Thermal management of automotive SiC-based on-board inverter with 500 W/cm² in heat flux, and Two-phase immersion cooling by breathing phenomenon spontaneously induced by lotus porous copper jointed onto a grooved heat transfer surface

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Abstract
In this study, we quantitatively discuss several kinds of thermal resistances that rise up to the surface in thermal management of automotive SiC-based on-board inverters with extremely high heat flux of 500 W/cm². It is proven that a stacking structure without a heat spreader and a direct cooling method utilizing two-phase immersion cooling technique can provide a breakthrough as the key technology. Furthermore, it is verified that a two-phase immersion cooling technique using “breathing phenomenon” spontaneously induced by a lotus copper jointed onto a grooved heat transfer surface can realize both the above-mentioned stacking structure and extremely high critical heat flux beyond 500 W/cm² in a saturated pool boiling environment (534 W/cm² at the maximum).

Keywords: SiC, Automotive on-board inverter, Thermal management, Thermal resistance, Heat spreader, Thermal contact resistance, High heat flux, Two-phase immersion cooling, Lotus copper, Breathing phenomenon, Pool boiling, Critical heat flux

1. Introduction

Japan is facing immense energy and environmental crises since the occurrence of the Fukushima Nuclear Power Plant accident in 2011. At present, “renovation and diffusion of energy-saving technology”, such as power electronics, seems to be one of the most promising solutions to solve the energy and environmental problems. Especially in Japan, silicon carbide (SiC) and gallium nitride (GaN) have been developed as alternative power semiconductor materials for replacing Si in the power electronics. For instance, big data society that employs a data center and a smart grid system increases the power consumption, ultimately increasing the need for using energy-saving power semiconductors such as SiC and GaN. In addition, in the transportation sector that accounts for ~20% of carbon dioxide emissions, next-generation electric and fuel-cell vehicles are expected to replace gasoline vehicles to reduce the CO₂ emissions. In particular, by introducing SiC to an automotive on-board inverter for the future vehicles, the power loss can be reduced and vehicle’s mileage can be also increased (Seal et al., 2017; Kimura et al., 2014).

Thermal management in power electronics will, however, become much more difficult than anticipated. For
Table 1  Comparison of present Si inverter system with future SiC inverter system in HV

| Present Si Inverter System | Future SiC Inverter System |
|----------------------------|----------------------------|
| Max Temp. of Chip (°C)     | ~180                       | ~250                       |
| Heat Flux (W/cm²)          | <150                       | <300 ~ 500                 |
| Inlet Temp. of Coolant (°C)| 65                         | 110                       |
| Pumping Power Q/QSi [-]    | 1                          | > 10 (by single-phase flow) |
| Cooling System             | Two loops                  | Single loop                |
| Single-phase Flow Cooling  | Possible                   | Not recommended            |
| Two-phase Flow Cooling     | Unnecessary                | ?                          |
| Immersion Cooling          | ?                          | ?                          |

Fig. 1 Typical structure of future SiC HV cooling system, and cross sectional view of typical power package. In the thermal management of the power electronics, various thermal resistances must be considered in a semiconductor chip, at an insulating substrate, TIM, Bonding material, in a heat spreader and a heat sink.

instance, using the SiC semiconductor has several drawbacks due to the operating temperature of over 200°C, such as the bonding material and the surrounding structural materials are influenced and damaged by high temperature operation and thermal fatigue (Hao Zhang et al., 2017; Ishiyama et al., 2018; Osaki et al., 2019). Moreover, the heat flux of the SiC-based on-vehicle inverters reaches 300–500 W/cm² (3–5 MW/m²) due to downsizing of the chip itself, their spec-up, and increasing current density inside the SiC semiconductor. Of course, similar to the present hybrid vehicles, liquid cooling systems with single-phase heat transfer can be applied also to the SiC-based inverters with high heat flux. A heavier weight and higher power pump will be, however, required to achieve the desired cooling performance. Table 1 compares the present Si inverter system with future SiC inverter system in HV from the view point of the cooling system. Introducing the SiC inverter system enables a single loop (see Fig. 1 on the left) that includes the inverter, motor generators, an engine, a radiator etc. due to its high temperature operation of ~250 °C although the present Si inverter system utilizes two separate loops for cooling the inverter (inlet temperature: 65 °C) and the engine (Tsuruta, 2011). As mentioned in section 2.4 in detail, however, there could not be a practical solution for cooling the future SiC-based inverter with the heat flux of 500 W/cm² for both the single-phase flow cooling and two-phase flow cooling systems.

