USE OF VAPOR CHAMBER ON SMALL HEAT SOURCES TO ELIMINATE HOT SPOTS UNDER FIN HEAT SINKS

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Abstract. *The growing challenge in the area of electronic components demands development of mechanisms to eliminate hot spots from electronic components. These hot spots are generated by high heat fluxes originating from the large amount of trails and high processing of microprocessors. . As these surfaces are usually small in some applications, even materials which are good thermal drivers such as aluminum and copper are not capable of dissipating the heat generated by the processor. Consequently, the component is damaged by high temperature levels. The technological development of computational equipments As these surfaces are usually small in some , has allowed the increase in the processing speed, and the reduction of its size, turning them faster and portable. These specific equipments generate large amount of heat in small areas. The present work analyses the increase of the heat sink efficiency by using a vapor chamber with a wick structure. A finned vapor chamber heat sink with dimensions 120 mm x 109 mm x 70 mm was built and tested with filling ratios ranging from 10% to 40% of the vapor chamber volume and heat power input ranging from 25W to 200 W. According to experiments, the filling ratio of 30% leads to the smallest vapor chamber thermal resistance of 0,21ºC/W at 200W. For comparison purposes, a conventional heat sink was also tested and presented 0,24ºC/W under these conditions, which corresponds in a decrease of 12,5% in the heat sink total thermal resistance.*

Keywords : *vapor chamber, heat sink, heat spreader, filling ratio, thermal resistance*

1. INTRODUCTION

During the last years, the development and the ascension of the microelectronics has leaded to a significant increase in the amount of heat generated by electronic components. More power in thermelectric devices with a larger capacity of cooling, microprocessors with faster capacity of information processing and smaller size demands for improvements in electronics cooling. The conventional passive systems have been inefficient to dissipate concentrated heat sources. That is explained by the fact that metals used in heat sinks have a limit in the capacity of heat transfer, which leads to hot spots. The vapor chamber works as a heat spreader of the heat sink and leads to an uniform temperature distribution over the whole heat sink base. With a better temperature control, one can optimize design parameters, such as reduction of the heat sink size, increase of heat flux, construction of lighter and cheaper heat sink systems and processors with capacity of larger processing.

Previous researches have already studied vapor chamber heat sinks, like Koito[1] that developed a vapor chamber with columns of sinterized copper through them where the steam circulated. Astrain [2] developed a termosyphon to be coupled to heat sinks to increase the efficiency of refrigerators with thermelectric systems, and Nguyen [3] concluded that the performance of the heat sink with vapor chamber is approximately 25% to 45% better than a conventional heat sink with copper and aluminum heat spreaders.The main objective of this work is to build and a vapor chamber heat sink in order to demonstrate its smaller total thermal resistance in comparison with the conventional approach. Also, a simple model with correlations available in the literature is employed to assess the thermal ressitance of the system. This work presents an experimental study of an aluminum vapor chamber built to spread heat over the base of a heat sink. The heat input was provided by an electrical heater, and the tests were performed with several filling ratios and heat power inputs. A conventional heat sink with the same external dimensions as the vapor chamber heat sink was also tested for comparison purposes.

2. HEAT SINK AND VAPOR CHAMBER

2.1 Heat Sink and Vapor Chamber

Fig. 1 shows the vapor chamber heat sink used in this study. It was built from a commercial heat sink designed to dissipate more than 200 watts. The heat sink base was taken off by machining and a box of aluminum of 120 mm x 109 mm and 9,7 mm of thickness was welded in the place of the base. Inside the aluminum box, a stainless steel screen was

welded as a wick structure. In order to obtain good vacuum, silicon rubber was put in some parts to avoid the entrance of non condesable gases. The working fluid is distillated water with several filling ratios employed.

