Thermal Characteristics of Graphite Foam Thermosyphon for Electronics Cooling

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Graphite foams consist of a network of interconnected graphite ligaments and are beginning to be applied to thermal management of electronics. The thermal conductivity of the bulk graphite foam is similar to aluminum, but graphite foam has one-fifth the density of aluminum. This combination of high thermal conductivity and low density results in a specific thermal conductivity about five times higher than that of aluminum, allowing heat to rapidly propagate into the foam. This heat is spread out over the very large surface area within the foam, enabling large amounts of energy to be transferred with relatively low temperature difference. For the purpose of graphite foam thermosyphon design in electronics cooling, various effects such as graphite foam geometry, sub-cooling, working fluid effect, and liquid level were investigated in this study. The best thermal performance was achieved with the large graphite foam, working fluid with the lowest boiling point, a liquid level with the exact height of the graphite foam, and at the lowest sub-cooling temperature.

Key Words: Graphite Foam, Electronics Cooling, Thermosyphon

Nomenclature

CTE : Coefficient of thermal expansion [ppm/℃]
h : Height of graphite foam [m]
k : Thermal conductivity [W/m-K]
k_s : Specific thermal conductivity [(W/m-K)/(g/cm^3)]
q : Heat dissipation at heating wall [W]
q'' : Heat flux at heating wall [W/cm^2]
R : Thermal resistance [℃/W]
T : Temperature [℃]
dt/dz : Temperature gradient [℃/m]
Δ : Uncertainity of a given parameter [%]

Subscripts

cu : Copper
s : Sub-cooling or specific
surr : Surrounding fluid
w : Heating wall

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1. Introduction

For several years, the heat flux of microprocessor has rapidly been increasing due to increased clock speed and numbers of transistors. Therefore, the thermal management of modern microprocessor has been a critical issue in computer systems. The Intel Pentium III processor has 10~18 W/cm^2 in heat flux and new Intel Pentium 4 processor must dissipate up to 20~34 W/cm^2. These processors use as cooling devices fan and heat sinks in desktops and heat pipes in laptops. High performance next-generation processors will increase to about 40~60 W/cm^2. As candidate for cooling these high performance next-generation processors are thermosyphons. These consist of an evaporator, a condenser, and a transfer section. Working fluid evaporates at the evaporator, the vapor rises through the transfer section, and condenses at the condenser releasing its latent heat. Finally, gravity returns the condensate back to the evaporator and the process repeats. Because thermosyphon use phase change heat transfer, it can transfer a large quantities of heat from the evaporator at low temperatures. Moreover, thermosyphons are passive and do not require cooling fan. Due to these advantages, the use of thermosyphon may be attractive for cooling high performance processors.

Jiang et al.(2001) studied the boiling heat transfer on machined porous surfaces distributed with micropores on channels. The numerical modeling which could predict the optimum microstructure dimensions was developed and verified by experimental data in the study. Wu et al. (2002) had performed the boiling heat transfer on porous coated surfaces with vapor channels. The optimum vapor channel density and size were determined and a simplified one-dimensional formulation for two-phase flow with a heterogeneous structure was proposed. Ramaswamy et al.(2000) investigated the thermal performance of their thermosyphon for electronics cooling. They employed a microstructure consisting of a six-layer stack of copper plates, which were grooved with rectangular microchannels, on the evaporator and have had heat transfer rates up to 100 W/cm^2 with a wall superheat of 27.8℃ in FC-72. Ramaswamy et al.(2001) reported that the performance of partial vacuum was more efficient than the high pressure. Mudawar and Anderson (1989) examined the parametric study of the effects of fluorocarbon coolants (FC-72 and FC-87), pressurization, sub-cooling, enhanced evaporator surfaces (pyramid, square, and diamond studs) and so on. Mudawar and Anderson (1989) achieved a heat flux of about 105 W/cm^2 while maintaining the surface temperature below 85℃.