In the thermal management of the automotive SiC-based inverter, in addition, various thermal resistances, which are generated between the power element and the coolant, must be considered more carefully than the other electronics. Fig. 1 on the right shows a cross sectional view of a typical power package (Zheng et al., 2015). In fact, a hot spot generated inside the power element (Kibushi et al., 2017) and thermal resistances generated at the interface of bonding material, thermal interface material (TIM), and insulating material as well as in the heat spreader rise up to the surface in the thermal management of high heat flux electronics as described in detail in section 2. However, to the best of our knowledge, the thermal resistances under high heat flux conditions of 500 W/cm² targeting the automotive SiC-based inverters have not been reported yet.

This study first discuses an ideal stacking structure around the SiC power element by quantitatively estimating the thermal resistances under high heat flux conditions of 300 - 500 W/cm². And then, a power-free two-phase immersion cooling technology, which utilizes lotus-type of porous copper jointed onto a grooved heat transfer surface, is proposed
to realize above-mentioned ideal stacking structure by experimentally demonstrating the heat transfer performance beyond 500 W/cm².

2. Issues on thermal resistance unique to high heat flux power electronics

In the thermal management of the SiC-based on-vehicle inverters, various kinds of thermal resistance issues rise up to the surface due to the increase in the heat flux. As it is not possible to propose a concrete cooling technique without consideration of these thermal resistances, the thermal resistances and the accompanied temperature jumps are quantitatively evaluated first.

2.1 Thermal resistance generated in an insulating substrate

As illustrated in Fig. 1, the insulating substrate exists between the power element and the cooling surface. Because the substrate thickness is usually of a micro scale for common electronics, the temperature difference generated is rarely paid attention. The thickness of the insulating substrate in the SiC-based on-vehicle inverters is, however, expected to be 400 μm at the maximum. Assuming that the thermal conductivity of a ceramic insulating substrate using aluminum oxide, aluminum nitride, silicon nitride etc. is approximately 50 W/m/K, the temperature difference of 40 K occurs in one single substrate at the heat flux of 500 W/cm² according to the Fourier’s law of heat conduction. The temperature difference of 40 K generated at the insulating substrate highly affects cooling conditions so that much thinner insulating substrate with much higher thermal conductivity of over 100 W/m/K should be developed.

2.2 Thermal resistance generated in a heat spreader

To enhance heat dissipation in electronics, in general, area of heat transfer surface, i.e. area of cooling surface, is expanded through a heat spreader, which makes it easier to remove the heat due to reduced heat flux. However, assuming that a copper flat-type heat spreader of 5 mm in thickness is used and that the heat is spread uniformly at 45°, the temperature difference of approximately 30 K occurs at the heat flux of 500 W/cm² (~20 K at 300 W/cm²). As there is no reason for the “45° heat spreading assumption” that is frequently used in thermal design, numerical simulations on heat conduction inside the heat spreader are performed under the actual heat flux conditions of 500 W/cm² (Kibushi et al., 2018). The area of the heat spreader is 2500 mm² (50 mm x 50 mm) and the thicknesses are 3.0, 5.0, and 7.0 mm, respectively. The heat flux of 500 W/cm² is given at the top surface of 10 mm x 10 mm as shown in Fig. 2 on the left, and the bottom surface is cooled at a certain heat transfer coefficient and a referential fluid-temperature of 60.0 °C. Adiabatic conditions are given at the other boundaries. In the Fig. 2 on the right, the horizontal axis represents the heat transfer coefficient of the coolant. Three broken lines represent temperature differences obtained by the 45° heat spreading assumption for each thickness. With increasing heat transfer coefficient, the temperature difference generated at the central axis of the heat spreader increases. In particular, at the thicknesses of 5.0 mm and 7.0 mm, the temperature