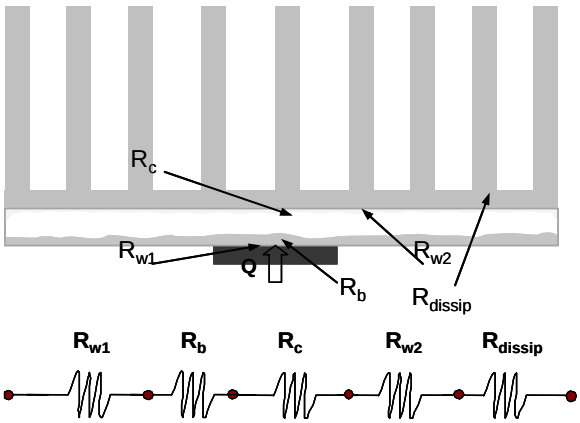


Figure 1. Vapor chamber heat sink equivalent electric circuit

As already mentioned, a conventional heat sink was tested for comparison purposes. It is basically the same as the commercial heat sink used in the vapor chamber heat sink. The only difference is the solid base instead of the vapor chamber.

2.2 Vapor chamber heat sink analytical model

An analytical model based on the method of equivalent thermal resistances was developed to estimate the approximate global heat transfer coefficient of the vapor chamber heat sink. Figure 1 presents the equivalent electric circuit for this problem. The total thermal resistance will be composed by the sum of several resistances, expressed by the following equation:

$$R_{TOT} = R_{W1} + R_b + R_c + R_{W2} + R_{dissip}$$

where,

$$R_{w1} = \frac{e}{kA_b}; \qquad R_b = \frac{1}{h_b \cdot A_b}; \qquad R_c = \frac{1}{h_c \cdot A_c}$$

$$R_{w2} = \frac{e}{kA_c}$$

R_{w1} = vapor chamber wall conduction thermal resistance (K/W)
R_b = boiling resistance (K/W)
R_c = condensation resistance (K/W)
R_{dissip} = fins thermal resistance (K/W)
R_{TOT} = total vapor chamber heat sink thermal resistance (K/W)
e = vapor chamber wall thickness (m)
k = thermal conductivity (W/m K)
h_b = boiling heat transfer coefficient (W/m^{2}K)
h_c = condensation heat transfer coefficient (W/ m^{2}K)
A_b = boiling area (m^2)
A_c = condensation area (m^2)

The boiling heat transfer coefficient used is presented by Zuber [4]:

$$h_b = \frac{0.00122 \Delta T_S^{0.24} \Delta P_{sat}^{0.75} c_{pl}^{0.45} \rho_l^{0.49} k_l^{0.79}}{\sigma_{\sup}^{0.5} h_{\lg}^{0.24} \mu_l^{0.29} \rho_g^{0.24}}$$

$$A_b = a \cdot b$$

ΔT_S = difference between the wall and saturation temperature (K)
ΔP_{sat} = increase of the ΔT_{sat} corresponding pressure (Pa)
c_{pl} = constant pressure specific heat (J/kg K)
ρ_l = density liquid (kg/m^3)
k_l = liquid thermal conductivity (W/m K)
$\sigma_{\sup}$ = surface tension (N/m)
$h_{\lg}$ = latent heat of vaporization (J/kg)
μ_l = dynamic viscosity (Pa s)
ρ_g = density gas (kg/m^3)
a = vapor chamber inner width (m)
b = vapor chamber inner length (m)

For laminar film condensation on the underside of a horizontal surface, the following heat transfer coefficient correlation is used [5]:

$$h_c = Nu \cdot k \left(\frac{g(\rho_l - \rho_v)\cos\theta}{\sigma_{\sup}} \right)^{1/2} \qquad (2)$$

$$Ra = \frac{g\cos\theta\rho_l(\rho_l - \rho_v)h_{fg}}{k\mu_l \Delta T}\left[\frac{\sigma_{\sup}}{g(\rho_l - \rho v)\cos\theta}\right]^{1/2}$$

$$\overline{Nu} = 0.81(Ra)^{0.193} \quad ; \quad 10^{10} > Ra > 10^8$$
$$\overline{Nu} = 0.69(Ra)^{0.20} \quad ; \quad 10^{8} > Ra > 10^6$$

$$A_c = a \cdot b$$

Nu = Nusselt number
Pr = Prandtl number
θ = inclination angle with the horizontal plan
g = gravity acceleration (m/s^2)

