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Experimental and Theoretical Study of Heat Transfer by Natural Convection of a Heat Sink Used for Cooling of Electronic Chip

[1]Sunil Hireholi, [2]K.S. Shashishekhar, [3]George. S. Milton
[123]Dept. of Mechanical Engineering
[12]S. I. T, [3]S. S. I. T
[123]Tumkur, Karnataka, India

Abstract:- In the present work, heat transfer analysis of a commercially available heat sink is carried out. The heat sink is used for cooling of electronic chip STK4141II and removes the heat generated by the chip through natural convection. The work includes experimental investigations and theoretical modelling.
In experimental investigations electrical heating is used to supply heat to the heat sink and the temperature of the heat sink is measured using RTD thermocouples attached to the heat sink. Theoretical temperature of heat sink is predicted employing 2-D modelling of the heat transfer process based on fundamental heat transfer principles. Theoretically predicted temperatures of the heat sink are compared with the measured temperatures.

Keywords:- heat sink, electronic cooling, RTD thermocouples.

I. INTRODUCTION

Advances in the field of electronics have resulted in significant increase in density integration, clock rates and emerging trend of miniaturization of modern electronics. This resulted in dissipation of high heat flux at chip level. In order to satisfy the junction temperature requirements in terms of performance and reliability, improvements in cooling technologies are required. As a result thermal management is becoming important and increasingly critical to the electronics industry. The task of maintaining acceptable junction temperature by dissipating the heat from integrated circuit chips is significant challenge to thermal engineers. The electronic cooling is viewed in three levels, which are non-separable. First, the maintenance of chip temperature at the relatively low level despite the high local heat density. Second, the heat flux must be handled at system or module level. Finally, the thermal management of the computer machine room, office space or telecommunication enclosure [1].

In electronic systems, a heat sink is a passive component that cools a device by dissipating heat into the surrounding . Heat sinks are used to cool electronic components such as high-power semiconductor devices, and optoelectronic devices such as higher-power lasers and light emitting diodes (LEDs). In the present work we carry out analysis of heat transfer by a commercially available heat sink used for chip level cooling.

II. LITERATURE SURVEY

The cooling aspects have been studied by many investigators. Among them Mobedi and Sunden [2] investigated the steady state conjugate conduction-convection on vertically placed fin arrays with small heat source inside. Thermal performance of a free standing fin Structure of copper heat sink has been reviewed by Jan Bijanpourian [3]. In his work he had used CFD to evaluate and compare heat dissipation capabilities of aluminium and copper heat sinks. Experimental Investigation of Pin Fin Heat Sink Effectiveness for forced convection is reviewed by Massimiliano Rizzi et al [4]. One of the earliest studies about natural convection heat transfer from fin arrays was conducted by Starner and McManus [5]. Four different fin array configurations with three base types were investigated and heat transfer coefficients were calculated. Leung et al. [6] performed an experimental study on heat transfer from vertically placed fin arrays produced from an aluminium alloy . It was found that for different configurations the maximum heat transfer rate from the fin arrays was obtained at the fin spacing value of 10 mm. Harahap et al. [7] performed experiments on miniaturized vertical rectangular fin arrays in order to investigate the effect of miniaturizing on steady state rate of natural heat transfer. A numerical analysis on natural convection heat transfer from horizontally placed rectangular shrouded fin arrays were performed by Yalcin et al. [8]. Commercially available CFD package PHOENICS was used to solve the three dimensional elliptic governing equations. Yüncü and Mobedi [9] investigated the three dimensional steady state natural convection from horizontally placed longitudinally short rectangular fin arrays numerically. A finite difference code in Cartesian coordinate system based on vorticity-vector potential approach was used to solve the problem. Kundu and Das [10] performed an analytical study to investigate performance and optimum design analysis of four fin array types viz, longitudinal rectangular fin array, annular rectangular fin array, longitudinal trapezoidal fin array and annular trapezoidal fin array under convective cooling conditions are investigated. Another experimental study regarding the natural convection heat transfer from both vertically and horizontally oriented fin arrays was done by Leung and Probert [11]. The effects of fin spacing and fin height were investigated for a limited number of fin array configurations. Vollaro et al. [12] analysed natural convection from rectangular and vertical finned plates numerically in order to optimize the fin configuration. The maximum heat transfer rate from fin array was investigated for the optimum fins spacing as a function of parameters such as dimensions, thermal conductivity, fins absorption coefficient and fluid thermo-physical properties.

