Applicability of Minichannel Cooling Fins to the Next Generation Power Devices as a Single-Phase-Flow Heat Transfer Device

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Abstract

The heat transfer characteristics of copper minichannel-finned heat sinks are experimentally investigated in order to clarify their applicability as a single-phase flow cooling device for next generation power devices. The influence of the channel width and the fin thickness are evaluated in detail. In particular, the minichannel-finned heat sink having a channel width of 0.3 mm and a fin thickness of 1.0 mm achieves a heat transfer performance of approximately 70,000~9,5000 W/m²K at 300 W/cm² even in a single-phase flow heat transfer regime. Simple estimation proves that single-phase-flow heat transfer with the minichannel-fins heat sink is able to sufficiently cool future power devices under the allowable pumping power conditions.

Keywords: Cooling, Minichannel-finned Heat Sink, Power Electronics, High Heat Flux, Single-phase Flow Heat transfer

1. Introduction

Recent advances in semiconductor elements have dramatically increased its processing speed and also allowed their sizes to be reduced. In addition, the heat-power density of a wide variety of electronic devices has increased, and some of the devices require cooling performance of almost 100 W/cm². Notably, this value is predicted to reach 300 W/cm² for SiC-based power electronics in the not-too-distant future. Because of this, the temperature of future semiconductor elements will need to be kept within allowable limits by means of innovative and economical cooling methods with high heat transfer rate under high heat flux conditions.

With the increase in the heat-power density, cooling methods for electronic devices have transitioned from forced-air cooling to forced-liquid cooling. However, in general, in order to remove a heat flux that exceeds 300 W/cm² at lower electrical power consumption rates, i.e. at lower pumping power, the introduction of the boiling heat transfer technique, which utilizes the thermal potential of the latent heat of vaporization, is one practical method. For instance, Suzuki et al. proposed a cooling device applying subcooled flow boiling and succeeded in a heat removal of 500 W/cm².[e.g. 1-3] Ohta et al. have tried high heat flux removal utilizing flow boiling heat transfer for an enlarged heating area and developed a unique cooling device with a liquid supplying system to prevent dry-out.[e.g. 4–6]

On the other hand, single-phase flow heat transfer utilizing a liquid coolant for high heat flux removal over 300 W/cm² has many merits, such as no vibration, no noise due to the boiling bubbles, and no corrosion of the heat transfer surface. In order to enhance single-phase heat transfer, we could first increase the area of the heat transfer surface using a finned heat sink. Here, focusing on an inverter cooling of a future electric vehicle (E.V.) using a finned heat sink, the cooling fins should be placed onto each chip. This enables the heat sink to take advantage of high heat transfer rate of a non-fully developed thermal boundary layer. Furthermore, by enhancing the fluid mixing between the finned heat sinks, each takes advantage of this thermal inertia effect. In addition, the introduction of a microchannel and a minichannel as the finned heat sink can highly enhance the local heat transfer rate itself due to the channel-narrowing effect. From the viewpoint of long term use of an E.V., the wisest technique to cool the inverter chip of the E.V. is single-phase-flow heat transfer working in a common drive and flow boiling heat transfer working as a backup for emergent cases such as driving
up a steep hill or rapidly accelerating.

Summarizing the vast amounts of conventional studies on finned heat sinks, most of them have evaluated the heat transfer characteristics of forced-air cooling. Recently, Ishizuka et al. have tried to more precisely construct the heat transfer correlation including a mixed convection regime.[7] Also, for forced-liquid cooling, there are also many studies with regards to the single-phase flow heat transfer characteristics in both the minichannel and microchannel fins/grooves for electronic cooling.[e.g. 8–13] However, most of these studies haven’t evaluated the concrete applicability of these fins to future power devices and, furthermore, their heat transfer experiments were performed under low heat flux conditions, below 200 W/cm². As to the microchannel fins, the pressure loss is conceivably quite large, so there aren’t any advantages in single-phase-flow cooling from the viewpoint of cost performance including the fabrication cost. In that sense, we investigate the single-phase flow heat transfer performance of copper minichannel finned heat sinks, which have a submillimeter channel, in order to clarify the applicability of the minichannel finned heat sink using water as the cooling liquid under high heat flux conditions of over 200 W/cm².