Another potentially enhanced design for the evaporator is one made from pitch-derived graphite foam. The graphite foam has high thermal conductivity, a low density, and a coefficient of thermal expansion that is close to that of silicon. Klett et al.(2000) have developed a new and less time consuming process for fabricating pitch-based graphite foam without the traditional blowing and stabilization steps. They also examined

Table 1 Thermal properties for various materials (Ramaswamy et al., 2000 ; 2001)

Material	Specific thermal conductivity, k_s* ((W/m-K)/(g/cm^3))	CTE (μm/m ℃)
Graphite foam	300	2～3
Copper (pure)	45	17
Aluminum 6061	64	23
Aluminum foam	24	-
Silicon chip	-	2.6
Aluminum nitride baseplate	-	3.3

*Defined as thermal conductivity divided by density

thermal conductivity of the Mitsubishi ARA drived graphite form and Conoco drived graphite form, and reported the thermal conductivity was higher at graphitization at 4℃/min than at 10℃/min. The comparisons of bulk specific thermal conductivity and coefficient of thermal expansion (CTE) for some materials are shown in Table 1. The graphite foam has a remarkably high specific thermal conductivity of 300 (W/m-K)/(g/cm^3), which is roughly five times higher than aluminum and six times higher than copper. The coefficient of thermal expansion of the foam may also offer a benefit for electronics applications because the foam CTE is much closer matched to the silicon chip and typical aluminum nitride baseplate of electronics component. This may help to reduce thermal stresses attributed to high mismatches in CTE and thereby, improving reliability.

In this study, various parameters such as two geometries of graphite foam (18 mm × 13 mm × 13 mm and 18 mm × 30 mm × 30 mm), two working fluids (PF-5060 and PF-5050), various liquid levels (2.25 mm～27.0 mm), and various condensing temperatures (20.0℃～45.3℃) were tested to determine the thermal performance of a graphite foam thermosyphon evaporator. The purpose of present study is to investigate the effects of these parameters. The thermal performances of the parameters were determined by generating boiling curves for the nucleate boiling regime.

2. Experimental Setup and Method

A half section view for the graphite thermosyphon pool boiling experimental rig used in present study is shown in Fig. 1. In the figure, the aluminum housing (108 mm high and 203 mm in maximum diameter) contains a heating block made from oxygen free high conductivity copper. The heating block is held in place using G10 insulation to minimize lateral heat loss. A cartridge heater installed in the heating block can provide a maximum of 500 W. A DC power supply (HP 6675) connected to the heater was controlled by a Visual C++ program. The power was increased by 2 or 4 W and the temperatures along the column of the heater block (1.0 cm^2 cross sectional area) was measured by 4 sets of type T thermocouples (1.27 mm diameter) spaced 6.35 mm apart. Two pieces of graphite foams (18 mm high × 13 mm wide × 13 mm deep, and 18 mm high × 30 mm wide × 30 mm deep) were brazed onto the copper using S-Bond™ (2002) process of Materials Resources International providing a highly thermally conductive interface. The properties of graphite foam used in this experiment are summarized in Table 2. Two kinds of the working fluid (PF-5060 and PF-5050) were used in present study. The working fluid was poured into a plexiglass chamber (102 mm high and 152 mm in diameter) at seven liquid levels of 1h/8, 2h/8, 4h/8, 6h/8, 1h, 10h/8, and 12h/8. Where, ‘h’ means the height of graphite foam. The vapor of the working fluid evaporated after heating was condensed using a water cooled condensing coil. The cooling water was set at four temperatures (T_s=20.0, 26.5, 34.6 and 45.3℃). The top of chamber was sealed with a lid, which has cooling water inlet, outlet, pressure transducer, and type T thermocouple probe. The transducer and the probe were used to measure the pressure in the chamber and the bulk temperature of surrounding working fluid in the chamber, respectively. Data from the transducer, the probe, and thermocouples were obtained using a Fluke Hydra and PC every 3 s throughout the experiment.