![Fig. 2 Heat conduction simulation in a heat spreader. With increasing heat transfer coefficient, the temperature difference generated in the heat spreader increases. In particular, at the thicknesses of 5.0 mm and 7.0 mm, the temperature difference obtained by the numerical simulation is larger than that obtained by the 45° assumption especially in a high heat transfer coefficient regime.](image-url)
2.3 Contact thermal resistance generated at an interface of TIM and bonding material

In an indirect-type of cooling technology, thermal interface material (TIM) is frequently used at the interface between the heat spreader and the heat sink. In addition, bonding material must be used at the interface near the power element. Because there haven’t been any studies on the contact thermal resistance under the high heat flux conditions of over 100 W/cm\(^2\), so that we evaluated the contact thermal resistance generated at the interface using TIM and the bonding material (Kibushi et al., 2018, 2019). The experimental setup shown in Fig. 3 on the left is similar to generally introduced contact thermal resistance test setup by a steady state method, in which two identical copper blocks are stacked each other through each circular surface of 10 mm in diameter. To achieve a high heat flux of over 100 W/cm\(^2\), the cross section of each copper block is reduced like Fig. 3. The details of the setup and the evaluation method of the contact thermal resistance are reported in the literature (Kibushi et al., 2018). Three types of interface materials, silicon grease (0.76 W/m/K), solder (33 W/m/K), and copper paste (N/A) are evaluated. The loads applied to the jointed surface are 0.33, 1.71, and 3.03 MPa, respectively. Fig. 3 on the right shows the temperature difference generated at the interface. The temperature difference obtained for the copper paste is much larger than 100 K at the heat flux of 500 W/cm\(^2\) even at the load of 3.03 MPa (also for the silicon grease at the load of 0.33 MPa). The temperature differences for the silicon grease at the loads of 1.71 and 3.03 MPa seem to exceed 70 K at the heat flux of 500 W/cm\(^2\). On the other hand, the temperature difference can be reduced to approximately 10 K at the heat flux of 500 W/cm\(^2\) in the solder case regardless of the load. As there are three bonding layers in Fig. 1, there exists the temperature gap of over 20 K at least, ignoring the bonding layer between the heat sink and the heat spreader. In addition, the solder cannot be used under the temperature conditions of over the melting point of 221 °C. If the maximum operating temperature of the SiC semiconductor reaches 250 °C, the development of the bonding material applicable for long time use under high temperature conditions, such as silver paste containing nanoparticles, is indispensable.

