heat dissipation.

5. Numerical Results:

The present work investigates the results of seven heat sources without dummies, seven heat sources with six dummies and seven heat sources with eight dummies. Six different configurations in each case have been studied to decide the optimum configuration in each case. From the 162 cases the database is generated. From the simulation it is found that a typical configuration with six dummies gives the lowest temperature compared with The numerical results obtained from the simulation using bakelite, FR4 and copper clad board substrate are shown in Table 1 and the experimental results are shown in Table 2. It is observed that the lowest temperature is 37.7°C associated with the copper clad board in case of six dummy heat sources with heat flux of 1500 W/m^2 and at velocity 1.4 m/s. The highest value of temperature exists with FR4 substrate board material and is 72 °C. Figures 3 and 4 shows the velocity plot and temperature plot for bakelite for Re = 12500, q = 1500 W/m^2 and Pr = 0.7. Figures 5 and 6 shows the temperature plot and contour plot using copper clad board as the substrate board. The values of temperature are much lower compared with bakelite board with no dummy. Whereas the temperatures in case of FR$ are more as compared to bakelite and copper clad board.

Cases	Temp °C		
	Bakelite	FR4	CCB
7heaters u = 0.6 m/s	70.4	72	49.2
1 m/s	60.9	67.7	44
1.4 m/s	53.9	54.1	42.4
6dummy u = 0.6 m/s	68.7	69.8	48.2
1 m/s	56.1	60.1	43.2
1.4 m/s	52.7	53.4	37.3
8dummy u = 0.6 m/s	67.7	69.2	49
1 m/s	55.4	56.4	43.2
1.4 m/s	52.1	53	39.1

Table 1. Numerical results

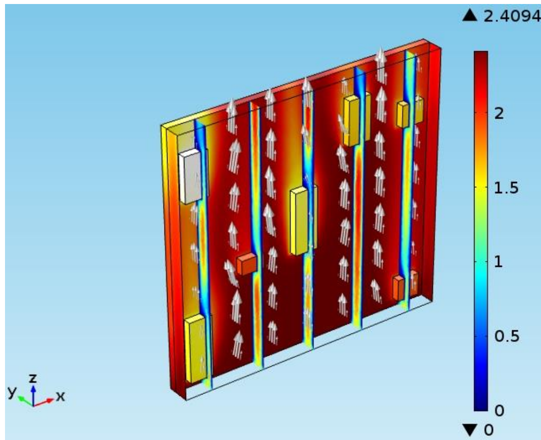

Figure 3. Velocity plot for bakelite, no dummy for q = 1500 W/m^2, Re = 12500, Pr = 0.7

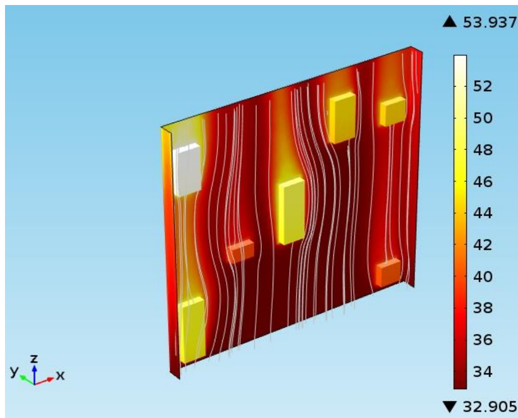

Figure 4. Temperature plot for bakelite, no dummy for q = 1500 W/m^2, Re = 12500, Pr = 0.7