III. EXPERIMENTAL STUDIES

The front, top and side views of the heat sink analysed are shown in Fig. 1. It consists of a vertical base and seven horizontal fins as per the dimensions shown in the figure in millimetres. In the experimental studies electrical heating is employed to supply heat to the heat sink. The electrical heater has same nominal dimensions as the electronic chip and occupies the same physical location on the heat sink as the chip. The thermocouples used for experiment were calibrated.

A. Experiment to determine heat transfer coefficient with heating coil

The method used for determining the heat transfer coefficient is to give a known heat input generated by electrical heating and measure the temperature attained by the heat sink. The temperature is measured using the RTD-thermocouples at three locations on the model and average of all the three values, T_1, T_2, T_3, is taken for the calculation of overall heat transfer coefficient. It is made sure that steady state conditions are reached before the temperature are taken and then the corresponding heat transfer coefficient is computed. The temperatures are taken at regular intervals till steady state is reached. The values of the temperature after it has reached steady state are used in calculating the corresponding heat transfer co-efficient for the given heat input. This is done for varying values of heat inputs

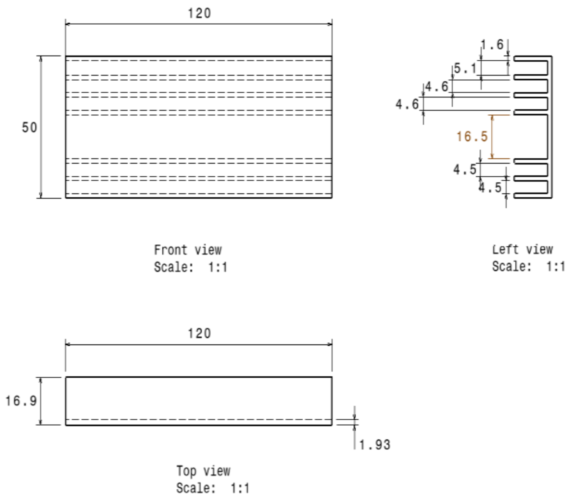

Fig.1 Front, top and side views of the heat sink

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Fig. 2 shows a photograph of the Experimental set-up used for measurement of temperature. The heat sink is kept at a height from the surface of the table to enable free flow of air from the bottom of the sink. Alternating current from mains is supplied to the heating coil attached to the heat sink through a dimmerstat (auto-transformer) as seen in the photograph given in Fig. 2. Voltage across the heating coil is measured using a voltmeter and the current is measured using an ammeter. as described above. Heat dissipated and the associated heat transfer co-efficient are calculated using the following basic equations [13], [14].

$$Q_{input} = V\,I \qquad (1)$$

$$T_{avg} = \frac{T1+T2+T3}{3} \qquad (2)$$

$$h = \frac{Q\text{input}}{A(Tavg - Tamb)} \qquad (3)$$

Where,

Q_{input} = Heat input by the heating coil (Watts)

V = Voltmeter reading (Volts),
I = Ammeter reading (Amperes),
T_1,T_2,T_3=Temperature readings of Thermocouples (^oC)
T_{avg} = Average temperature of heat sink (^oC),
h = Heat transfer coefficient (W/m^2K) ,
A = Effective area of the heat sink available for heat transfer,
T_{amb}= Ambient temperature (^oC)

Results are given in **Table 1**.

Fig.2 Experimental setup

Table 1 shows time in seconds and the corresponding temperature readings of the three thermocouples T_1, T_2, T_3, for different values of the heat inputs along with their voltage and current values. The average temperature, T_{avg} is also given in the table.

Table 1 Time history of temperature showing how steady state is reached for heat sink with heating coil kept in open air for 26 ^oC ambient temperature.

