Cooling Behavior in a Novel Heat Sink Based on Multilayer Staggered Honeycomb Structure

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Abstract: A novel heatsink based on a multilayer stack of thin metal plates with staggered honeycomb cell microchannels was investigated in this paper. A series of working-parametric tests such as different heat sink pipe diameter and pumping power were conducted for the microchannel cooling system to determine the heat transfer performance under small flow rate conditions. For the double fluid flow inlets and outlets heatsink design, experimental results showed that more uniform substrate temperature distribution was obtained than the single inlet and outlet ones. It showed that the heatsink design provided a good choice for electronic chips cooling applications.

Key words: Multilayer, honeycomb, off-set fins, heatsink.

1. Introduction

The increasing demand of high performance and multifunctionality electronics chips has brought great challenges to thermal management field. Conventionally, air cooled heat sinks with fans have been the first choice for the applications of heat removal because of the advantages of low cost, easily implementation and safe enough without leakage problems. With the utilization of optimized fin designs and advanced fans technique, the cooling capacity of air cooled heat sinks can be enhanced, which made them suitable for the low and medium power dissipation applications. However, facing to the challenge of increasing power heat generation level, it is generally agreed that the current air-cooling technology is approaching limits even using larger and larger heat sinks and increasing heat sink airflow to keep the electronic components working in the safe model. This approach can lead to mechanical stress resulting from the large, heavy heat sinks and intolerable noise levels. Liquid, generally having several tens of times the thermal capability of air, has the ability of providing the high heat removal rates while maintaining the compact volumes. One of the most common configurations for indirect liquid cooling of electronic systems is heat sinks or cold plates where a liquid is forced to flow through channels embedded in a solid matrix. For hydrodynamically and thermally developed laminar duct flow, the heat transfer coefficient is inversely proportional to the hydraulic diameter. Achieving high value of the heat transfer coefficient therefore requires shrinking the channels size to microscopic dimensions. This was first proposed by Tuckerman and Pease [1], one of the experimental microchannel heat sinks which was the structure with an array of parallel microscale rectangle channels of 50 μm width, 302 μm depth and 1 cm length, separated by walls of width 50 μm, could effectively removed 790 W/m² power density with a substrate temperature rise of 71°C. Inspired by this foundational work, much research has been conducted to analyze and optimize this prospect cooling technology, as reviewed by S. V. Garimella and C. B. Sobhan [2].
When liquid flows through these long parallel heated rectangular microchannels which are fabricated by traditional etching technology in silicon or copper base, its temperature will rise as a result of the heat input along the flow direction. At the same time, the heat transfer coefficient decreases along the flow direction due to the growing boundary layer. For single-phase liquid fluid flow in microchannels, cooling capability enhancement is a result of reduced thermal boundary layer thickness and increased heat transfer area to volume ratio. Kishimoto and Sasaki [3] proposed a diamond-shaped interrupted microgrooved cooling fin to decrease the junction temperature variation across the chip. The cooling fins were arranged into a staggered construction, so that the thermal boundary layer up- and downstream from the cooling fins did not affect each other. A new thermal boundary layer was restarted along the small pieces, resulting in a thinner thermal boundary layer. Compared to the conventional microgrooved cooling fins with the parallel plate wall passage, the analytical results indicated that the junction temperature variation across the chip decreased to less than 25%. Y.P. Chen and P. Cheng [4] designed a fractal tree-like microchannel net. Their results showed that a treeshaped channel network had a strong heat transfer capability and required lower pumping power as compared to parallel straight channels. X.Q. Wang, A.S. Mujumdar and C. Yap [5] numerically investigated the thermal characteristics of tree-shaped microchannel nets for cooling of a rectangular heat sink. They found that such channel networks had certain advantages over conventional parallel channel nets such as lower and more uniform temperature distribution and better stability in case of accidental blockage in channel segments. But fabrication of such patterns is more complex and will require new fabrication technology.

For reducing the fabrication difficulty, stacking or creating multilayer of channels can be an alternative way to obtain high surface-to-volume ratio in limited space. Vafai and Zhu [6] proposed a two-layered microchannel heat sink with counter-current flow arrangement. They found that the temperature rise and pressure drop along the two-layer microchannel structure reduced compared to an equivalent single layered heat sink. X.J. Wei and Y. Joshi [7-9] analyzed a stacked microchannel heat sink integrated many layers of liquid cooled microchannels. Because the stacked microchannels provided a larger heat transfer area, significant less pumping power was found to be required to remove a certain rate of heat than a single-layered micro-channel.