2.4 Thermal resistance in a heat sink and cooling issues

When maintaining the temperature of the power element at 200 ~ 250 °C with the heat flux of 500 W/cm\(^2\), the
temperature difference of over 100 K would be generated between the power element to the cooling surface in the heat sink because a temperature gradient is also generated in the SiC semiconductor element due to the hot spot issue. Thus, the temperature of the cooling surface needs to be maintained at ~100°C or less. Generally, boiling heat transfer utilizing latent heat of vaporization is expected for cooling system against the heat flux of 500 W/cm², but a saturation temperature of 50 vol.% concentration of long life coolant (LLC), which is expected to be applied as the cooling liquid for the automotive inverter, is approximately 110 °C. Thus, there is no solution for applying the conventional mounting structure around the semiconductor element to the SiC-based automotive inverter with the heat sink utilizing boiling heat transfer. In addition, even if the boiling heat transfer can be utilized, since LLC contains not only ethylene glycol as the main component but also a rust inhibitor and a defoaming agent, there is a concern of scale deposition on the heat transfer surface and of generation of floating material in the liquid (Ma et al., 2017). In contrast, when maintaining the temperature of the cooling surface at ~100 °C using single-phase heat transfer cooling system while avoiding boiling heat transfer, the heat transfer coefficient required for the heat flux of 500 W/cm² is over 70 kW/m²/K (assuming a coolant inlet-temperature of 60 °C) based on the Newton’s cooling law. This value can be attained by high-velocity water cooling in narrow channels such as minichannel fins (Yuki et al., 2011). However, since LLC is categorized as a high Pr number fluid, as shown in Fig. 4 on the left, the cooling performance of LLC as a single-phase flow is considerably deteriorated. The heat transfer coefficient of LLC flow at 30 vol.% and 60 °C is approximately 60% of that of water flow, as estimated by the Dittus-Boelter’s heat transfer correlation for turbulent flow in a circular pipe of 1.0 mm and 5.0 mm in diameter as shown in Fig. 4 on the right (the Sieder-Tate heat transfer correlation for laminar flow). The sudden change in Fig. 4 on the right indicates transition from the laminar flow to the turbulent flow. Furthermore, to apply the existing single-phase cooling system for the Si-based inverter toward the SiC-based inverter with the heat flux of 500 W/cm², the required pumping power is likely to exceed 10 times the current one. In the development of electric and fuel-cell vehicles, there is a strong need to minimize the power consumption except for the motor, so that it is desirable to develop a cooling technology with lower pumping power.

Summarizing the above-mentioned discussions, various thermal resistances that rise up to the surface under the high heat flux conditions of 500 W/cm² lead to the limitation on the cooling method. In addition, considering the large temperature difference between the semiconductor element and the heat sink as well as the applicability of LLC to boiling heat transfer and the increase in pumping power in single-phase flow cooling, a direct cooling method without using the heat spreader and with much thinner insulating substrate with higher thermal conductivity is quite reasonable in such high heat flux environment of 500 W/cm².

3. Two-phase Immersion Cooling by Lotus-type Porous Copper jointed onto a Grooved Surface

3.1 Improvement of critical heat flux by breathing phenomenon

For the thermal management of the SiC-based automotive inverter, the application of two-phase immersion cooling
Fig. 5 Breathing phenomenon spontaneously induced by lotus copper on a grooved heat transfer surface (Two spontaneous breathing modes can be expected: the one where the coolant is supplied from the upper side of the lotus copper and the vapor is discharged from the groove outlets (on the left) and the one where the vapor is discharged from the lotus copper where the liquid is supplied from the groove (on the right))

without circulating the coolant is of great advantage. For example, because a circulating pump is not required, the cooling system can be made very simple and compact, which also contributes to weight-saving and an extended vehicle’s mileage. Furthermore, in a closed container type of immersion cooling, the use of LLC containing the rust inhibitor and the defoaming agent is unnecessary because it doesn’t need to circulate the liquid to the radiator. Hence, there is a possibility that antifreeze such as ethylene glycol solution or water can be utilized as a coolant. The only issue is that the two-phase immersion cooling isn’t available due to the aforementioned large temperature difference between the chip and the cooling surface. However, if the temperature of the cooling surface can be made higher by introducing the direct cooling method without the heat spreader and with thinner and higher thermal conductivity insulator substrate, the two-phase immersion cooling can be introduced. Thus, whether the critical heat flux (CHF) of the two-phase immersion cooling, i.e. CHF of saturated pool boiling, exceeds 500 W/cm² remains a challenge.