2. Heat Transfer Experiments

Figure 1 shows an overview of the experimental apparatus. The cooling liquid used in this experiment is distilled water. This apparatus mainly consists of the cooling liquid supply section, the flow rate measuring section, and the heat transfer test section. In the cooling liquid supply section, the cooling liquid is discharged with a magnetic pump-2, and then flows into the heat transfer test section after passing through the flow rate measuring fins. The cooling liquid heated in the test section is returned to the tank-1, and the inlet temperature of the coolant is then adjusted by the heat exchanger. The flow rate of the cooling liquid is controlled by several valves and measured with a turbine flow meter.

Figure 2 shows a cross-sectional view of the test section. The test section is composed of the flow channel, the heat sink, the thermal insulation plate, and the copper heat transfer blocks. Five cartridge heaters, each 500 W in capacity, are inserted into the bottom part of the heat transfer block. The thermal output of each heater is controlled by a voltage slider. The heat flux from the heater is increased though the pyramid-shaped heat transfer block and then transferred to the convex heat transfer block. The heat transferred to the convex block is removed by the heat sink installed onto the top of the convex block. Four sheathed K-thermocouples, each 1.0 mm in diameter, are inserted into the pyramid-shaped heat transfer block, and five sheathed K-thermocouples, each 0.5 mm in diameter, are inserted into the convex heat transfer block.
The minichannel finned heat sink used in this experiment is a cube 15 mm on each side, which is made as an all-in-one heat sink integrated with the convex heat transfer block. To reduce the heat loss from the convex heat transfer block to the stainless flow duct channel, a ceramic thermal insulation plate is installed between them. The flow channel is a square tube 15 mm on a side. A thermocouple TC1 is installed 8.5 mm upstream from the inlet of the heat sink, and two K-thermocouples (TC2 and TC3), are installed 8.5 mm downstream from the outlet. At each location, the inlet and outlet temperatures of the liquid are measured. Moreover, a pressure measuring port is installed 8.5 mm upstream and 8.5 mm downstream from the heat sink in order to measure the differential pressure produced by the heat sink. Table 1 shows the heat sink parameters used in this experiment. The fins used are copper minichannel fins. For instance, Heat Sink #10-03 has a fin thickness of 1.0 mm and a channel width of 0.3 mm. Comparing #10-06 with #05-03 having the same porosity enables us to evaluate the channel narrowing effect on the heat transfer characteristics. The influence of the fin thickness on the heat transfer performance can also be evaluated by comparing #10-06 with #03-06, and #05-03 with #10-03, respectively. By comparing #10-06 with #10-03, the influence of the channel width on the heat transfer performance can be evaluated.

The heat transfer experiment starts under lower flow velocity and low heat flux conditions. After a steady state of each temperature is confirmed, temperature data is obtained for 40 seconds. The same process is repeated, raising the heat flux until the temperature of the cartridge heater exceeds 800°C. An averaged wall-temperature and a temperature gradient at the bottom of the fin are estimated using the above-mentioned temperature data in the convex heat transfer block. The removal heat flux can be calculated by Fourier’s law, and the overall heat transfer coefficient is defined by the temperature difference between the wall temperature and the mixed bulk temperature, which is the average temperature of the inlet and outlet fluids. The inlet temperature is around 17 to 23°C and the maximum outlet temperature is 32°C in these experiments. The uncertainty of the heat transfer coefficient is 9.5%.