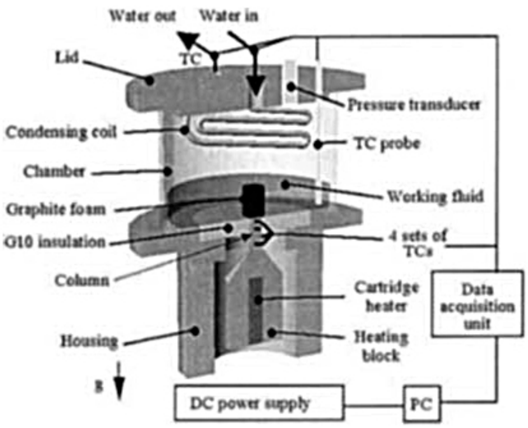

Fig. 1 Experimental setup

At the experiments commenced, a test chamber leak test with pressurized N_2 gas was performed to make sure there were no leaks. After the chamber was filled with liquid, the liquid was degassed by pulling a vacuum until the pressure in the chamber decreased to the pressure corresponding to the saturated vapor pressure at the liquid temperature.

The heat flux to the graphite foam was determined by measuring the temperature gradient, dT/dx, in the neck of the column using a least-squares fit of the temperature measurements. The heat flux at the heating wall, q'' calculated from the measurements was about 70% compared to the electric power of DC power supply. The temperature at the heating wall, Tw, was determined

Table 2 PocoFoam properties (Viswanath et al., 2000)

Property	Value
Pore Diameter (Average)	350 μm
Specific Area	>4 m^2/g
Open Porosity	>96%
Total Porosity	73~82%
Density	0.2-0.6 g/cm^3
Thermal Conductivity	100~150 W/m-K
Specific Heat	0.70 J/g-K
Thermal Diffusivity	3.71 cm^3/s
Coefficient of Thermal Expansion	2-3 μm/m-K
Compressive Strength (when density=0.5 g/cm^3)	2.07 MPa

by extrapolating the temperature profile to the wall. The criterion for steady state was that the average temperature within the neck of heater block changed by less than 0.1 K over a 60-second period. The time required to reach new steady state was taken about 15-20 minutes after increasing the power at the previous steady state, and total time to obtain the full boiling curve for an experimental run was taken about 4~6 hours.

3. Uncertainty Analysis

The standard deviations in the heat flux results were calculated using the sum of squares method :

$$\Delta q''=\sqrt{\left(\frac{dT}{dz}\Delta k\right)^2+\left(k\Delta\frac{dT}{dz}\right)^2} \quad (1)$$

The copper rod used for the heater block was an OFHC copper of unknown origin. A constant thermal conductivity of 3.867 W/cm-K was assumed and was calculated by determining the average temperature dependent conductivity over the range of temperatures seen in this study using the temperature dependent relationship given by Kedzierski (2002):

$$k_{cu}=78.113e^{74.924/T}(\ln T)^{0.7842} \quad (2)$$

The standard deviation associated with this constant thermal conductivity was estimated to be 3.2% based on the uncertainty of the model and the range of temperatures. The standard deviation of the temperature gradient was calculated from the uncertainty of the linear regression. The standard deviation of the temperature gradient was calculated at every time step and in general was found to be larger at higher temperature gradients. The standard deviations were combined according to Eq. (1), and the standard deviation of the heat flux was determined at every time step. The maximum standard deviation of the steady-state heat flux was found to be 3.6 W/cm², which occurred at the highest heat flux. The minimum standard deviation occurred at the lowest heat flux and was found to be 0.4 W/cm². The average standard deviation of the heat flux was 2.2 W/cm². The standard deviation in the temperature measurements was estimated to be 1.35℃ based on standard accuracies of type T thermocouples and the specifications of the Hydra DAQ Unit. Uncertainties for the specified conditions were also estimated. The standard deviations in the cooling water temperatures were found to be 3.4℃ for the low temperature and 1.3℃ for the high temperature based on the accuracy of the thermocouple measurement and direct inspection of the measured data. The standard deviation in the liquid level was estimated to be 0.5 mm. The standard deviation in the geometry of the graphite foam sample was estimated to be 0.2 mm.

