EXPERIMENTAL STUDY OF THE THERMAL EFFICIENCY OF HEAT SINKS USED IN PROCESSORS

Martins, Karlos Roberto da Silva Braga, karlos@fem.unicamp.br
Gonçalves, Rubens Soeiro, soeiro@cefet-ma.br
Santana , André Pereira, andre.persan@gmail.com
Instituto Federal de Educação, Ciência e Tecnologia do Maranhão – IFM; Campus Monte Castelo
Departamento de Mecânica e Materiais
Av Getúlio Vargas 04, Monte Castelo, CEP 65025-001, São Luís-MA, Brasil

Abstract. *This work presents an experimental study of the thermal exchange efficiency of a heat sink used in a 1.67 Ghz microcomputer with a 224 MB RAM. The heat sink was analyzed when it was submitted to different temperatures and computational charges. In addition, the theoretical model of a fin was obtained according to the best fitting experimental data. The experimental and theoretical heat transfer coefficient, the heat transfer rate and efficiency of the fin were also compared. The study showed that heat sink is more efficient at 19ºC.*

***Keywords**: heat sinks, processor, thermal efficiency, fitting experimental data*

1. INTRODUCTION

The usefulness of heat sinks is based upon their general applicability to most of electronic devices with specific heat removal demands. The thermal engineering researchers' interest in these electronic devices was stimulated by the increase of the microprocessor processing capacity, demanding more and more efficient heat sink projects in order to guarantee a reliable operational temperature.

Among the several electronic devices available in the market, there are the so-called processors, which are the ones responsible for a microcomputer processing speed. This microprocessor processing capacity has increased with the demand of an increasing electric power dissipation in its inside (Souza, 2005). A semiconductor device operating at temperatures higher than the specified ones may be damaged and consequently endanger the processor functionality. A solution to maintain the semiconductor devices at a desirable level, as well as to guarantee a very good operation of an electronic device is the use of intensification mechanisms of heat exchange, the heat sinks, and then the importance of investigating their thermal exchange efficiency is emphasized, chiefly due to their use in thermal surroundings in which the temperatures are different from the one this device was designed for and this directly affects the heat exchange of a heat sink fins. This work investigated the thermal exchange efficiency of a heat sink used in processors, which operated in a temperature between 19 and 30 Celsius degree, a typical one for São Luís-MA city.

2. MATERIALS AND METHODS

The PC used was a 1.67 Ghz one, with a 224 MB RAM. The chip – the processor – was 7.47 mm x 11.33 mm. Electrical features: voltage – 1.60 volts; maximum current 38.75 A. Thermal features: maximum dissipated power - 62 W; maximum temperature operation - 90 Celsius degrees.

Figure 1 shows the heat sink analysed. It is made up of rectangular parallel aluminium plates, with a copper plate in its base which accelerates the heat conduction to its aluminium fins. The lateral dimensions of the heat sink are: 60 mm long at frontal base and 75 mm deep. The fins analysed were 36 mm long, 1 mm thick and 34 mm wide, with a 2 mm space between them. When the PC is turned on, an air flow is forced by the top of the heat sink by means of a 12 volt DC and 0.16 A fan coupled to it.

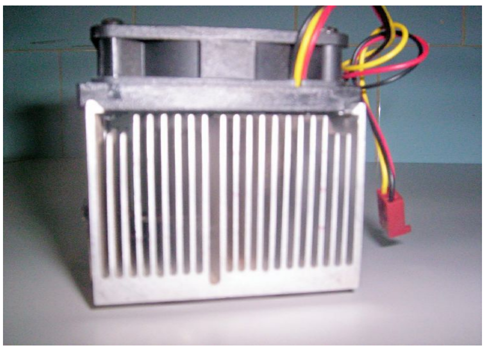

Figure 1. Heat sink analysed

The data acquisition plate had 8 inputs for type k thermocouples with output connected to a computer. Temperature accuracy: approximately 0.2% reading and $\pm 0{,}5^{o}C$. Speed accuracy: approximately 0.2% reading and $\pm 10\ \mu V$. Resolution: 20 bits. For the air flow speed measurement, provided by the fan, an anemometer with measurement range of 0.4 to 30 m/s and 0.1m/s resolution was used.

