CONJUGATE HEAT TRANSFER FROM A RAISED ISOTHERMAL HEAT SOURCE ATTACHED TO A VERTICAL BOARD

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ABSTRACT

Natural convection cooling of large raised bodies attached to conductive substrates are studied analytically and compared with experimental results to ascertain the significance of conjugate heat transfer in applications related to microelectronic cooling.

It is shown that good agreement between experimental data and simulated results, obtained using a model based on the boundary layer equations, is possible when heat transfer from a raised isothermal body is accounted for by reducing the thermal resistance between the heat source and the cooling fluid in proportion to the increase in wetted surface area. The importance of radiative heat transfer between the heated object and the surroundings is shown to be very important, especially in natural convection applications where the dissipation of heat due to radiation and convection are of similar magnitude.

The importance of using high conductivity, copper lands as a means of lowering heat source temperatures is demonstrated for cube-on-board applications. The most significant effect of the land is attributed to a very small region closest to the heat source, with increases in the size of the land contributing very little to the overall cooling of the heat source.

NOMENCLATURE

A_c	-	base area of cube, m^2
A_l	-	plan area of land, m^2
A_f	-	side wall area factor $\equiv A_{sw}/A_c$
A_{sw}	-	side wall area of cube, m^2
Bi	-	Biot number $\equiv ht/k_e$
h	-	heat transfer coefficient, W/m^2K
h_c	-	contact conductance, W/m^2K
k	-	thermal conductivity, W/mK
M	-	number of board layers
L	-	board length, m
q	-	heat flux, W/m^2
Q	-	heat flow rate, W
R	-	thermal resistance, $°C/W$
t	-	board thickness, m
T_c	-	cube temperature, $°C$
T_w	-	wall temperature, $°C$
$\overline{T}_b$	-	mean board temperature, $°C$
T_∞	-	free stream temperature, $°C$
u, v, w	-	boundary layer velocity in the x, y and z-direction, respectively, m/s
W	-	board width, m
x, y, z	-	Cartesian coordinates

Greek Symbols

α	-	thermal diffusivity $\equiv k_f/(\rho \cdot c_p)$, m^2/s
ϵ	-	emissivity
ν	-	kinematic viscosity, m^2/s
σ	-	Stefan-Boltzmann constant $\equiv 5.67 \times 10^{-8}$, $W/(m^2K^4)$

Subscripts

c	-	cube
cl	-	copper land
$cond$	-	conduction
$conv$	-	convection
ct	-	contact
e	-	effective
f	-	fluid
fe	-	fiberglass-epoxy
in	-	interface
j	-	Joulean
p	-	parallel
r	-	radiation
rsw	-	radiation side wall
s	-	series
sk	-	sink
sw	-	side wall
w	-	wall
∞	-	free stream
1,2	-	front and back surfaces, respectively

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INTRODUCTION

Electronic modules attached to multilayered printed circuit boards offer a significant challenge to thermal designers because of the complex heat transfer paths established between the heated modules, the board and the cooling fluid. Depending on the flow conditions, the thickness and the thermal conductivity of the individual layers within the board, a significant percentage of heat liberated from the module may pass through the board and then to the cooling fluid as opposed to being dissipated directly from the exposed surfaces of the modules. This study will examine an electronic module/circuit board prototype consisting of an aluminum cube attached to a flat board suspended in a vertical orientation. The significance of conjugate heat transfer, including conduction, natural convection and radiation, from the aluminum cube to the plate and the surrounding air medium will be examined by monitoring temperatures at selected locations in the cube and on the surfaces of the board.

The dimensions and thermophysical properties of the aluminum cube are maintained in each experiment while the internal thermal resistance of the board is altered by varying the board thickness and conductivity. As the relationship between the internal thermal resistance of the board and the resistance of the cooling fluid is altered, the temperature of the cube and the temperature distribution in the board is monitored using embedded thermocouples located along the centerline of the board in the flow direction.

In addition to using homogeneous substrates made of stainless steel, a series of experiments are conducted using a fiberglass reinforced epoxy board ($k = 0.41\ W/mK$) with a one ounce layer of copper attached to one side, representative of a typical two layer printed circuit board. Separate tests are conducted for cases where regions of the copper layer are selectively etched away from the fiberglass reinforced epoxy, thereby creating copper lands of various dimensions under and around the aluminum cube. This has the effect of acting as a heat spreader, thus influencing heat transfer within the board.