In the authors’ previous works [10, 11], a novel design of periodic staggered honeycomb microchannel heat sink structure was reported. The construction of the heat sink combined the characteristics of off-set fins and multilayer channels structure with a cost-effective way. Two experimental heatsinks with different honeycomb microchannel plate size were tested integrated with various cooling system configuration. It found that the heatsink design with larger honeycomb size and thinner plate thickness had better cooling capability. However, the substrate temperature distribution is also important for evaluating the system performance and needs to be further discussed. In this paper, a new double inlet / outlet heatsink design is introduced to evaluate the substrate temperature distribution performance compared with the other two single inlet / outlet ones with different pipe diameter. All of the experimental heatsinks had the same honeycomb microchannel plate size and were tested in a long-distance miniature cooling system under small flow rate, which involving the total length of 2.3 m long stainless steel tube with $\phi$ 2 mm inner diameter as the system piping for the close loop. It also showed that the heatsink design provided a good choice for electronic chips cooling applications under low flow rate.

2. Multilayer Staggered Honeycomb Microchannel Heatsink

The prototype structure of the multilayer honeycomb
heat sink is built up by stacking several flat thin rectangle metal plates with etched honeycomb holes inside together to form the staggered fluid flow passage channels. Each of the metal plates has the same size of \(L \times W \times H\), which is etched to have a number of small regular hexagonal honeycomb cells with ex-circle diameter \(d\) and fin thickness \(t\) as showed in Fig. 1 (a) and (b). In order to keep tight connecting between two layers, there is a wide seal region \(a\) left around the perimeter of the metal plate with no honeycomb cells. By rotating one plate clockwise 180° angle and bonded with the other one, the honeycomb holes are divided by cell fins to form the serpentine fluid flow channels in the direction of normal layer plane, as showed in Fig. 2. The heat sink inner structure comes into being just by stacking the multilayered such honeycomb microchannel plates together. For application of indirect liquid cooling of electronic systems, the working liquid is forced to flow through the well-designed forming channels embedded in the cellular metal core for heat dissipation.

The dimensions of the hexagonal honeycomb cells are well controlled by metal etching method. This is the method through chemical corrosion of the unprotected metal material. In contrast to the traditional machine process, the etching method has short production circle about several days. Anything you can draw, or pattern you like can be copied onto metal. Changing dimension is easy and fast. And the most important is that the development cost is also low. A sample of fabricated honeycomb microchannel plate is showed in Fig. 3. The multilayered sample is showed in Fig. 4.

The configuration of the staggered honeycomb microchannel heat sink is illustrated in Fig. 5. The heat sink and the channels are made of brass (H62). Each heat sink is composed of fifteen honeycomb layers where the individual layers have the same dimension of 40×20×0.16 mm. Rows of regular hexagonal honeycomb cells in the flat thin brass plate also have the same dimensions of cell ex-circle diameter of 2.49 mm and fin width of 0.2 mm. Although the contact heat resistance can be minimized by all points of wire contact among metal layers, multilayers of these honeycomb plates are just bonded together by the 14 laser welding line in the side region around the perimeter of the plates to reduce the welding difficulty. The chamber which containing the honeycomb microchannel structure is bonded with cover board with inlet / outlet pipes at the top side of the channels using miniature “o- rings” for fluidic seal. In this work,
three different diameter inlet/outlet pipes and designs heat sink are tested to evaluate the cooling characteristics for the heat dissipative applications of electronic components.