For the calculation of fin assembly thermal resistance, the following equation is employed [6]:

$$R_{dissip} = \left\{ \left[1 - \frac{NA_a}{A_t}(1-\eta_a) \right] \cdot h \cdot A_t \right\}^{-1}$$

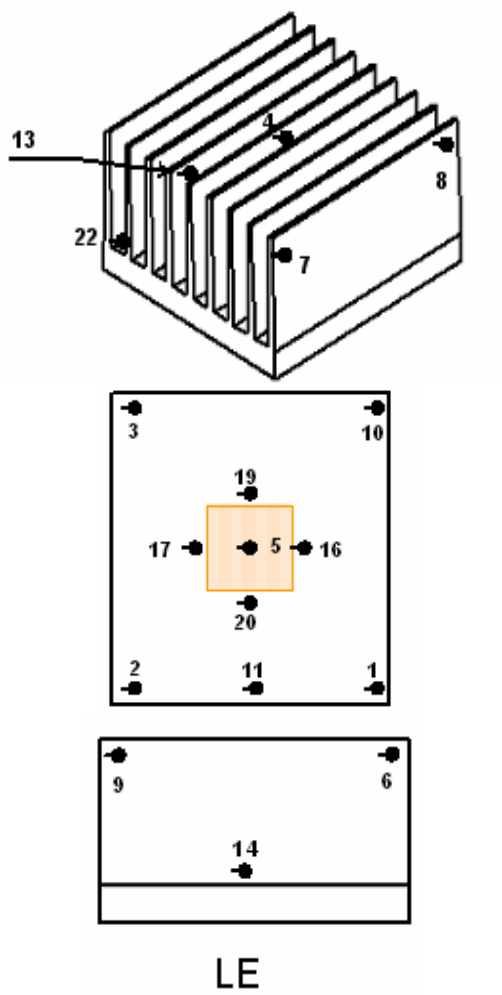
$$A_t = N \cdot A_a \cdot A_b$$
$$P = 2w + 2t$$
$$A_c = wt$$
$$A_a = 2wL_c$$
$$L_c = L + (t/2)$$
$$A_p = t \cdot L$$
$$\eta_a = \frac{\tanh mL_c}{mL_c} \quad Where \quad m^2 = \frac{hP}{kA_c}$$

N = number of fins
A_a = fin area (m²)
A_t = total area (m²)
A_b = exposed base area (m²)
h = convection coefficient (W/m²K)

η_a = fin effectiveness
p = fin perimeter (m)
w = fin width (m)
t = fin thickness (m)
L_c = fin length (m)

3. EXPERIMENTAL SETUP

The experimental apparatus consists of a aluminium vacuum chamber heat sink with dimensions 120 mm x 109 mm and 9,7 mm with nine 60 mm long fins. The thickness of the fins is 4.6 mm, and the width is 120 mm. Twenty K type thermocouples were distributed according to Fig. 2. A conventional heat sink with the same dimensions as the vacuum chamber heat sink was also tested in order to have a comparative benchmark. In the base of the apparatus, a rigid copper heater of 40 mm x 40 mm was placed to emulate the heat of an electronic component. The rest of the base was isolated so all the heat goes towards the vapor chamber and the fins. A commercial 12VDC fan was put to provide forced convection on the surface of the fins. The filling ratios tested were 10%, 20%, 30% and 40% of the chamber internal volume. The total heat transfer ranged from 25 to 200 W. The objective is to study the influence of the filling ratio and the heat power input on the thermal resistance and temperature distribution of the heat sink.