Thermal simulations are performed using META, an analytical-numerical model, developed by the Microelectronics Heat Transfer Laboratory. META uses an analytical fluid-side model based on the integral form of the boundary layer equations. Natural convection effects are introduced through an effective velocity term which incorporates the positional dependence of the buoyancy driven velocity, accounting for the coupled nature of the momentum and energy equations in natural convection heat transfer. The solid-side model is based on a two dimensional finite volume model with allowances for radiation heat transfer between the board, the heat source and the surrounding surfaces.

The importance of incorporating a conjugate model when simulating microelectronic applications will be shown through a comparison of results with an isolated cubic heat source, where the "fin" effect associated with the circuit board is absent, and the various cube-on-board combinations described above.

THERMAL MODEL

META, an iterative conjugate model (Culham et al., 1991), is used to simulate heat transfer in the cube-on-board combinations presented in this study. The models used in META consist of an analytical boundary layer solution based on an existing forced convection solution method (Sparrow and Lin, 1965) for laminar boundary layer flows over flat plates where the integral form of the boundary layer equations are solved, given a flux specified boundary condition. Natural convection can be simulated by substituting a local effective velocity attributed to buoyancy forces, as determined in Lee and Yovanovich, 1992, in place of the free stream velocity term associated with forced convection analyses. The resulting formulation for the local Nusselt number is used as a convective boundary condition in the solid-side model, which is a two-dimensional finite volume formulation.

Despite the apparent invalidation of the boundary layer assumptions associated with introducing a large raised object into the flow stream, it will be shown that raised objects can be accurately modeled using boundary layer models if the additional surface area for heat transfer is taken into consideration in the calculation of the thermal resistance between the wall and the free stream of the fluid.

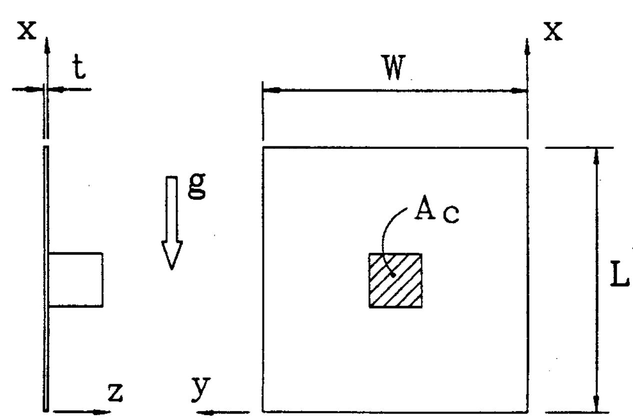

Figure 1: Local Coordinate System Used in META

In a global sense, convection heat transfer in the flow direction is assumed to dominate, therefore the transport of momentum and energy by diffusion is assumed negligible in the x-y plane, parallel to the board. Although, local regions may exist, especially in the vicinity of the raised object, where a y-component of the velocity is sizable, overall velocity components across the width of the board are small compared to buoyancy induced velocities in the x-direction. This, combined with a scale analysis, indicates that the convective as well as the diffusive transport in the y-direction can be ignored.

Since the surface area along the edges of a circuit board is much less than the area over the planar surfaces, it is reasonable to assume that the operating temperature and the heat transfer characteristics of the board are not influenced by edge effects.

The fluid-side model used in META is based on the two-dimensional boundary layer equations for forced convection over a flat plate where x is defined parallel to the principal flow direction and z is defined as the coordinate through the thickness of the board, as shown in Fig. 1. The continuity, momentum and energy equations can be written as

$$\frac{\partial u}{\partial x} + \frac{\partial w}{\partial z} = 0 \qquad (1)$$

$$u\frac{\partial u}{\partial x} + w\frac{\partial u}{\partial z} = \nu\frac{\partial^2 u}{\partial z^2} \qquad (2)$$

$$u\frac{\partial T}{\partial x} + w\frac{\partial T}{\partial z} = \alpha\frac{\partial^2 T}{\partial z^2} \qquad (3)$$

where the following boundary conditions are applied.

$$z \rightarrow \infty \quad , \quad u \rightarrow u_e \ , \ T \rightarrow T_\infty \qquad (4)$$

$$x = 0 \quad , \quad u = u_e \ , \ T = T_\infty \qquad (5)$$

$$z = 0 \quad , \quad u = w = 0 \qquad (6)$$

$$z = -t \quad , \quad u = w = 0 \qquad (7)$$

As mentioned above, these boundary conditions apply for all values of $0 \leq y \leq W$, where edge losses are assumed negligible.