3. Experimental Results

3.1 Flow characteristics of minichannel finned heat sink

Figure 3 shows the flow characteristics of each minichannel finned heat sink. The temperature of the water is 21.0 ± 1.0°C. First, the results with the same channel widths (#03-06 vs #10-06; and, #05-03 vs #10-03) are compared. In general, the pressure loss of the finned heat sink can be approximated by the square of the flow velocity in the fin channel, (i.e. the square of porosity), the channel width, and the number of the flow channels. The results show that the pressure loss becomes higher where the finned heat sink has fewer flow channels and lower porosity at the same channel width. The flow velocity in the fin channel becomes higher with the decrease in porosity, which is a more dominant cause than the effect of decreasing the number of channels. On the other hand, judging from the comparison of #10-06 with #05-03, which have the same porosity (i.e., almost the same flow velocity in the fin channel), the higher pressure loss of #05-03 is due to the effects of the channel width and the number of channels. However, the pressure losses for all the minichannel finned heat sinks are higher than predicted.[14] As the causes of the frictional resistance of the finned heat sink, the entrance loss and the discharge loss can probably also be regarded as major factors. In addition, development of a velocity boundary layer in the fin channel also depends on the inlet flow conditions, which are conceivable determined by the porosity and the channel width. In that sense, both the entrance loss and the discharge loss should be taken into account more precisely when establishing a pressure loss correlation. In the meantime, the pressure losses of the finned heat sinks are on the order of 1~10 kPa even #10-03, which suffered the largest pressure

<table>
<thead>
<tr>
<th>Fins</th>
<th>Fin thickness (mm)</th>
<th>Channel width (mm)</th>
<th>Porosity (%)</th>
<th>Number of fins</th>
<th>Surface area (m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>#10-06</td>
<td>1.0</td>
<td>0.6</td>
<td>36</td>
<td>10</td>
<td>0.00458</td>
</tr>
<tr>
<td>#05-03</td>
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<td>0.3</td>
<td>36</td>
<td>19</td>
<td>0.00908</td>
</tr>
<tr>
<td>#10-03</td>
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<td>0.3</td>
<td>22</td>
<td>12</td>
<td>0.00555</td>
</tr>
<tr>
<td>#03-06</td>
<td>0.3</td>
<td>0.6</td>
<td>64</td>
<td>17</td>
<td>0.00814</td>
</tr>
</tbody>
</table>
loss. This suggests the use of a parallel circuit of finned heat sinks, rather than a series circuit, with the large lift force of a circulation pump.

3.2 General heat transfer characteristics of copper minichannel finned heat sink

Figures 4–7 show the heat transfer characteristics of the copper minichannel finned heat sink. The heat transfer characteristics of all the fins are broadly divided into the following two regimes: a single-phase flow heat transfer regime and a flow boiling heat transfer regime only observed in Fig. 4 & Fig. 7. The heat transfer coefficient increases higher as the flow velocity increases. One interesting thing is that the heat transfer coefficients increase by more than 10,000 W/m²K in the heat flux range of 1~8 MW/m² (i.e. 100~800 W/cm²) even in the single-phase heat transfer regime except for #03-06. This is contrary to Newton’s cooling law in which the heat transfer coefficient basically doesn’t depend on the heat flux. The increase in the heat transfer coefficient in the single-phase flow heat transfer regime is also caused when the viscosity of the cooling liquid in the vicinity of the heat transfer surface significantly declines due to the increase in fluid temperature, compared with the viscosity at the mixed bulk temperature. However, this effect is conceivably small especially for low-Pr number fluids including water. The temperature distribution within the fins and the fin efficiency highly depend on the heat flux itself, so that this cause should be discussed with conjugate CFD analysis, including the occurrence of natural convection. On the other hand, this discussion suggests that the conventional simple heat transfer correlations would underestimate the heat transfer performance of the minichannel fins in this heat flux region. In the flow boiling regime, the heat transfer coefficients increase rapidly as shown in Figs. 4 and 7. If a point where the heat transfer coefficient starts to increase more rapidly is defined as the onset of nucleate boiling (ONB), the ONB appears at the higher wall superheat with increasing flow velocity. This is because of the velocity boundary layer becoming thinner, which condenses the boiling bubbles more easily before growing up.

Summarizing all the data, a heat removal performance of 60,000~120,000 W/m²K is achieved even in a single-phase heat transfer regime for the finned heat sinks #10-03 and #05-03. In particular, #10-03 has the highest heat transfer performance, achieving a heat transfer coefficient of 70,000~95,000 W/m²K at the targeted heat flux of 300 W/cm² (i.e., 3 MW/m²) as shown in Fig. 6. This proves that the heat transfer performance of the minichannel fins is highly enhanced due to the fin effect and the channel narrowing effect. These effects seem to overcome the boiling heat transfer performance observed for heat sinks #03-06 and #10-06 with the wider channel width.