Volt-age (Volts)	Current (Amps)	Power Input (W)	Time (s)	T_1 (^0C)	T_2 (^0C)	T_3 (^0C)	T_{avg} (^oC)
10	0.23	2.3	0	26	26	26	26
			65	28	28	28	28
			289	30	30	31	30.33
			608	32	33	34	33
			1034	33	34	35	34
			1584	34	34	36	34.66
			2184	34	35	36	35
13	0.31	4.03	0	32	32	32	32
			122	33	34	34	33.67
			269	35	35	35	35
			401	36	38	38	37.33
			723	38	40	40	39.37
			1055	41	41	41	41
			1655	41	41	42	41.33
16	0.38	6.08	0	36	36	36	36
			11’’	38	39	39	38.67
			188	40	40	41	40.33
			367	42	43	44	43
			812	44	46	46	45.33
			1412	44	47	47	46

19	0.44	8.36	0	32	32	32	32
			84	36	37	37	36.67
			350	39	40	40	39.67
			619	44	46	46	45.33
			1138	49	51	52	50.67
			1738	49	53	53	51.67

Fig. 3 shows the plots of the time versus the average temperature, T_{avg}, , for different values of the heat input. From this figure one can observe the temperature reaching steady state for each case of the heat input. The final value of the average temperature T_{avg} , is plotted against the heat input in Fig. 4.

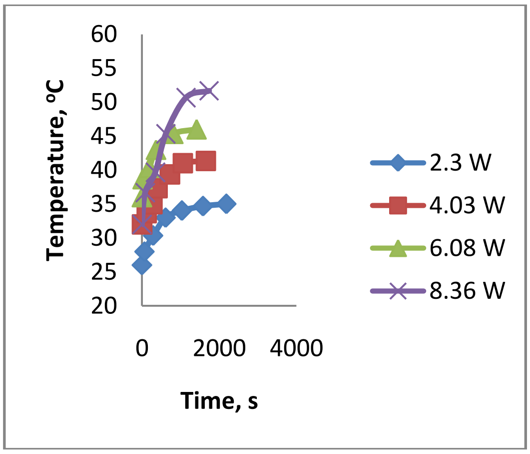

Fig. 3 Time history of the average temperature for different values of heat input

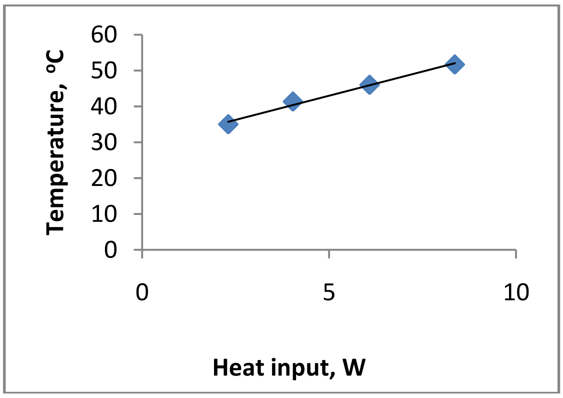

Fig. 4 Steady state values of average temperature of the heat sink plotted against the corresponding heat input values.

The overall heat transfer co-efficient is computed for the above heat input values from equation (3). The values of the heat transfer co-efficient are presented in Table 2 and are plotted against ($T_{avg} - T_{amb}$) in Fig. 5.

Table 2 Heat transfer coefficient values computed from steady state temperatures obtained from experiments for heat sink with heating coil kept in open air (ambient temperature, $T_a = 26\ ^oC$).

Voltage (Volts)	Current (Amperes)	Heat input (Watts)	Temperature, T_{avg} (^oC)	Heat transfer coefficient, h ($W/m^{2o}C$)
10	0.23	2.3	35	11.77
13	0.31	4.03	41.33	12.11
16	0.38	6.08	46	14.00
19	044	8.36	51.67	15.00

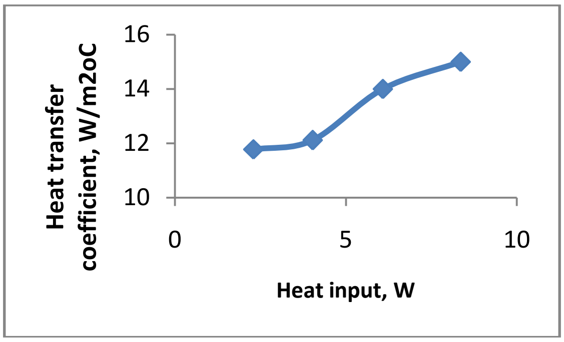

Fig. 5 Heat transfer coefficient values as computed from experimentally obtained steady state temperatures of the heat sink.