For forced convection applications, the effective velocity at the outer edge of the boundary layer is equivalent to the free stream velocity, u_∞ but for natural convection applications an effective velocity must be used which accounts for local velocity changes due to buoyancy. The effective velocity for natural convection, base on the accumulated heat flow into the air stream from the leading edge of the board up to the point where the flow velocity (attributed to natural convection) is calculated as presented in Culham et al., 1991.

The solid-side model is based on the three-dimensional Laplace equation for heat flow in a homogeneous solid.

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} = 0 \qquad (8)$$

where the boundary conditions along the edges of the board, are considered to be adiabatic and can be written as

$$x = 0 \text{ and } L \quad , \quad \frac{\partial T}{\partial x} = 0 \qquad (9)$$

$$y = 0 \text{ and } W \quad , \quad \frac{\partial T}{\partial y} = 0 \qquad (10)$$

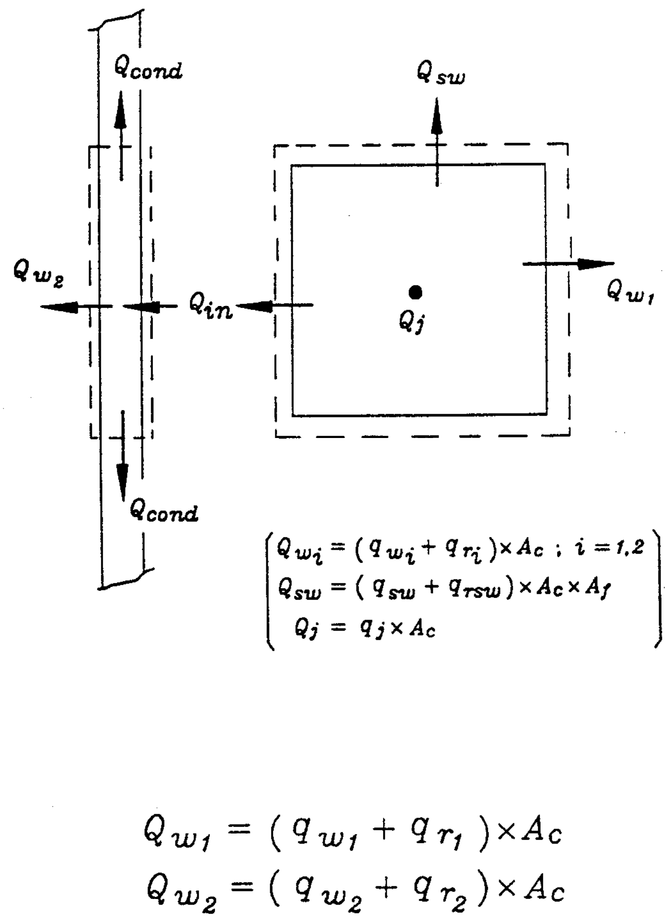

Figure 2: Global Heat Balance at Raised Heat Source

Integrating across the thickness of the board, the above three-dimensional Laplace equation, reduces to a two-dimensional "fin" equation.

$$\frac{\partial^2 \overline{T}_b}{\partial x^2} + \frac{\partial^2 \overline{T}_b}{\partial y^2} + \frac{1}{k_e t}\left[k_e \frac{\partial T}{\partial x}\right]_{-t}^{0} = 0 \qquad (11)$$

where

$$\overline{T}_b = \frac{1}{t}\int_{-t}^{0} T \, dz \qquad (12)$$

and the boundary conditions at $z = -t$ and $z = 0$ can be obtained directly from an energy balance on C.V.1. in Fig. 2 over a differential surface area ($dx \times dy$), as follows

$$z = 0 \quad , \quad k_e \frac{\partial T}{\partial z} = q_{in} \tag{13}$$

$$z = -t \quad , \quad k_e \frac{\partial T}{\partial z} = q_{w_2} + q_{r_2} \tag{14}$$

The solid-side model is based on the premise that the substrate is homogeneous across the thickness of the board. However, an effective conductivity can be obtained, as shown in Lemczyk et al., 1991, that allows a representative thermal conductivity to be calculated for layered materials. The effective conductivity is based on the harmonic mean of the conductivities calculated for predominately series and parallel heat flows. The effective conductivity can be written as

$$k_e = 2\left(\frac{1}{k_p} + \frac{1}{k_s}\right)^{-1} \tag{15}$$

where

$$k_p = \sum_{j=1}^{M}(k_j t_j) / \sum_{j=1}^{M} t_j \tag{16}$$

$$k_s = \sum_{j=1}^{M} t_j / \sum_{j=1}^{M}(t_j/k_j) \tag{17}$$

and M is the total number of layers in the board.