Figure 8 shows the heat transfer performances of all the fins at the inlet flow velocity of u_in = 0.23 m/s. It is obvious that the heat transfer performance of the minichannel finned heat sink is much higher than that of a flat plate.
Fig. 4  Heat transfer characteristics of #10-06.

Fig. 5  Heat transfer characteristics of #05-03.

Fig. 6  Heat transfer characteristics of #10-03.

Fig. 7  Heat transfer characteristics of #03-06.
The heat transfer coefficients of #10-03 and #05-03, having a narrower channel width, show higher heat transfer performance, and the fins with the thicker width also lead to higher heat transfer performance. This suggests that a thicker finned heat sink is more suitable for cooling with water and some other liquids, compared with air-cooling, in order to keep fin efficiency high. The heat transfer performance of the finned heat sink is determined based on the fin efficiency, which is determined mainly by the flow conditions in the fin channel and the fin thickness, as well as the effective heat transfer area. Therefore, this point will be discussed more systematically in the following sections.

3.3 Influence of fin thickness on heat transfer characteristics

To simplify the discussion, the heat transfer characteristics are compared for the same flow velocity in the fin channel of the same width. Figure 9 (a) shows the heat transfer coefficients of #10-03 and #05-03, and Fig. 9 (b) shows those of #10-06 and #03-06, respectively. Since the Re number in the fin channel is the same in each figure (Re ~300 and 360 at 30°C, respectively), the distribution of the local heat transfer rate on the fin surface for each heat sink is almost equivalent in a single-phase flow heat transfer regime. Table 2 shows the fin efficiency and the effective heat transfer area predicted for each minichannel finned heat sink in order to compare each heat transfer performance. The flow velocity in the fin channel is assumed at 0.7 m/s. The heat transfer correlation for a channel flow was applied to estimate the heat transfer coefficient. Where the distribution of the local heat transfer coefficient is identical for each fin, the overall heat transfer coefficient, especially in the single-phase flow heat transfer regime, must depend on the effective heat transfer area. Therefore, #05-03, which has a slightly larger effective heat transfer area, shows a slightly higher heat transfer coefficient compared to #10-03. Furthermore, the effective heat transfer areas for #10-06 and #03-06 are almost equal, so that #10-06 shows almost the same heat transfer coefficients for #03-06 though there exists a difference probably due to a mixed convection regime over the wall temperature of 80°C. In that sense, if the velocity in the fin channel is the same, the fin thickness does not affect the heat transfer performance so much due to a trade-off relationship between the fin efficiency and the heat transfer area. This is conceivably due to the development of a thermal boundary layer similar to each other because the Re number is the same though the inlet flow condition for the fin channel is different due to different porosity. There could be a need to take into account actual flow conditions in the
3.4 Influence of fin channel width on heat transfer characteristics

The influence of the fin channel width on the heat transfer characteristics is discussed for the same flow velocity in the fin channel. Comparing #10-03 and #10-06 having the same fin thickness but different channel widths (see Fig. 10) confirms that #10-03 shows a much higher overall heat transfer coefficient. This suggests that the increase in the heat transfer coefficient due to the channel narrowing effect is more significant than the influence of the decrease of the Re number. In this case, the effective heat transfer area is one factor which can determine the overall heat transfer coefficient of the finned heat sink, unlike the last discussion where the fin thickness changes. Even under the same conditions of flow velocity in the fin channel, the local heat transfer coefficient dramatically increases due to the channel narrowing effect. For this reason, not only the effective heat transfer area but also the local heat transfer coefficient is an important factor in determining the overall heat transfer coefficient of the finned heat sink. Table 2 also shows that #10-03 has significantly better results in the effective heat transfer area and the local heat transfer coefficient.