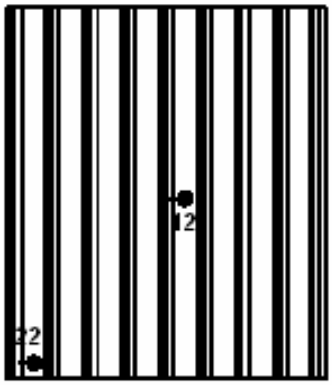


Figure 2. Thermocouple distribution

4. RESULTS AND DISCUSSION

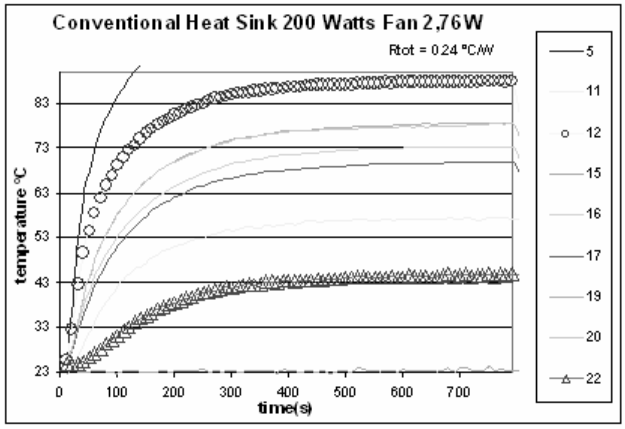
The total thermal resistance of the heat sinks were computed under steady state as the difference between the average of thermocouples 16, 17, 19 and 20 (see Fig. 2) and the environment air. The best result, i.e. the smallest resistance, was found for a filling ratio of 30% with heat flux of 200 W. The total thermal resistance of the vapor chamber heat sink was found to be 0,21 K/W under these conditions. The conventional heat sink presented a constant value of 0,24 K/W, which means the vapor chamber reduced the heat sink total thermal resistance by approximately 12.5%.

It was also observed that the vapor chamber thermal resistance decreases with the increase of the heat input. For heat input smaller than 200 W, the thermal resistance of the vapor chamber heat sink is larger than the conventional heat sink, which demonstrates that the vapor chamber is especially useful for large heat input levels. Unfortunately, the experimental set-up did not supported a heat power input larger than 200 W, otherwise the difference between the vapor chamber and conventional heat sinks would be larger.

The graph of Fig. 3 shows temperature readings as a function of time for both the conventional heat sink and vapor chamber heat sink. The thermocouple numbers appearing in Fig.3 correspond to those indicated in Fig.2. Thermocouples 12 and 22 are highlighted in Fig.3 to demonstrate the effect of the vapor chamber in homogenizing the temperature distribution. Thermocouple 12 is located at the center of heat sink and thermocouple 22 is placed at the corner of the heat sink. When the conventional heat sink dissipates 200 W, a temperature difference of 45ºC exists between these thermocouples. When the same measurement is made with the heat sink coupled with vapor chamber, that gap reduces drastically to 4ºC showing the homogeneity of temperature of the heat sink with vapor chamber.

Figure 4 shows pictures taken by a thermographic camera with the objective of visualizing the temperature distribution in both the conventional heat sink and in the vapor chamber heat sink. As one can see, the temperature distribution is much more homogeneous for the heat sink with vapor chamber while the temperature is concentrated on the central fins in the heat sink without the vapor chamber.

Under the conditions that gave the smallest thermal resistance, i.e. 200 W of heat input, the theoretical model yielded 0.29 K/W, against 0.21 K/W measured. The model overpredicts the data in 38%, which can be considered reasonable considering the fact that the heat transfer coefficient between the fins and the air was obtained from the correlation for turbulent flow over a single flat plate. The air velocity leaving the fan was measured without the heat sink attached to it, which obviously introduces errors to the process.