The convective heat flux dissipated from the back side of the board, as given in Eq. (14), can be calculated using the convective thermal resistance, R_{w_2}, calculate in the fluid side model of META, such that

$$q_{w_2} = \frac{T_{w_2} - T_\infty}{R_{w_2} A_c} \tag{18}$$

The radiative heat flux from the backside of the board is calculated as

$$q_{r_2} = \epsilon_2 \sigma (T_{w_2}^4 - T_{sk}^4) \tag{19}$$

where T_{w_2} is taken to be the wall temperature in the previous iteration within META, thus avoiding the computational complexities associated with solving a non-linear equation obtained when Eq. (19) is substituted into Eq. (11).

As shown in Fig. 2, the heat flow from the cube (C.V.2.) to the front surface of the board can be expressed as

$$Q_{in} = Q_j - Q_{w_1} - Q_{sw} \tag{20}$$

By dividing through by the interfacial area, A_c, one obtains

$$q_{in} = q_j - (q_{w_1} + q_{r_1}) - (q_{sw} + q_{rsw}) \times A_f \tag{21}$$

In order to maintain a common reference area which is implicit in Eq. (20), i.e. the plan area of the wall at the fluid-solid interface (A_c), the heat flow through the side walls must be written as

$$Q_{sw} = (q_{sw} + q_{rsw}) \cdot A_c \cdot A_f \tag{22}$$

where

$$A_f = A_{sw}/A_c \geq 0 \tag{23}$$

The individual heat fluxes in Eq. (21) can be expressed as

$$q_{w_1} = \frac{T_c - T_\infty}{R_{w_1} A_c} \tag{24}$$

$$q_{r_1} = \epsilon_1 \sigma (T_{w_1}^4 - T_{sk}^4) \tag{25}$$

$$q_{sw} = \frac{T_c - T_\infty}{R_{sw} A_c \cdot A_f} \tag{26}$$

$$q_{rsw} = \epsilon_c \sigma (T_c^4 - T_{sk}^4) \tag{27}$$

The thermal resistances are computed using the fluid-side model of META. For flush mounted heat sources, $A_f = 0$, and the resistance of the side walls, defined as $R_{sw} = (h_{conv} \cdot A_c \cdot A_f)^{-1} \rightarrow \infty$ and in turn $Q_{sw} = 0$. In this instance the last term in Eq. (20) goes to zero and the heat flux into the board, q_{in}, consists of a balance between the applied Joulean heating and the heat flux from the top of the source resulting from convection and radiation. The emissivity is assumed constant over the surface of the cube and T_{sk} is the radiative sink temperature which is taken to be T_∞ in this study.

If the $Bi = h \cdot t / k_e < 0.1$, where h is the total heat transfer coefficient including radiation and convection, then the temperature difference across the thickness of the board is small and a single average value of the cross sectional temperature can be used for the local wall temperature, T_w, such that

$$\overline{T}_b = T_{w_1} = T_{w_2} = T_w(x, y) \tag{28}$$

Equations (11), (13) and (14) can be combined as

$$\frac{\partial^2 T_w}{\partial x^2} + \frac{\partial^2 T_w}{\partial y^2} + \frac{1}{k_e t}(-q_{w_2} - q_{r_2} + q_{in}) = 0 \tag{29}$$

By substituting Eq. (21) into Eq. (29), the governing equation for heat flow within the board becomes

$$\frac{\partial^2 T_w}{\partial x^2} + \frac{\partial^2 T_w}{\partial y^2} + \frac{1}{k_e t}[q_j - q_w - q_r - (q_{sw} + q_{rsw}) \cdot A_f] = 0 \tag{30}$$

where

$$q_w = q_{w_1} + q_{w_2} \tag{31}$$

$$q_r = q_{r_1} + q_{r_2} \tag{32}$$

If the contact resistance between the cube and the board is assumed to be negligible, then $T_c = T_w$ and Eq. (30) can be solved directly. However, if a contact resistance, R_{ct}, is present at the joint between the cube and the board, the temperature drop across the contact can be written as

$$T_c - T_w = R_{ct} Q_{in} \quad (33)$$

where Q_{in} is given in Eq. (20). As with the calculation of the radiative heat flux, the non-linearity created by substituting Eq. (33) into Eq. (30) is avoided by using the wall temperature from the previous iteration in META to calculate the cube temperature in Eq. (33). The process is continued within META until convergence is obtained.