In this background, the authors, in collaboration with Lotus Thermal Solution Co., Ltd., proposed the application of lotus type of porous copper (hereafter, lotus copper) having a unidirectional pore structure jointed onto a grooved heat transfer surface as illustrated in Fig. 5 in order to increase CHF of flow and pool boiling (Yuki et al., 2016a, 2017a, 2018). Of course, there have been a lot of tremendous porous studies on boiling/evaporation heat transfer and CHF enhancements utilizing the already known capillary phenomenon and further activation of cavity (e.g. Allingham et al., 1961; Cornwell et al., 1976; Nishikawa et al., 1978; Fuji et al., 1984; Liter et al., 2001; Williams et al., 2005; Hwang et al., 2006; Li et al., 2007; Chinmay et al., 2014; Mori et al., 2017; Yuki 2017b). In special, recent studies on porous modification techniques for boiling heat transfer enhancement and critical heat flux improvement are summarized and reviewed by Chinmay et al. (2014) and Mori et al. (2017). On the other hand, our idea doesn’t depend on the capillary phenomenon and is based on the fact that the inlet pressure sharply decreases when the phase change in the porous medium becomes intense and the vapor is ejected from the porous medium in the flow boiling experiments using porous media presented in our past researches (e.g. Yuki et al., 2010). In other words, there are three characteristic features of the present technology: (I) the liquid is spontaneously supplied into the porous medium by the strong discharge of the vapor; (II) the capillary phenomenon does not contribute to the heat transfer performance; (III) because of the availability of a flow-through liquid supply and vapor discharge, liquid supply and vapor discharge can be completely separated. We call this phenomenon as “breathing phenomenon or breathing effect”. Specifically, two spontaneous breathing modes can be expected: the one where the coolant is supplied from the upper side of the lotus copper and the vapor is discharged from the groove outlets as shown in Fig. 5 on the left and the one where the vapor is discharged from the lotus copper where the liquid is supplied from the groove as illustrated in Fig. 5 on the right. One of the advantages of using the lotus copper is that when the liquid is supplied from the lotus side, the flow resistance can be reduced and also higher effective thermal conductivity in the pore direction leads to higher heat transfer inside the pore because the pore structure is unidirectional.

In our previous research (Yuki et al., 2016b), we acquired reference data for the case when the lotus copper (10 mm square, 2 mm in thickness, porosity of 65.9%, and average pore diameter of 0.49 mm) was attached onto a smooth heat transfer surface and demonstrated that the critical heat flux could exceed 200 W/cm² only by increasing the heat transfer area. Furthermore, when the lotus copper was attached onto grooved heat transfer surface (the groove size: 0.2 mm), the critical heat flux of saturated pool boiling under atmospheric pressure increased to 242 W/cm², which proves that the critical heat flux was enhanced by providing grooves (Yuki et al., 2017a, 2018). Herein, we discuss the possibility of
increasing the critical heat flux when the groove size is further increased.

3.2 Saturated Pool boiling experiment using lotus copper

The experimental setup is shown schematically in Fig. 6. The equipment is mainly composed of a heat transfer copper block, a test container, and a condenser. The test container is made of glass with the width of 63 mm and the height of 72 mm, permitting the observation of the boiling behavior. The heat transfer copper block consists of a square column part and a base part, and heat generated in four cartridge heaters inserted into the base part are regulated. The upper surface of the square column part with the cross section of 10 × 10 mm² is unidirectional grooved one, and the lotus copper is jointed onto the grooved surface by soldering (Solder: Sn96.5Ag3.5) at a certain load with a spring. The cross-sectional size of the groove is 0.4 × 0.4 mm², 0.5 × 0.5 mm², and 1.0 × 1.0 mm². The groove pitches are 1.0 mm for the 0.4 and 0.5 mm grooves and 1.57 mm for the 1.0 mm groove, respectively. Each groove is fabricated by polishing wheel using man-made diamond with the same particle size, so that the roughness of each groove surface is almost equivalent. The coolant is distilled-water, and saturated pool boiling tests are performed at the pressure of 0.1 MPa and a saturation temperature of 100 °C after fully degassing air. During the experiment, the valve of the vent line is completely open, and the degassed distilled-water is occasionally supplied from another tank to keep the liquid level. To confirm the temperature of water to be a saturation temperature of 100 °C, two sheathed T-type thermocouples are inserted into water. The vapor generated during the experiment is condensed with a water-cooled condenser placed at the top of the test container. Four sheathed K-type thermocouples of 0.5 mm in diameter are embedded at the central axis of the copper block square column part. The positions of the thermocouple are 3.0, 7.0, 11.0, and 15.0 mm from the bottom surface of the groove. The temperature profile is linearly approximated from the temperatures at three points on the heat transfer surface side in the steady state, and the heat flux is evaluated based on the Fourier’s law of heat conduction. The coefficient of determination of the least-square method is more than 0.99, and the uncertainty obtained with respect to the critical heat flux value is 7.4 % as in the smooth surface case and 2.3 % at the highest critical heat flux value obtained in the present study.