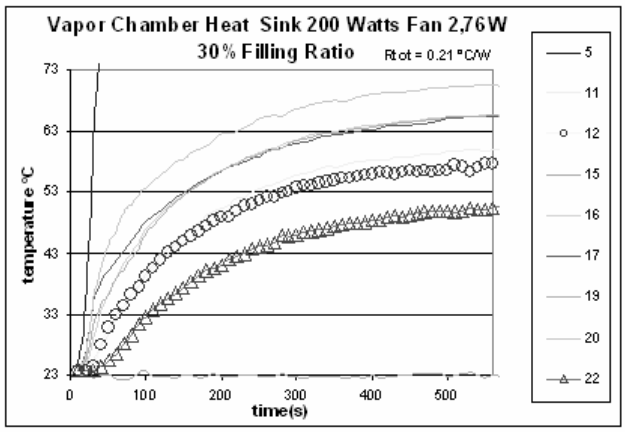
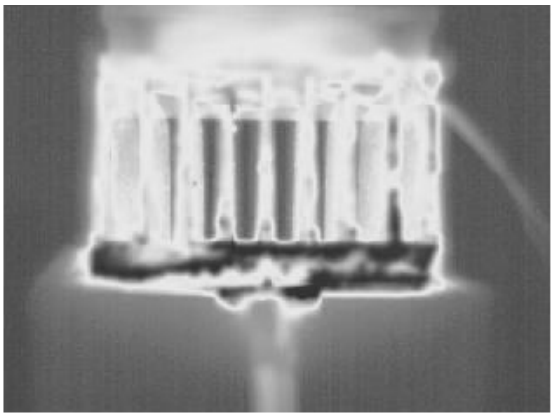


Figure 3. Temperature distributions for the conventional and the vapor chamber heat sinks

5. CONCLUSION

The main objective of this work was to build and test a vapor chamber heat sink in order to demonstrate that the use of the vapor chamber concept leads to a decrease of the heat sink total resistance and to a more uniform temperature distribution on the heat sink base. Also, a simple model was developed based on the equivalent electric circuit method and simple correlations available in the literature.

To heat sinks were tested in this work: a conventional heat sink and a vapor chamber heat sink. As expected, the use of vapor chamber allowed a homogeneous distribution of temperature for the whole base of the heat sink. The thermal resistance of the vapor chamber is also smaller than in the conventional heat sink with the same external dimensions. The conventional heat sink tested was obtained commercially and was designed to dissipate 200W. For this heat power level, the thermal resistance of the heat sink with vapor chamber was 12.5% smaller. This difference should increase if larger heat power inputs were tested. At this power level, the theoretical model predicts the experimental data within 38%, showing that, despite its simplicity, the model can make a reasonable estimation of the system overall resistance.



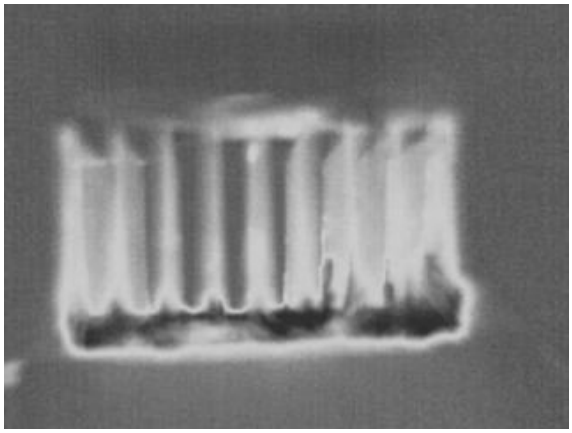


Figure 4. Temperature distribution on the conventional heat sink and vapor chamber heat sink.

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7. RESPONSIBILITY NOTICE

The authors Alexandre S. Oliveira, Márcia B.H. Mantelli and Fernando Milanez are the only responsible for the printed material included in this paper.