3. Experimental Setup

The experimental facility and flow loop are showed in Fig. 6. The loop consists of a micropump, test module, the liquid-to-air heat exchanger and a reservoir. The total length of 2.3 m long stainless steel tubes with $\phi 2 \text{ mm}$ inner diameter are introduced as the system pipes for the close loop. The tube is 1.2 m long from the outlet of heat sink test module to the heat exchanger inlet and 1.1 m from the outlet of the heat exchanger to the heat sink inlet. Water is pumped from the reservoir through the inlet to the microchannel heatsink, and exits through the outlet to the heat exchanger. The temperatures of the water are measured at the inlet and outlet of the test module by T-type thermocouples with an accuracy of ±0.2 K. The flow rate of the water is controlled by adjusting the input voltage of the micropump and is measured by the standard weighing method as follows. Firstly, the liquid exiting the test module is accumulated in a glass beaker, then a balance with an accuracy of ±1g is used to measure the weight of the accumulated liquid. The input voltage and current of the micropump are measured with the voltage meter of ±1V accuracy and the current meter of ±0.1A accuracy respectively.

During the experiments, the substrate temperatures of the heat sink are very important evaluation parameters for the cooling system. Heat is supplied using two 80 W cartridge heaters that are installed inside a power intensifier made of copper. The input heating power is controlled by an adjustable AC voltage power with an accuracy of ±2W to provide electrical power to the device. The experimental heatsink is mounted on the power intensifier’s top surface with thermal grease to minimize the contact heat resistances. Two little holes of $\phi 1 \text{ mm}$ are drilled in the center positions, which are 7 mm left from the two sides of the top surface of the copper intensifier block to accommodate two T-Type thermocouples. Based on the good thermal conductivity of the copper material, the bottom surface (substrate) temperatures of heatsink are approximately equal to the measured ones.

All thermocouples are read into a data acquisition system (Keithley 2700), which are connected to the PC with RS232 port. The parameters used in the data reduction and analysis are summarized below: the steady state total heat transfer rate $Q$ removed by water and the heat flux $q$ at the substrate of the microchannel heat sink removed by water are given by following respectively:

$$Q = \dot{m}c_p\Delta T$$

(1)

$$q = \frac{Q}{A}$$

(2)

Where $\dot{m}$ is the total mass flow rate, $c_p$ is the specific heat of the fluid, and $\Delta T$ is the temperature change. The temperature change is determined from the measured inlet and exit temperatures, and $A$ is the area of the substrate of the microchannel heat sink.

In order to evaluate the pressure drop of the cooling
system, the power consumption of the micropump $P$ is also important parameter to be known and it is determined by the product of input voltage $V$ and the input current $I$.

$$P = VI \quad (3)$$

4. Results and Discussion

4.1 Experimental Results with Different in / out Diameter Pipe Design

Fig. 7 showed the influence of different in / outlet diameters of heat sink pipe on the system cooling performance. Three arrangements with the same heat sink structure but using different inner diameter pipes such as $\phi 2 \text{ mm}$ pipe with single inlet and outlet, $\phi 4 \text{ mm}$ pipe with single inlet and outlet and $\phi 4 \text{ mm}$ pipe with double inlets and outlets were tested under constant 120W input heating power and pumping power (DC3V, 0.25A). It could be found that the two temperature points of the measured substrate (bottom surface of the heatsink) for $\phi 4 \text{ mm}$ pipes with double inlets and outlets arrangement were nearly the same, which indicated the more uniform temperature distribution on the heat sink substrate. Compared to the other two arrangements, the minimal flow rate of 123 $\text{ml/min}$ was obtained in the $\phi 2 \text{ mm}$ tube with single inlet and outlet case under the same pumping power. In Fig. 7(a), the slight oscillation of the outlet temperature and two substrate temperatures showed that the system was not stable for this small size pipe design. The experiment test also showed that the flow rate of $\phi 4 \text{ mm}$ tube with double inlets and outlets was 136 $\text{ml/min}$, which was relatively less than 141 $\text{ml/min}$ under the case of $\phi 4 \text{ mm}$ tube with single inlet and outlet. This is attributed to the hydrodynamic head loss of separate and combinational flow in double pipes. However, more uniform water flow distribution was formed inside the heat sink structure.