The lotus copper used in this test was manufactured by Lotus Thermal Solution and has the same area of 10 ×10 mm² as the heat transfer surface. The lotus copper is the same as the one used in the previous study with the average pore diameter of 0.49 mm, the thickness of 2 mm, and the porosity of 65.9%. A photograph of the lotus copper jointed onto the grooved heat transfer surface is shown in Fig. 6 on the right (The groove width, in this case, is 0.5 mm).

3.3 Results and Discussion

Fig. 7 shows the boiling curve when the lotus copper is jointed onto the grooves of 0.4 mm square. For comparison,
the boiling curve obtained for a smooth surface and the boiling curve for the lotus copper jointed onto the grooves of 0.2 mm square are also presented. Considering first the boiling curve for the smooth surface, the critical heat flux is 138 W/cm² (the maximum value of three experiments), which is slightly higher than Zuber's CHF of 110 W/cm², but reasonable when compared to the CHF values in similar studies. On the other hand, the critical heat flux for the lotus copper on the groove size of 0.2 mm square is 242 W/cm² and that for the groove size of 0.4 mm square increases up to 269 W/cm². This value is 1.95 times as high as the critical heat flux for the smooth surface. From these results, it is clear that the critical heat flux increases with increasing groove size. Meanwhile, in the heat flux range up to 200 W/cm², the boiling curves for the two groove sizes do not show a significant difference.

Next, we evaluate the boiling curve obtained for the lotus copper on the grooves of 0.5 mm square (see Fig. 8). Fig. 8 is shown separately from Fig. 7 because all the thermocouples were replaced by the new ones after the experiments of Fig. 7. The critical heat flux of the smooth surface was again evaluated three times and the maximum value is 144 W/cm². In this experiment, we evaluated the boiling curve of the lotus copper on the groove twice by turning the lotus copper plate clockwise at 90° (#2). We confirmed that the critical heat flux for the grooved heat transfer surface (without the lotus copper) is 162 W/cm², which is possible caused by the effect of increased heat transfer area. In contrast, when the lotus is jointed onto the 0.5 mm square grooves, the boiling curve shifts to the side of lower wall superheat compared with the case of 0.4 mm square groove, which proves that the boiling heat transfer is significantly improved. For
instance, the heat flux of the lotus copper is more than 15 times as high as that of the smooth surface at a wall superheat of 6 K, and about five times as high as that of 10 K. This value is even higher than the value obtained for the grooved heat transfer surface. Furthermore, the critical heat flux is also dramatically improved, reaching 461 W/cm² at the wall superheat of 59 K. This value is 3.2 times and 2.8 times higher than the critical heat flux of the smooth and the grooved heat transfer surfaces, respectively. The difference between #1 and #2 conceivably results from non-uniformity of the pore distribution of the lotus copper (see Fig. 6 on the right), which suggests us that it will be needed to evaluate the uncertainty for CHF due to the non-uniformity of the pore distribution. From the visualization of the boiling phenomenon near the critical heat flux, we confirmed that vapor bubbles are ejected from the groove like a vapor curtain as shown in Fig. 9 especially at the groove size of 0.5 mm and 1.0 mm. There is a possibility that the critical heat flux is improved due to vapor discharge from the grooves along with liquid supply from the lotus side, which contributes to retarding formation of a coalescent bubble on the heat transfer surface and also to inducing “flow-through pathway flow” from the lotus copper to the grooves that separates the vapor discharge and the liquid supply.