EXPERIMENTAL PROCEDURE

Testing was conducted by Mack, 1991 and Huang, 1993 to study natural convection heat transfer from isothermal aluminum cubes attached to flat plates with a wide range of thicknesses, thermal conductivities and emissivities. These factors within the plate contribute directly to the spreading of heat from the cube.

The experiments of both Mack and Huang were conducted in a vertical wind tunnel with a 300 mm × 300 mm cross section. The boards were held in a vertical rigid position using corner mounted support wires which were in turn attached to tensioning screws around the perimeter of the test section. Testing was limited to natural convection over a range of Rayleigh numbers between 5.4×10^5 to 6.7×10^6, where the characteristic length of the Rayleigh number is based on the square root of the exposed surface area of the cube. All testing was performed under steady state conditions.

Power, over a range of 0.5 - 7.0 W, was dissipated within the cube using two 82 Ω Philips PR37J metal film resistors connected in series and embedded in the center of the cube. Both power control and temperature acquisition were regulated through a Fluke Helios I data logger. Temperatures were monitored using 3 mil T-type thermocouples embedded in the cube and the board using a high conductivity aluminum based epoxy. The thermocouples were shown to have a calibrated accuracy of ±0.5°C. Conductive losses through the thermocouple wires and the power leads was estimated to be less than 1%.

Because of the significance of radiative heat losses in natural convection studies, care was taken to accurately measure the emissivity of the aluminum cube. The method used is based on the transient response of the cube during cooling and necessitates a prior knowledge of the heat capacity of the cube. An emissivity of 0.085 was measured for the cube used in both studies (Mack and Huang).

The plan dimensions of the boards used in all studies was maintained at 228.6 mm × 228.6 mm. The cube had dimensions of 43.26 mm × 43.26 mm × 43.26 mm and was mounted in the center of the board using four nylon screws. In order to minimize the contact resistance between the cube and the boards a conductive grease was applied at the interface.

RESULTS FROM META SIMULATIONS

CASE 1: Homogeneous Boards

Mack, 1991 examined the thermal response of a heated aluminum cube (43.26 mm on all sides) for steady state natural convection cooling. Tests were conducted for an isolated cube suspended in a vertical orientation with four sides parallel to the gravity vector and for the cube attached to a 304 stainless steel board, also in a vertical orientation. In the case of the cube-on-board experiment, two situations were examined, first, a SS304 plate with an emissivity of 0.3 and then a SS304 plate painted flat black, with an emissivity of 0.9, denoted as case 1B and 1C, respectively, in Table 1. For each of the cases examined, the power was fixed at 7.0 W.

Table 1: Board Specifications, Mack, 1991

	CASE	
	1B	1C
material	SS304	SS304
thickness, (mm)	1.3	1.3
conductivity, (W/mK)	13.4	13.4
emissivity	0.3	0.9

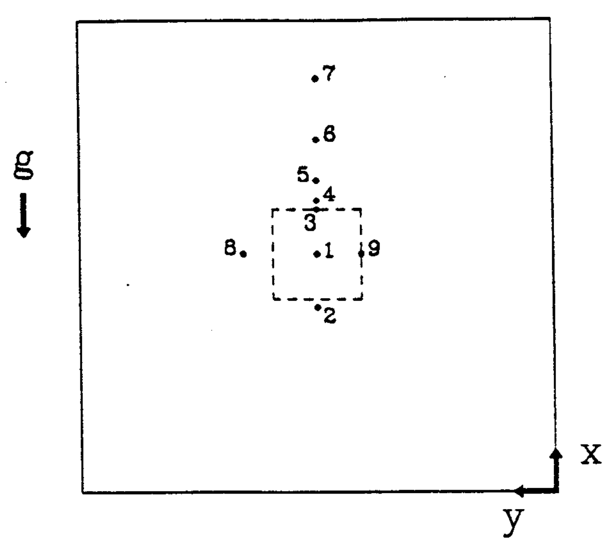

Figure 3: Thermocouple Locations Used in Table 2

Isolated Cube

An isolated cube was suspended in a wind tunnel using thin, low conductivity support wires to minimize secondary heat losses while supporting the cube in a rigid, vertical orientation (case 1A). Once the cube reached steady state, the mean cube temperature was recorded using a four thermocouple average. As shown in Table 2, the measured mean temperature excess was found to be 76.5 °C for a power input of 7 W. META was used to simulate the isolated cube by treating the cube as a raised heat source attached to a very thin, high conductivity board with the same plan area as the cube. The cube was assumed in perfect contact with a board 43.26 mm × 43.26 mm × .01 mm with a thermal conductivity and emissivity identical to the aluminum cube (180 W/mK and 0.085, respectively). Using this method, the mean cube temperature excess of the cube was calculated to be 77.8 °C, or a difference of 1.7% from the measured result.