IV. THEORETICAL MODELLING

2-D Analysis

In 2-D analysis, the 2-D conduction equation with heat generation is solved. The equation is as given below [14]

$$\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) + \frac{Q}{k} = 0 \qquad (4)$$

Where Q the source term and k is is the thermal conductivity of the material.

Eq. (4) is used to model the heat transfer in the vertical base plate. The finite volume discretization is used to approximate the partial differential equation and the resulting algebraic equations are solved using Gauss-Seidal approach. The vertical base plate is divided into a number of finite volume cells. A typical finite control volume around a point P of the plate is shown in the Fig. 6. T_P represents the temperature at P while T_W, T_E, T_S and T_N represent the temperatures at the West, East, South and North neighbours of the point P. Δx and Δy are the dimensions of the control volume while $(\delta x)_w$, $(\delta x)_e$, $(\delta y)_s$, and $(\delta y)_n$ are the distances of the West, East, South and North neighbours from the point P respectively. The finite volume discretization equation can be written, following the notation in Patankar [15], as

$a_P T_P = a_E T_E + a_W T_W + a_N T_N + a_S T_S + b$ (5)

Where

$a_E = \frac{k_e t \Delta y}{(\delta x)_e}$, (5a)

$a_W = \frac{k_w t \Delta y}{(\delta x)_w}$, (5b)

$a_N = \frac{k_n t \Delta X}{(\delta y)_n}$, (5c)

$a_S = \frac{k_s t \Delta y}{(\delta y)_s}$ (5d)

t = thickness of the plate

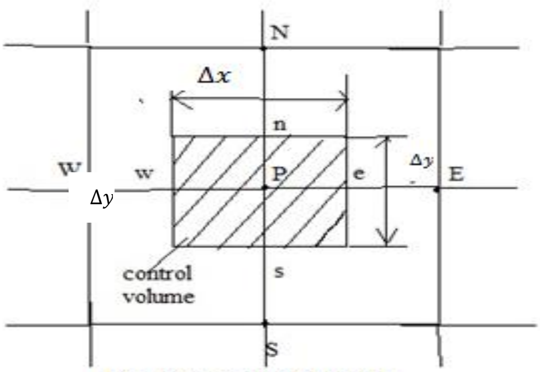

Fig. 6 The control volume

The source term Q can have the following components

$Q = Q_{gen} + Q_{flux} + Q_{conv} + Q_{fin}$ (6)

where

Q_{gen} =heat generated within the control volume

$= q_{gen}\, t \Delta x\, \Delta y$, (6a)

Q_{flux}=heat flux through the surface

$= q_{flux}\, \Delta x\, \Delta y$, (6b)

Q_{conv}=convective heat transfer with the surroundings

$= h_{face}\,\Delta x\,\Delta y\,(T_P - T_{amb}),$ (6c)

Q_{fin}=heat transferred with the surroundings through fin

$= \eta h_{fin}\, A_{fin}(T_P - T_{amb})$ (6d)

The coefficient a_p and the constant b in Eq. 5 can now be given as,

$a_P = a_E + a_W + a_S + a_N + h_{face}\,\Delta x\,\Delta y + \eta h_{fin}\, A_{fin},$ (7)

$b = q_{gen}\, t\Delta x\,\Delta y + Q_{flux}\,\Delta x\,\Delta y + h_{face}\,\Delta x\,\Delta y\, T_{amb} + \eta h_{fin}\, A_{fin} T_{amb}\,.$ (8)

In the equations above q_{gen} is the heat generated per unit volume, q_{flux} is the heat flux per unit area, h_{face} is the heat transfer coefficient for exchanging heat with surroundings, η is fin efficiency, h_{fin} is the effective heat transfer coefficient of fin, A_{fin} is the effective surface area of fin through which heat is exchanged with surroundings and T_{amb} is the ambient temperature.