To discuss a principal factor of the CHF improvement, we evaluate the enlargement effect of the heat transfer area, assuming that the surface area when soldering the lotus copper onto a smooth surface is 1. The enlargement ratio of area $R_{\text{area}}$ and the increase ratio of critical heat flux $R_{\text{CHF}}$ are shown in Table 2. Accordingly, in the case of 0.2 mm square and 0.4 mm square grooves, no increase in CHF commensurable with the enlargement effect of the heat transfer area due to grooves can’t be confirmed. In the case of the groove size of 0.5 mm square, however, the CHF increasing rate and the heat transfer area enlargement rate are almost the same. This means that some other phenomena beyond the heat transfer area enlargement effect is likely to contribute to the enhanced critical heat flux by the increase of the groove size. To confirm the effect of the groove size again, we conducted the same pool boiling experiment at the groove size of 1.0 mm (see Fig. 10). In this case, the contacting width of the groove top to the lotus copper is 0.57 mm, almost equal to the width of the groove top to the lotus copper 0.57 mm, almost equal to the width of the groove top to the lotus copper.

Table 2  Relationship between enlargement effect of heat transfer surface and CHF improvement. In the case of the groove size of 0.5 mm square, the CHF increasing rate and the heat transfer area enlargement rate are almost the same, which means that some phenomenon beyond the heat transfer area enlargement effect is likely to contribute to the enhanced critical heat flux by the increase of the groove size.

| Area (cm²) | Plane | No groove | 0.2 × 0.2 | 0.4 × 0.4 | 0.5 × 0.5 |
|------------|-------|-----------|-----------|-----------|-----------|
| R_{\text{area}} | 1.0 | 1.4 | 1.8 | 2.05 |
| CHF (W/cm²) | 138 | 228 | 242 | 269 | 461 |
| R_{\text{CHF}} | 1.0 | 1.06 | 1.18 | 2.02 |

Fig. 10  Boiling curves of lotus with the groove of 1.0 mm. CHF increases up to 534 W/cm² at the wall superheat of 111 K. This value is 3.71 times higher than CHF for the plane surface.
same as those for the 0.4 mm and 0.5 mm grooves. The results show that CHF increases up to 534 W/cm² at the maximum at the wall superheat of 111 K. This value is 3.71 times higher than that of the smooth surface. In addition, the CHF increasing rate ($R_{\text{CHF}} = 2.34$), which was discussed in Table 2, overcomes the heat transfer area enlargement rate ($R_{\text{area}} = 2.28$), which proves that larger size of groove definitely enhances the breathing effect (vapor discharge from the groove and liquid suction from the top of the lotus copper plate in this case) and increases the critical heat flux. Of course, there could be an optimal groove size for the CHF improvement, so that, in the next step, we intend to clarify the mechanism of critical heat flux improvement and the main factors to control the breathing phenomenon via pool boiling experiments with larger groove size, detailed visualization experiments, and CFD simulation.

4. Conclusions

In this study, we quantitatively evaluated several types of thermal resistances that manifest in thermal management of SiC-based on-board inverters with a high heat flux of 500 W/cm², and proposed that a mounting structure without a heat spreader and the direct cooling method utilizing two-phase immersion cooling technique as the key and breakthrough technology. Furthermore, it was proven that the two-phase immersion cooling technology using breathing effect by the lotus copper can realize this mounting structure and that very high critical heat flux of 534 W/cm² can be attained in saturated pool boiling at an atmospheric pressure. In addition, the critical heat flux and also the breathing phenomenon are highly affected by the groove size, as the critical heat flux increases with increasing groove size. Because this is still a fundamental test regarding the new topic of boiling heat transfer, in the future, it is necessary to conduct research having in mind the development of automotive cooling units as well as clarification of the mechanism of the breathing phenomenon and its control. The issues to be dealt with include: (1) vapor condensation technology in a closed container, (2) evaluation of heat transfer performance in multiple heat transfer surfaces, (3) heat transfer performance in low liquid level environment, (4) securing long-term soundness of boiling heat transfer surface, (5) gravitational effects, and (6) heat dissipation technology from cooling units that all require further investigations.

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