Cube-on-Board

The effects of conjugate heat transfer were studied by attaching the cube to a 304 stainless steel board. The cube was attached to the board using four nylon screws and the contact resistance between the cube and the board was minimized by applying a thermal grease over the interface. Figure 3 shows the location of the nine thermocouples used to monitor the steady state temperature distribution in the cube and the board. The corresponding temperature rises at these locations are shown in Table 2. The effect of the board in lowering the cube temperature is clearly observed in both cases 1B and 1C. The conjugate effect reduced the cube temperature rise from 76.5 °C in the isolated cube to 42.5 °C for the cube-on-board with ϵ=0.3 to 38.2 °C for the cube-on-board with ϵ=0.9.

META simulations for cases 1B and 1C were performed using the dimensions and thermophysical properties given in Mack, 1991. A contact resistance of 1 °C/W was used for both cases. The temperature rises calculated using META are presented in Table 2. For case 1B, the cube temperature calculated using META was within 2.5% of the experimental value while temperatures off the cube had a slightly larger deviation. The largest deviation between experimental data and the META predictions, occurs at thermocouples 2, 3 and 9 which are attached to the board, immediately adjacent to the cube. The META simulations assume a uniform contact resistance over the full plan area of the cube, however, it is likely that the cube does not make intimate contact with the board around the perimeter of the cube even with the presence of an interfacial grease. This results in a lower measured board temperature at the perimeter of the cube.

Case 1C is similar to case 1B, except the board is painted flat black to raise the emissivity from 0.3, for the bare board, to 0.9 for the painted board. The change in the board emissivity produced a 10% reduction in the temperature rise of the cube. The board temperature away from the cube was reduced by up to 30% as a result of the change in board emissivity. For natural convection experiments, radiative heat losses can be of similar magnitude to the convective losses while for forced convection the significance of radiative losses are minimized as the flow velocity increases. The temperature predictions obtained for the case 1C simulation, as shown in Table 2, show good agreement at the nine thermocouple locations. Again the maximum deviation between experimental and simulated data occurs at the thermocouples around the perimeter of the cube.

CASE 2: Fiberglass-Epoxy Boards With Surface Lands

Huang, 1992 monitored the temperature of an isothermal cube attached to a fiberglass-epoxy circuit board as the thermal resistance around the perimeter of the cube was varied by introducing copper lands of various sizes to act as heat spreaders. An aluminum cube (43.26 mm × 43.26 mm × 43.26 mm), similar to that used in Mack's experiments, was attached to the center of a fiberglass-epoxy circuit board with a thickness of 1.59 mm and a conductivity of 0.41 W/mK. The contact resistance between the cube and the board was kept to a minimum by applying a thin layer of a conductive grease at the interface. Power to the resistance heater, embedded in the cube, was stepped between 0.5 and 6.5 watts in one watt increments for each test case.

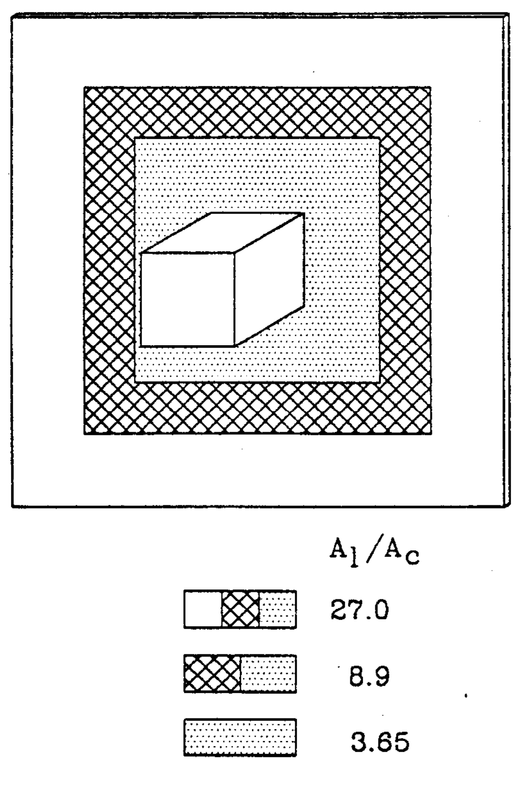

Figure 4: Copper Lands Used in Huang, 1993

Table 2: Comparison: Experimental Data (Mack, 1991) and META Simulations with $Q = 7$ W