3. EXPERIMENTAL PROCEDURE

In the experimental tests, the PC above mentioned was turned on in two distinct situations: in the first condition, operating with 35% of its processing maximum charge and in the second one, operating with 75% of its processing maximum charge. In both situations five thermocouples in equidistant points were fixed to the heat sink fins. The first one was fixed to the heat sink base; the second one , 1/4 distant from the base; the third one, 2/4; the fourth one, ¾, and the last one to the top of the fin. Insulated metallic claws for fixing the thermocouples to the fins were used. Experimental data were obtained according the two conditions previously defined and also in different thermal surroudings, that is, at 19ºC and 35ºC. These temperatures were selected due the lowest one be not representative whereas the highest one is considered to be the local average temperature (São Luís-MA). Experimental measurements of the temperature profile in the fins, according to length, were undertaken , taking into consideration the normal direction of the processor, the fins at central and edge positions of the heat sink. The sampling interval considered was 10 sec. During the tests, a thermal paste of zinc oxide base was used to improve the thermal contact between the heat sink base and the processor.

3.1. Theoretical Treatment

Taking the one-dimensional treatment in longitudinal (*x)* direction into consideration, although the heat conduction inside the fin be actually bidimensional, the rate at which the energy flows to fluid by convection, at any point of the fin surface, must be equal to the liquid rate at which the energy reaches that point due to the conduction in normal direction (*y,z*). Once in a pragmatic situation the fin shows to be thin and temperature variations in normal direction in its inside are low, when compared to temperature differences between the fin and the adjacent fluid, it can be thus considered that the temperature is uniform along at the thickness of the fin, that is, it is only *x*-direction function

The equations for heat transfer (Incropera, 2001) in rectangular fins (temperature profile, heat transfer rate and efficiency) are presented below, based on some considerations: steady-state, unidirectional conduction in *x*, steady thermal conductivity and radiation on low surface, without internal generation heat, and uniform thermal convection along the fin surface, are given by the three fin's models: infinite, adiabatic and convective (Eq. 1 to 11). According Zografos and Sunderland (1990) the radiation can be despised due to the temperature range used in these experiments.

Infinite fin model:

$$\theta(x) = (\theta_b).e^{-mx} \; ; \tag{1}$$

$$q_a = \sqrt{h.P.k.A_{sr}}.\theta_b \; ; \tag{2}$$

$$\eta_a = \frac{1}{m.L} ; \tag{3}$$

Adiabatic fin model:

$$\theta(x) = .\theta_b \frac{\cosh\ m(L - x)}{\cosh\ m.L} \; ; \tag{4}$$

$$q_a = \sqrt{h.P.k.A_{sr}}.\theta_b.\tanh(m.L) ; \tag{5}$$

$$\eta_a = \frac{\tanh(\ m.L)}{m.L} ; \tag{6}$$

Convective fin model:

$$\frac{\theta(x)}{\theta_b} = \frac{\cosh\ m(L-x) + \left(h/_{m.k}\right) senhm\ .(L-x)}{\cosh\ m.L + \left(h/_{m.k}\right) senhm\ .L} \quad ; \tag{7}$$

$$q_a = \sqrt{h.P.k.A_{sr}}\,.\theta_b.\frac{senhmL. + \left(h/_{m.k}\right)\cosh m.L}{\cosh m.L + \left(h/_{m.k}\right) senhm.L} \quad ; \tag{8}$$

$$\eta_a = \frac{\tanh(m.L_c)}{m.L_c}, \quad L_c = L + t/2 \quad ; \tag{9}$$

$$\theta(x) = T(x) - T_a; \ \theta_b = T_b - T_a \tag{10}$$

where: $\theta(x)$ - excess temperature; $T(x)$ - temperature; T_b - temperature at the base; T_a - room temperature; q_a - Heat transfer rate; η_a - thermal exchange efficiency; *h* - heat transfer coefficient; *P* – perimeter; *k* - Thermal conductivity; A_{sr} - cross-sectional area; L – length; t – thickness.