3.5 Comparison of heat transfer performance by pumping power

The results of the heat transfer characteristics obtained suggested that the heat sink with narrower channels and thicker fins (i.e. lower porosity) had a higher overall heat transfer coefficient. Further, the heat transfer performance of water is much superior to that of air, which needs a larger heat capacity fin to enhance the fin efficiency. However, if the channel becomes narrower, frictional resistance in the channel increases inversely proportional to the width. Additionally, the minichannel finned heat sink with lower porosity produces additional pressure loss due to the entrance and discharge losses at the inlet and outlet ports of the finned heat sink. Figure 11 shows the relationship between the pumping power and the heat transfer coefficients at the heat fluxes of approximately 0.9, 6.8, and 8.5 MW/m². Under the conditions of the single-phase flow, the heat transfer coefficient increases with an increase in pumping power. In addition, at the same heat flux, the finned heat sink having thicker fins and narrower channels also shows higher heat transfer performance, which allows us to introduce the minichannel fins for power device cooling under the allowable pumping pressure conditions.
4. Applicability of Minichannel Cooling Fins to the Next Generation Power Devices

Finally, the feasibility of the minichannel finned heat sink for power device cooling is discussed. Here, it is assumed that the heat removal performance is 90,000 W/m²K at 300 W/cm² which was obtained for #10-03 at \( u_{in} = 0.5 \) m/s, and that the bulk temperature of the cooling water is from 50 to 80°C at atmospheric pressure. The heat flux from the chip is 300 W/cm² and the thickness of the base of the fins is \( 5.0 \times 10^{-3} \) m as shown in Fig. 12. This calculation model corresponds to the most severe assumption because a heat spreader generally used isn’t installed and the heat transfer coefficient used is the one obtained for the bulk inlet-temperature of approximately 20°C, though the thermal resistance between the chip and the heat sink isn’t taken into account. In general, the length to achieve a fully-developed flow and temperature fields for forced water cooling is much longer than that of forced air cooling. For instance, at the same inlet flow velocity of \( u_{in} = 0.5 \) m/s \( (u_{local} = 2.5 \) m/s) for #10-03, the length of the entrance region estimated by \( 0.0575 \cdot Re \cdot Pr \cdot D \) is more than 100 mm (against the fin length of 15 mm), although that for air cooling is less than 1.0 mm, as shown in Fig. 13. This rough estimation suggests us that the conventional heat transfer correlations proposed for the fully-developed flow aren’t applicable to forced water cooling in the present cases. However, this fact leads to an advantage of utilizing liquid cooling from the view point of enhanced heat transfer performance due to the thermal inertia effect, but also leads to a much more complicated situation for predicting the temperature distribution within the fin and the fin efficiency. In that sense, the present estimation roughly evaluates the applicability of the minichannel finned heat sink to power devices. Fig. 14 shows the wall temperature and the surface temperature of the chip in the middle of the fin, which are estimated by Newton’s cooling law. This result proves us that the minichannel finned heat sink is applicable for cooling electronic power devices with a heat flux of 300 W/cm², because the wall temperature is below ONB and the surface temperature of the chip is also much lower than the allowable temperature of a SiC element (~200°C). A safer cooling operation could be possible, for example by using a heat spreader. In the next step, a 3-dimensional CFD simulation with conjugate analysis should be performed in order to evaluate the flow structure in the fin channel and the 3-dimensional temperature distribution within the fin.

5. Conclusion

In this research, we experimentally evaluated the heat transfer characteristics of copper minichannel fins mainly in a single-phase heat transfer regime under heat flux conditions over 200 W/cm². The minichannel finned heat sink having the narrowest channel width of 0.3 mm and the thickest fin thickness of 1.0 mm achieves the highest heat transfer performance of over 100,000 W/m²K. Assuming that the heat transfer performance of this heat sink is 90,000 W/m²K at the heat flux generated from the inverter of 300 W/cm², the temperature difference between the wall temperature and the bulk fluid temperature is 33°C, which results in a wall temperature of 113°C against the bulk fluid temperature of 80°C. The temperature of the chip itself is much lower than the allowable temperature of
200°C. This simple estimation sufficiently proves that single-phase heat transfer using the minichannel finned heat sink is also applicable to cooling future power devices.

References