4. Results and Discussion

4.1 Graphite foam effect

Using the graphite foam in thermosyphon, how much will the thermal performance be more increased than no graphite foam? This could be first interesting question. The variations in heat transfer with wall temperature for large graphite foam, small graphite foam and no graphite foam are compared in Figure 2 for a liquid level of 12h/8, a condenser temperature of T_c=20℃ and working fluid of PF5050. For a wall temperature of T_w=85℃, q''_{85} are 15.7 W/cm² for the case of no graphite foam, 49.2 W/cm² for the case of small graphite foam, and 113.3 W/cm² for the case of large graphite foam, respectively. An evaporator

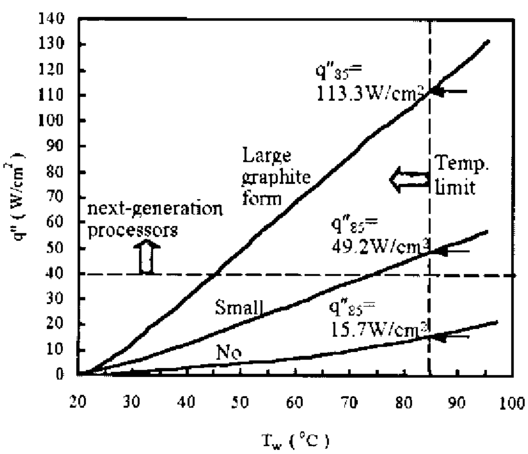

Fig. 2 Graphite foam effect on thermal performance for three kinds of graphite foam geometry

with the graphite foam attached is seen to transfer more than three times the heat transfer from the bare heater surface for the small graphite foam and seven times for the large graphite foam. Specially, the large graphite foam is very effective material in the pool boiling thermosyphon because it can fully manage high performance next-generation processors with 40-60 W/cm² of heat flux for keeping below 85℃, as seen in Fig. 2.

4.2 Condenser temperature effect

The vapor of working fluid after boiling condenses into liquid phase at the condensing coil. Therefore, sub-cooling temperature, T_s certainly has an effect on thermal performance of graphite thermosyphon. The effect of condenser temperature, T_c on the thermal performance is shown on Fig. 3. The experimental conditions were under a liquid level of 12h/8 and working fluid of PF5050. For the case of small graphite foam, as the condenser temperature, T_c increases from 20.0℃ to 45.3℃, the heat flux decrease by up to 28.4% in the range of T_w=60-95℃. The condenser temperature has a smaller effect at lower wall temperatures. For the case of large graphite foam, increasing T_c from 20.0℃ to 45.3℃ results in a decrease in heat flux by about 38.5%, significantly more than for the small foam case.

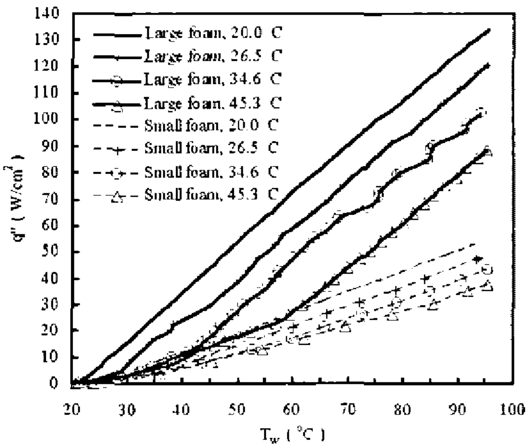

Fig. 3 Sub-cooling effect on thermal performance between large and small graphite foams