and the parameter "*m*" of the fin is given by:

$$m^2 = \frac{h.P}{k.A_{sr}} \tag{11}$$

To calculate the theoretical heat transfer coefficient *h*, the empirical Nusselt's correlation (Homan, 2005) Eq. (12) was used, where the air flow between the fins was characterized by Reynolds' numbers; the hydraulic diameter was based in the input dimensions of the heat sink channel. The relationship between Reynolds and Grashof number will be available $\left(Gr_L/_{\mathrm{Re}^2{}_L}\right)$ to check what the predominate effect, i.e. the natural-convection or forced-convection. This way, the Nusselt's number was calculated using correlation to rectangular pipe assuming uniform surface temperature.

$$N_u = h.D_h/_{k_{ar}} \tag{12}$$

4. RESULTS AND DISCUSSION

The temperature profile of the system is shown in Fig. 2. The steady-state occurs at approximately after 16 minutes to a consumption charge of 35% of the maximum PC charge, at a room temperature of 19ºC.

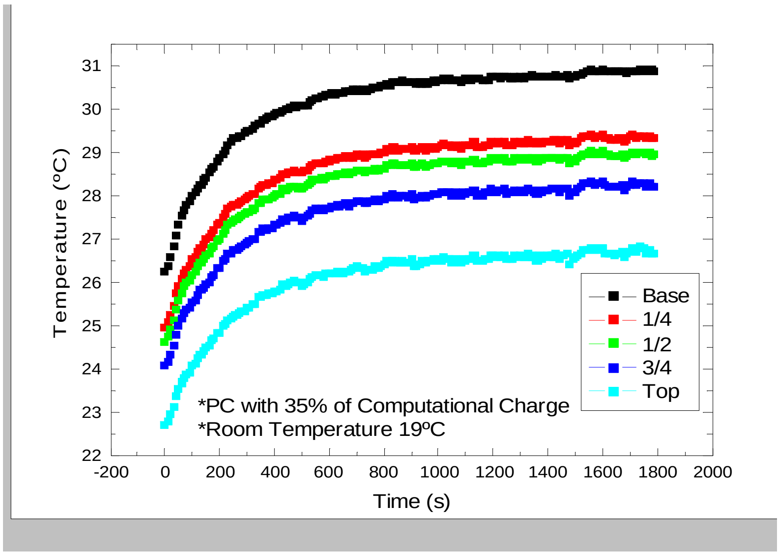

Figure 2. Temperature profile of the system for central fin

Table 1 shows the experimental tests, indicating the time to steady-state and the operational temperature at the base of the heat sink in steady-state. In all experiments the time to steady-state and the operational temperature at the base were 25 min and 36 ºC in average, respectively.

Table 1. Experimental tests

EXPERIMENTS: Position of the fin	Room Temperature (ºC)	Computational Charge (%)	Time to Steady-State(mim)	Operational Temperature at the Base (ºC)
1 – CENTRAL	19	35	16	30.64
2 - CENTRAL	19	75	25	30.87
3 - CENTRAL	30	35	27	41.13
4 - CENTRAL	30	75	30	42.45
5 – EDGE	30	35	16	41.16
6 – EDGE	30	75	28	41.65

Figure 3 shows a comparison between experimental and theoretical temperature profiles according to the three fin models: infinite, adiabatic and convective to respect to experiment 1.