T/C location			$\Delta T(^\circ C)$					
#	x	y	Case 1A		Case 1B		Case 1C	
	mm	mm	Mack	META	Mack	META	Mack	META
	isolated cube		76.5	77.8				
1	114.3	114.3			42.5	41.5	38.2	36.9
2	89.0	114.3			25.5	30.3	24.2	26.4
3	135.9	114.3			35.6	38.1	30.5	32.9
4	140.0	114.3			35.1	35.7	30.2	30.3
5	150.0	114.3			26.1	27.6	22.0	22.7
6	170.0	114.3			17.4	19.1	13.7	14.6
7	200.0	114.3			11.8	13.3	8.4	9.5
8	114.3	150.0			21.6	22.4	18.4	17.9
9	114.3	92.8			32.3	35.4	30.0	30.9

The experiment was initiated by testing a circuit board which was fully covered on one side with a one ounce layer of copper (0.03426 mm thick). The exposed area of the copper layer around the cube, denoted as A_l, can be related to the plan area of the cube (A_c) to provide a convenient measure of the size of the copper land. For the fully covered board, $A_l/A_c = 27.0$. Further tests were conducted by progressively etching away the copper layer closest to the edge of the board, as shown in Fig. 4, providing tests with $A_l/A_c = 8.9$, 3.65 and 0.0, respectively. Table 3 presents the circuit board and copper land specifications for each of the four cases.

Thermal simulations of Huang's experiments were performed using META. The only parameters varied between each run were power, ambient temperature and the size of the copper land.

A contact resistance of 1 $^\circ C/W$ was assumed at the interface between the cube and the circuit board. This translates into an interface conductance coefficient of $h_c \approx 500\ W/m^2K$. The emissivity of the cube, as measured (Mack, 1991), was taken to be $\epsilon_c = 0.085$. The emissivities of the fiberglass-epoxy ($\epsilon_{fe} = 0.9$) and the copper lands ($\epsilon_{cl} = 0.05$) were measured using a Gier-Dunkle DB100 infrared reflectometer.

Table 3: Copper Land Specifications, Huang, 1993

LAND	CASE			
	2A	2B	2C	2D
A_l/A_c	0.0	3.65	8.90	27.0
thickness, (μm)	34.26	34.26	34.26	34.26
conductivity, (W/mK)	386	386	386	386
emissivity	0.05	0.05	0.05	0.05

Tables 4-7 show the agreement between isothermal cube temperatures from experimental data (Huang, 1993) and the predicted results of META. Agreement over the full range of studies is 1 °C. The studies clearly show that the heat spreading effect of the copper land lowers the temperature of the cube by approximately 9 - 10 °C. However, this reduction in cube temperature is not significantly affected by increasing the size of the land from $A_l/A_c = 3.65$ to $A_l/A_c = 27.0$. The magnified region in Fig. 5 for $A_l/A_c < 4.0$, shows that more than 80% of the reduction in cube temperature occurs over the range $A_l/A_c < 2.0$. The addition of a larger copper land does not significantly contribute to heat dissipation.

Table 4: Case 2A - Cube on FR4 Board with $A_l/A_c = 0.0$

$Q\ (W)$	$T_\infty\ (^\circ C)$	$\Delta T_c(^\circ C)$	
		Huang	META
6.5	20.6	57.3	56.6
5.5	20.2	50.0	49.3
4.5	20.0	42.4	41.5
3.5	20.0	34.5	33.4
2.5	20.3	25.9	25.0
1.5	20.6	16.8	16.1
0.5	20.9	6.5	6.1