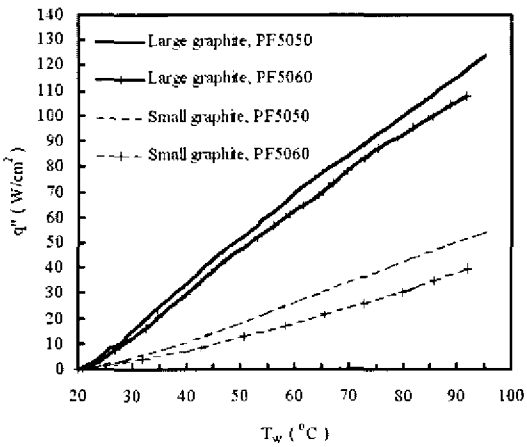

Fig. 4 Working fluid effect on thermal performance

4.3 Working fluid effect

Two working fluids, PF5060 (2002a) and PF5050 (2002b) with atmospheric pressure boiling points of 30℃ and 56℃, respectively were tested. Fig. 4 shows the boiling curves of the large and small graphite foams with two working fluids. Experimental conditions were for a liquid level of 4h/8 and T_s=20℃. The liquids were degassed prior to testing. For the case of small graphite foam, the thermal performance of PF5060 decreases about 30% than that of PF5050 in the range of T_w=60-95℃. For the case of large graphite foam, the thermal performance of PF5050 is still better, as well. The thermal performances of PF5060 decrease about 15% than that of PF5050 in the same range above. Thus, it could be decided that the working fluid with the lowest boiling point gives the greatest thermal performance.

4.4 Liquid level effect

At which liquid level of working fluid can the graphite foam thermosyphon show the best thermal performance? This was another interesting question in present study. In fact, the information for optimum liquid level must be very important factor at electronics cooling design of the graphite foam thermosyphon. Fig. 5 shows the effect of liquid level on graphite thermosyphon. Data were obtained at T_c=20℃ using PF5050. For the small graphite foam, as the liquid level increase from 1h/8 to 12h/8, the heat flux increases about 48% between T_w=60-95℃. This result indicates that

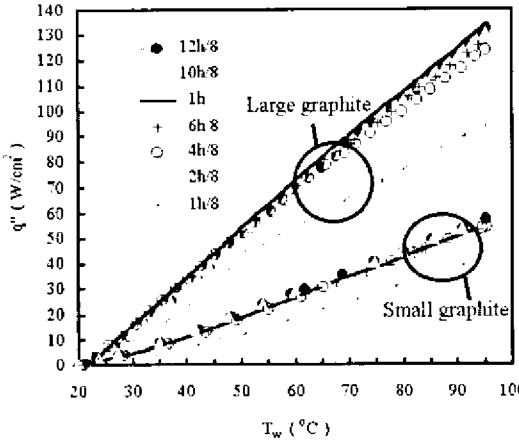

Fig. 5 Liquid level effect on thermal performance using PF5050

higher liquid level is more effective because of increasing wetted surface in the graphite foam. However, the thermal performance in the range of 4h/8-1h is almost the same because all surfaces in small graphite foam are fully wetted. For the large graphite foam, as the liquid level increases from 1h/8 to 12h/8, the heat flux increases about 34% in the range of T_w=60-95℃. Surprisingly, it has been found that 1h, exact height of graphite foam, is optimum liquid level in large graphite foam. This may be due to liquid impeding the outflow of vapor from the graphite foam when the liquid layer is higher than the foam height.

5. Conclusions

The effect of various parameters on a graphite foam based evaporator performance enhancement purpose with application to electronics cooling was investigated. A summary of the results is given below.

(1) The thermal performance of large graphite foam is about 2.3 times greater than that of small graphite foam and 7 times greater than no foam. The large graphite foam can be very effective as an evaporator because it can fully manage the thermal load of high performance next-generation processors (40-60 W/cm^2) if the condenser temperature is kept low enough.

(2) The condenser temperature significantly impacts the performance of the thermosyphon. As the condenser temperature increases from 20.0℃ to 45.3℃, the heat fluxes decrease by about 28% for the small graphite foam, and about 39% for the large graphite foam in the range of T_w=60-95℃. As Ts increase, the heat transfer decreases more for the large graphite foam.

(3) The working fluid, PF5050 gives better thermal performance because of the lowest boiling point.

(4) Liquid levels close to the height of the foam were found to produce the maximum heat transfer. All of the foam was not wetted when liquid levels were significantly lower than the foam height. Increased vapor removal resistance may be the reason for the deterioration in heat transfer when the liquid height was higher than the graphite foam height.

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