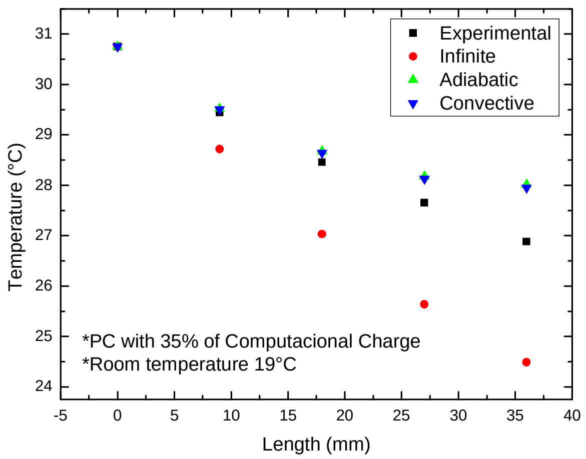

Figure 3. Comparative profile between experimental and theoretical temperature

A mathematical adjustment was undertaken by means of the minimum square method, using the experimental and theoretical temperature profile, in order to determine the theoretical model that would better fit to the experimental data, enabling then to get the *m* fin coefficient, the *h* heat transfer coefficient, *n* efficiency, the q heat transfer rate of the heat sink, and the contact thermal resistance. These results are shown in Table 2. An analysis of the r^2 determination coefficients indicate that the best fin model for experimental data was the convective one.

Table 2. Adjustment by Minimum Square Method.

EXPERIMENTS		**Adjustments**			
Fin's Model		m $[m^{-1}]$	T_a $[°C]$	T_b $[°C]$	r^2
1	Infinite	11.47	19	30.76	0.9501
	Adiabatic	24.34	19	30.75	0.96009
	Convective	24.07	19	30.75	0.96281
2	Infinite	9.7	19	30.84	0.92615
	Adiabatic	22.39	19	30.84	0.9899
	Convective	22.15	19	30.84	0.99044
3	Infinite	10.71	30	41.20	0.9539
	Adiabatic	23.56	30	41.20	0.98629
	Convective	23.30	30	41.20	0.98783
4	Infinite	10.82	30	42.46	0.95837
	Adiabatic	23.72	30	42.46	0.99247
	Convective	23.47	30	42.46	0.99357
5	Infinite	9.92	30	41.21	0.97463
	Adiabatic	22.58	30	41.21	0.98187
	Convective	22.33	30	41.21	0.8363
6	Infinite	10.16	30	41.19	0.89022
	Adiabatic	23.02	30	41.65	0.99573
	Convective	22.77	30	41.65	0.99517

Table 3 presents the experimental and theoretical *h* heat transfer coefficient given by empirical correlation *Nusselt. Reynolds*' numbers $\mathrm{Re} < 1.500$ indicate that the laminar flow between the fin channels (ducts). It also presents the experimental and theoretical efficiency. As the relationship between Reynolds and Grashof number $\left(Gr_L / \mathrm{Re}^2_L \approx 0.86 \right)$ characterizes that there are natural convection and forced together. This way, the Nusselt's number was calculated using correlation to rectangular pipe assuming uniform surface temperature where the natural convection were depressed due it your magnitude in relation to forced convection.

A comparison of the results of the *h* heat transfer coefficient, given by *e*mpirical correlation *Nussel,.* with the experimental data showed a 70% to 80% deviation in relation to fin infinite model ; 10% to 21% in relation to the adiabatic model ,whereas the convective model showed a 8% to 20% deviation.

An analysis of the theoretical results of fin efficiency showed a 65% to 70% deviation in relation to the infinite model; 2% to 4% in relation to the adiabatic model, and the convective model also showed a 2% to 4% deviation.

Table 3. Heat Transfer Coefficient and Efficiency.