The heat transfer coefficient h_{face} is modelled using the standard correlations [13], [14]:

$Nu = 0.59\,(Gr\,Pr)^{0.25}$ (9)

Where

Nusselt number $Nu = \frac{h_{face}\; y}{k}$ (9a)

Grashof number $Gr = \frac{g\beta\,\Delta T y^3}{\vartheta^2}$ (9b)

Prandtl number $Pr = \frac{C_p \mu}{k}$ (9c)

Fin efficiency η is given by

$\eta = \frac{\tan h\,(mL)}{mL}$ (10)

Where

$mL = L_c^{1.5}(\frac{h}{kA_m})^{0.5}$ (10a)

$L_c = L + \frac{t}{2}$ (10b)

$A_m = t_{fin}\, L_c$ (10c)

t_{fin} is thickness of fin.

The heat transfer coefficient h_{fin} is modelled using the standard correlations [13][14]:

for upper surface of fin

$Nu = 0.54\,(Gr\,Pr)^{0.25}$ (11)

for lower surface of fin

$Nu = 0.27\,(Gr\,Pr)^{0.25}$ (12)

h_{fin} is obtained from equations (11) and (12) by first obtaining the average Nusselt number as

$Nu = 0.405\,(Gr\,Pr)^{0.25}$ (13)

$Nu = \frac{h_{face}\; Le}{k}$ (14)

The Grashof number in equations (11), (12) and (13) is given by

$Gr = \frac{g\beta\,\Delta T Le^3}{\vartheta^2}$ (14a)

$Pr = \frac{C_p \mu}{k}$ (14b)

where L_e is given by $L_e = \frac{l_{fin}\, b_{fin}}{2(l_{fin} + b_{fin})}$ (14c)

A_{fin} in equation (6d) is given as

$A_{fin} = 2\Delta x l_{fin}$ (15)

where

l_{fin} is length of the fin,

b_{fin} is breadth of the fin.

Boundary Conditions:

Convective heat transfer boundary condition has been applied on all the four edges of the plate viz.

At the boundary, in general,

$k(\frac{\partial T}{\partial n})_B = h(T_B - T_{amb})$ (16)

Where T_B is the temperature at the boundary,

$(\frac{\partial T}{\partial n})_B$ is the temperature gradient at the boundary.

The resulting algebraic equations are solved using Gauss-Seidal approach as mentioned earlier. Iteration method has been used for achieving the convergence of the solution of the equations. A computer programme has been written in C language for computation of temperature field.

A mesh of size 60x32 has been used for the solution. It is uniform in x-direction and non-uniform in y-direction. The details of mesh co-ordinates are given in the appendix.

V. RESULTS AND DISCUSSION

The value of heat input and the corresponding heat flux value is given as input to the C-program to get the temperature field by using the approach as given in section IV above. Temperature values corresponding to the three thermocouple locations are noted from the temperature field and average of these three temperatures, T_{avg} is calculated. This is done for different values of the heat input. Heat input values, and the corresponding heat flux values, the average temperature of the heat sink and heat transfer coefficient is shown in Table 7. Heat transfer coefficient is computed from equation similar to Eq. (3). Fig. 7 shows the average temperature of the heat sink plotted against the corresponding heat inputs. Fig. 8 shows the heat transfer coefficient values plotted against the corresponding heat inputs. Temperature profile showing the variation of temperature along the mid-line (x=60e-3) as function of the vertical distance y is shown in Fig. 9.

Table 7 Heat sink temperature as evaluated by 2D code for heat sink with heating coil kept in open air

Heat input (W)	Heat flux (W/m^2)	Temperature, T_{avg} (^oC)	Heat transfer coefficient, h ($W/m^{2o}C$)
2.3	1288.52	34.88	11.93
4.03	2257.70	39.72	13.54
6.08	3406.16	44.60	15.06
8.36	4683.47	49.99	16.06