4.2 Experimental Results of the Cooling Performance of $\phi 4\text{mm}$ Pipe with Double Inlets and Outlets

Fig. 7 Temperature variation of difference pipe diameter with different inlet and outlet arrangements.
For the heat sink design with $\phi 4\, mm$ pipe size and double inlets and outlets, its cooling performance was tested under 150W input heating power. The results were showed in Fig. 8. In this case, flow rate was 115 ml/min and power consuming was 0.72W (DC3V, 0.24A), the environment temperature was 17.8°C, the inlet water fluid temperature was 18.9°C, outlet water temperature was 36.6°C, and the two substrate temperatures of the heatsink were 61.4°C and 61.2°C respectively. According to (1) and (2), the total heat transfer rate $Q$ was 142W, and the heat flux $q$ at the substrate of the microchannel heatsink removed by water was 17.7 W/cm². It was also noted from Fig.8 that the measured temperatures decreased sharply when stopped heating.

Fig. 8  Temperature variation of double in / out pipes heatsink design under 150W input heating power.

Fig. 9 showed the variations of the measured heatsink temperature with time at different pumping power. During the experiments, the input heating power was kept to 80W and environment temperature was 21°C. The electric voltage of micropump was firstly set to DC2.5V for steady state, and then increased to DC3V, until to DC4V. It could be found that with the pumping power increasing, the substrate temperatures decreased correspondingly. When the pumping power of micropump was 0.5W (DC2.5V, 0.2A), corresponding flow rate changed as 85 ml/min, the substrate temperatures were 55.2°C and 55.3°C. When the pumping power increased to 1.24W (DC4V, 0.31A), corresponding flow rate was 144 ml/min, the substrate temperatures were decreased to 49.5°C and 49.4°C. It was easy to explain the phenomena. When pumping power increased, more water fluid was pushed through the heat sink. So the heat transfer coefficient of the multilayered honeycomb microchannel heat sink would increased, thus the heat from power intensifier would be taken out more efficaciously. However, it should be notified that with the pumping power increased, the micropump would consume more power, which would increase the operation cost. In real application, there should be some tradeoff in design between the heat transfer efficiency and the power consumption.

5. Conclusions

For the purposes of interrupting the development of thermal boundary layer and reducing the fabrication difficulty, a novel multilayer staggered heat sink was investigated. By stacking multilayered flat thin rectangular plates with a number of regular honeycomb cells etched inside, the well designed staggered fluid flow channels were formed to enhance heat transfer. In order to evaluate the heatsink substrate temperature distribution, three experimental heat sinks with different pipe diameter and design were tested in a long-distance miniature cooling system to determine the heat transfer performance. For the cooling system with the double in / out pipes design heat sink,
experimental results showed that more uniform temperature distributions were obtained on the heat sink substrate. Experimental investigations were also conducted under different pumping powers to evaluate the cooling performances of such heat sink design. Although better cooling capability relied on increased pumping power, there should be some tradeoff in design between the heat transfer efficiency and the power consumption. In the present experiments, the heat power density of $17.7 \text{W/cm}^2$ could effectively be removed with the substrate temperature of $61.4\,^\circ\text{C}$ and $61.2\,^\circ\text{C}$ under $0.72\text{W}$ pump power. The results show that the heat sink design also provides a good choice for long-distance electronic products cooling applications under small flow rate.

Acknowledgments

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References


Appendix: Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$A$</td>
<td>area of the bottom wall of the microchannel heatsink</td>
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<tr>
<td>$H, W, L$</td>
<td>thickness, width and length of microchannel plate</td>
</tr>
<tr>
<td>$P$</td>
<td>input power</td>
</tr>
<tr>
<td>$V$</td>
<td>electric voltage</td>
</tr>
<tr>
<td>$I$</td>
<td>electric current</td>
</tr>
<tr>
<td>$Q$</td>
<td>heat transfer rate removed by water</td>
</tr>
<tr>
<td>$T$</td>
<td>temperature</td>
</tr>
<tr>
<td>$\Delta T$</td>
<td>change in temperature</td>
</tr>
<tr>
<td>$a$</td>
<td>seal region width around the microchannel plate perimeter with no honeycomb</td>
</tr>
<tr>
<td>$c_p$</td>
<td>specific heat</td>
</tr>
<tr>
<td>$m$</td>
<td>total mass flow rate</td>
</tr>
<tr>
<td>$q$</td>
<td>heat flux at the bottom wall of the microchannel heatsink removed by water</td>
</tr>
<tr>
<td>$d$</td>
<td>honeycomb cell ex-circle diameter</td>
</tr>
<tr>
<td>$t$</td>
<td>honeycomb cell thickness</td>
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