EXPERIMENTS	**Fin's Model**	Heat Transfer Coefficient [w/m².K]		Efficiency [%]	
		Experimental	*Nusselt's* *C*orrelation	Experimental	Theoretical
1	Infinite	15.13	51.86	243	83.93
	Adiabatic	68.62		80.40	
	Convective	67.10		80.32	
2	Infinite	10.89	51.87	286.36	83.93
	Adiabatic	58.06		82.79	
	Convective	56.82		82.71	
3	Infinite	13.29	53.51	256.33	83.53
	Adiabatic	64.33		81.36	
	Convective	62.93		81.28	
4	Infinite	13.56	53.61	256.72	83.51
	Adiabatic	65.21		81.16	
	Convective	63.84		81.07	
5	Infinite	11.40	53.51	280.01	83.53
	Adiabatic	59.09		82.56	
	Convective	57.79		82.49	
6	Infinite	11.96	53.54	273.40	83.52
	Adiabatic	61.42		82.02	
	Convective	60.09		81.94	

Table 4 presents the results obtained from the heat transfer rate of the experimental heat sink and the respective theoretical values, obtained from the equations above. A comparison of the theoretical results with the experimental ones showed a 26% to 32% deviation in relation to the infinite model; 8% to 14% in relation to the adiabatic model, whereas the convective model showed a 6% to 13% deviation.

Table 4 Experimental and Theoretical Heat Transfer Rate

Heat Transfer Rate (**q**) [W]				
EXPERIMENTS	Experimental			Theoretical
	Infinite	Adiabatic	Convective	
1	43.13	68.33	66.78	53.75
2	36.72	59.86	58.53	54.14
3	38,51	61.79	60.36	52.64
4	40.45	64.95	67.95	58.63
5	35.60	57.53	56.22	52.66
6	37.92	61.77	60.42	54.80

A comparative analysis between the maximum power dissipated by the processor of 1.67 GHz PC, with a 224 MB RAM, 62 W, with the experimental values obtained for the convective model, showed a 8% to 10% deviation.

The thermal resistance results, the relation *ºC/W*, showed lower values to a temperature of 19 ºC, once the increase of the temperature by dissipated power in the form of heat exchange decreases around 20%.

5. CONCLUSIONS

The experimental results showed that the best fin model for the heat sink used in 1.67 Ghz microcomputers with a 224 MB RAM was the convective one. The efficiency of thermal exchange of the fin presented an average value of 82%, whereas the heat transfer rate presented an average value of 61.71 W, according to theoretical values. The heat sink analysed showed to be more efficient when submitted to a 19ºC room temperature. The results obtained from performance analysis, in terms of thermal resistance, i.e. the relation between temperature increase and dissipated power as heat, showed a lower value to a temperature of 19 ºC and a 20% increase.

6. ACKNOWLEDGEMENTS

Instituto Federal de Ciência e Tecnologia do Maranhão – IFM; Campus Monte Castelo
Departamento de Mecânica e Materiais - DMM
Valéria Regina da Silva, Bibliotecária CRB/13-523/06.

7. REFERENCES

Haper, W. B.; Brown, D. R. **Mathematical Equations for Heat Condction in the Fins of Air** – cooled Engines. NACA**.** Rep. 158, 1922.

Harahap, F.; Rudianto, E. **Measurements of Steady-State Heat Dissipation From Miniaturized Horizontally Based Straight Rectangular Fin Arrays,** Heat and Mass Transfer, v.41, pp. 280-288. 2005.

Homan, J. P. **Heat Transfer.** São Paulo: Mc Graw Hill. Kogakusha (2005).

Incropera; Dewitt, D.P. **Fundamentals of Heat and Mass Transfer.** 5 ed**.** [s.l.]: John Wiley & Sons, 2001.

Souza, Valter Cesar de. **Dissipadores térmicos de placas paralelas com influxo de topo.** Campinas, SP: [s.n.], 2005, Tese (Doutorado) - Universidade Estadual de Campinas, Faculdade de Engenharia Mecânica.

Zografos, A. I.; Sunderland, J. E., **Natural Convection From Pin Fin Arrays**, Exp. Thermal and Fluid Sci**.**, v.3, pp 440-449. 1990**.**

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