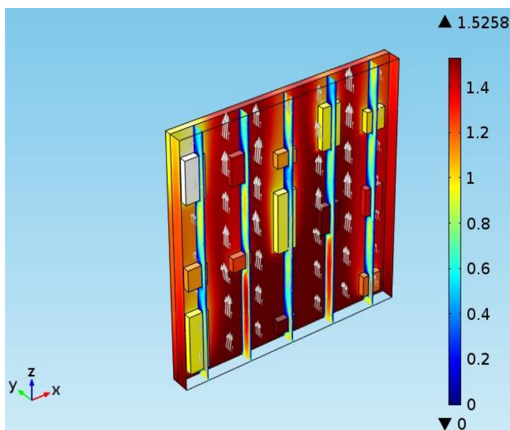

Figure 7. Velocity plot for FR4 for q = 1500 W/m^2, Re = 10700, Pr = 0.7

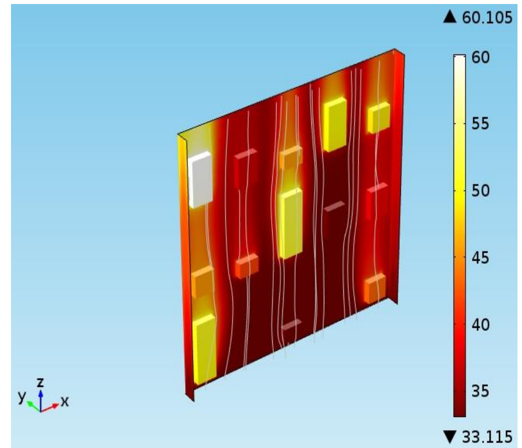

Figure 8. Temperature plot for FR4, Re = 10700,

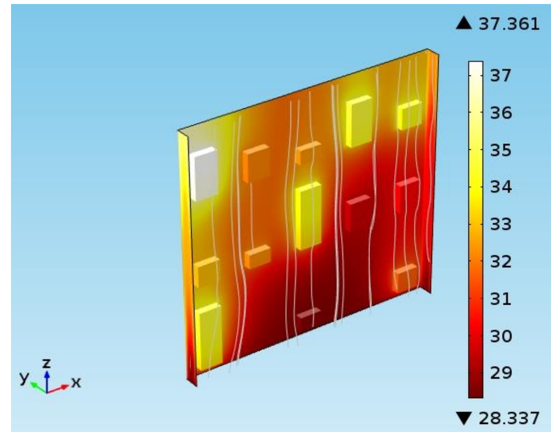

Figure 5. Temperature plot for copper clad board, six dummy for q = 1500 W/m^2, Re = 12500

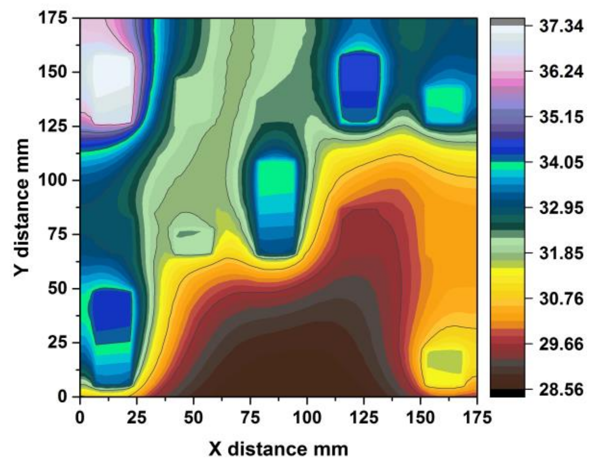

Figure 6. Temperature plot for copper clad board, six dummy for q = 1500 W/m^2, Re = 12500

Cases	Temp °C		
	Bakelite	FR4	CCB
7heaters u = 0.6 m/s	68.2	70.1	48.4
1 m/s	59.7	65.9	42.6
1.4 m/s	51.8	52.6	41.5
6dummy u = 0.6 m/s	67.8	70.1	47.1
1 m/s	54.7	57.9	41.7
1.4 m/s	50.9	51.4	36.4
8dummy u = 0.6 m/s	66.2	67.3	47.6
1 m/s	53.8	55.1	41.7
1.4 m/s	53.2	51.6	38.1

Table 2. Experimental results

6. Results and Discussion

From Figures 3-8 shows temperature, velocity and contour plots for bakelite, FR4 and copper clad board. From these plots and Table 1 showing numerical results it is found that the temperature in case of no dummy are higher compared to six and eight dummy cases. It indicates that using dummy heat sources there is an enhancement in heat dissipation as the temperatures in case of six and eight dummy heat sources are lower compared to no dummy case. It is also found that substrate conductivity of the substrate board has a significant effect in heat dissipation. The heat is first conducted from board and then convected by air. The dummy heat sources are acting as extended surfaces and contributing in heat dissipation. It is observed that a typical arrangement of six dummy heat sources shown in Figure 5 is a better choice among all six configurations considered in the present study. From the comparison of data given in Table 1 and 2 it is seen that numerical and experimental results are in good agreement. Some experimental values in Table 2 are higher than that of numerical values shown in Table 1, it is because of the thermal contact resistances between substrate board and the heat source, which is ignored in the numerical model.

7. Conclusions

A novel concept of dummy heat sources for heat transfer enhancement in a vertical channel has been studied numerically and experimentally in forced convection. From the present study it is concluded that a typical configuration of six dunny heat sources shows the significant enhancement in heat transfer compared with the cases of no dummy and eight dummy. From the experimental and numerical investigations it is also concluded that the single sided copper clad board having more effective thermal conductivity compared to bakelite and FR4 is the contributing to improve heat dissipation from the discrete heat sources.

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