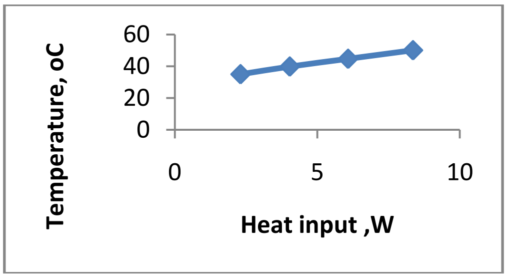

Fig. 7 Temperature obtained by 2-D analysis

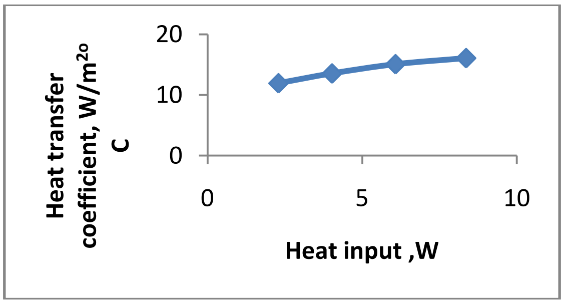

Fig. 8 Heat transfer coefficient obtained by 2-D analysis

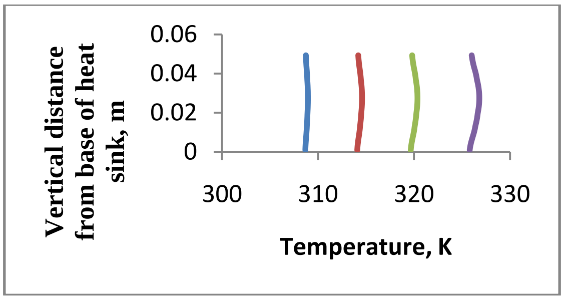

Fig. 9 Temperature profile 2.3 w , 4.03 W , 6.08 W , 8.36 W

V. COMPARISION OF EXPERIMENTAL RESULTS WITH THEORETICAL RESULTS

Comparison between heat sink temperatures from experiment and theory is shown in Fig. 10.

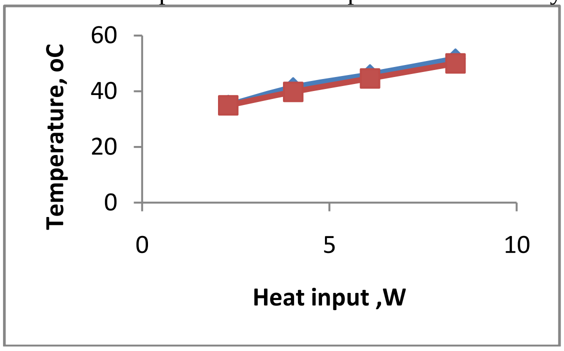

Fig. 10 Comparison between heat sink temperatures from experiment and theory

Comparison between heat transfer coefficient from experiment and theory is shown in Fig. 11

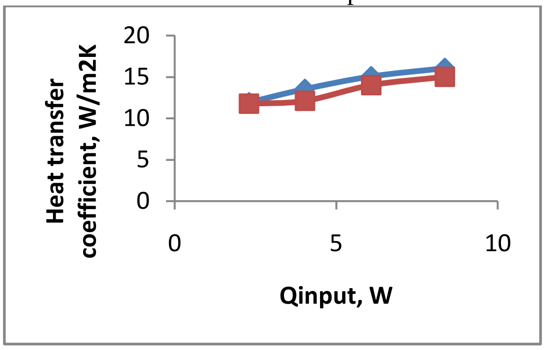

Fig. 11 Comparison between heat transfer coefficient from experiment and theory

VI. SUMMARY AND CONCLUSION

Temperature of heat sink is measured experimentally. The experimentally measured temperatures have been compared with those predicted by the theory and have been found to compare well with each other.

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APPENDIX

The mesh used in this work (section IV) of 60x32 i.e. it has 60 divisions in x-direction and 32 divisions in y-direction.

The mesh is uniform in x-direction with $\Delta x = 0.002\ m.$ The y co-ordinates are as shown below:

y (in m) = 0.000000, 0.001563, 0.003125, 0.004688, 0.006250, 0.007813, 0.009375, 0.010938, 0.012500, 0.014063, 0.015625, 0.017188, 0.018750, 0.020313, 0.021875, 0.023438, 0.025000, 0.026563, 0.028125, 0.029688, 0.031250, 0.032813, 0.034375, 0.035937, 0.037500, 0.039062, 0.040625, 0.042187, 0.043750, 0.045312, 0.046875, 0.048437, 0.050000.