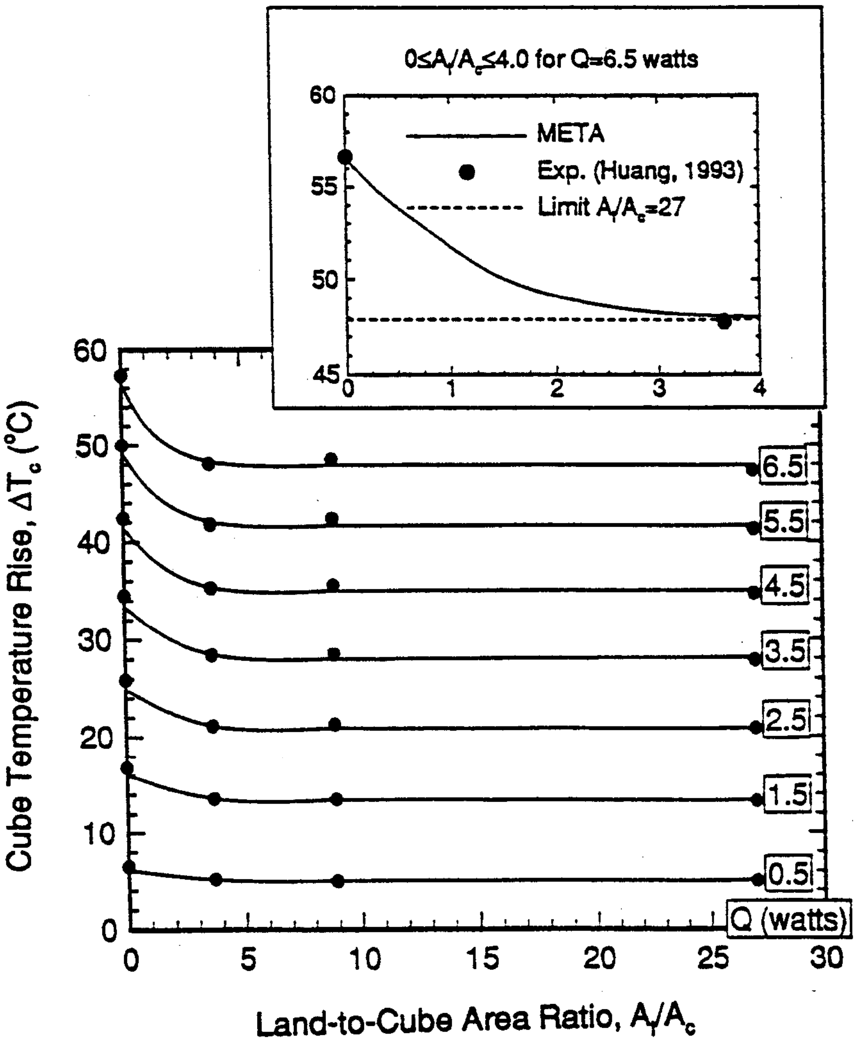

Figure 5: Comparison of Experiments and Predictions for Copper Lands of Various Sizes

Table 5: Case 2B - Cube on FR4 Board with $A_l/A_c = 3.65$

$Q\ (W)$	$T_\infty\ (^\circ C)$	$\Delta T_c (^\circ C)$	
		Huang	META
6.5	21.5	48.1	48.4
5.5	21.3	41.8	42.1
4.5	20.5	35.3	35.4
3.5	21.0	28.5	28.5
2.5	20.7	21.2	21.2
1.5	20.7	13.6	13.6
0.5	19.0	5.2	5.2

Table 6: Case 2C - Cube on FR4 Board with $A_l/A_c = 8.9$

$Q\ (W)$	$T_\infty\ (^\circ C)$	$\Delta T_c (^\circ C)$	
		Huang	META
6.5	21.3	48.5	47.9
5.5	21.1	42.3	41.6
4.5	22.5	35.5	35.0
3.5	22.6	28.5	28.0
2.5	21.7	21.3	20.9
1.5	21.3	13.5	13.5
0.5	22.0	5.0	5.0

Table 7: Case 2D - Cube on FR4 Board with $A_l/A_c = 27.0$

$Q\ (W)$	$T_\infty\ (^\circ C)$	$\Delta T_c (^\circ C)$	
		Huang	META
6.5	21.1	47.4	47.9
5.5	19.7	41.3	41.6
4.5	21.1	34.7	35.0
3.5	21.2	27.9	28.2
2.5	21.0	20.9	20.9
1.5	20.1	13.3	13.4
0.5	20.8	5.0	5.0

SUMMARY AND CONCLUSIONS

A thermal analysis model based on the boundary layer equations can be used to accurately predict wall temperatures for natural convection applications with large raised objects attached to vertical flat substrates. The heat dissipated over the source can be determined based on an equivalent heat transfer coefficient ($1/RA$) where the area, A is based on the wetted surface area of the raised object and the thermal resistance, R, is based on conventional boundary layer methods for calculating the resistance between a flat surface and a surrounding free stream.

Radiative heat losses and contact resistance are also important factors in conjugate applications, especially for natural convection cooling where convection and radiation are of equal magnitude.

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