Evaluation of forced convective boiling heat transfer with layered parallel microchannels

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Abstract
In this study, we experimentally evaluated heat transfer performance using a configuration of layered parallel microchannels, with a focus on the effect of refrigerant mass flow rate on heat transfer performance. HFC-245fa was used as a refrigerant owing to its chemical stability and appropriate saturation pressure for the design of the cooling system under actual conditions. Experimental results showed that heat transfer rate and heat transfer coefficient increased with increasing refrigerant mass flow rate under the same superheat. However, the maximum heat transfer coefficient reached a certain value even as the mass flow rate increased. Further, on comparing the experimental results with numerical ones performed in this study, it was confirmed that the heat transfer performance with the configuration of layered parallel microchannels depended on the thermal resistance between each layer. Furthermore, to realize the maximum potential of the heat sink with the layered parallel microchannels, it was important to prevent a decrease in wall superheat at the channels on the far side from the heat source. Eventually, in the series of experiments, a heat flux of 3.04×106 W/m2 and a heat transfer coefficient of 5.20×104 W/(m2∙K) were reached at a pressure drop of 48.4 kPa under a mass flow rate of 3.33×10⁻² kg/s. This was achieved despite the use of a simple configuration for heat transfer enhancement and a refrigerant having poor latent heat dissipation.

Keywords : Boiling, Microchannel, Heat transfer enhancement, Fluorochemicals, Refrigerant, Phase change, Heat sink

1. Introduction
In general industry, power consumption of electronic devices, such as central processing units and insulated gate bipolar transistors, has been increasing year on year, although device size has been decreasing. The thermal issues caused by the increase in power density of these devices have become quite significant. Furthermore, this problem has restricted the development of mechatronic products, such as electric vehicles (EVs). To overcome this problem, a satisfactory high-performance cooling application using an appropriate refrigerant is required to enhance heat transfer. Furthermore, if the refrigerant needs to be chemically stable and have antifreeze properties, fluorochemical series should be applied, although their latent heat is poor compared with water, as shown in Table 1.
On the other hand, the extension of heat transfer surfaces and utilization of latent heat are well-known methods of dissipating high heat flux for convective boiling heat transfer. Cooling applications with microchannels have shown the potential to overcome this issue in previous studies (Bowers and Mudawar, 1994). However, for heat transfer enhancement with microchannels, more accurate phenomenological models need to be explored, and therefore, more fundamental experimental and analytical studies need to be conducted (Thome, 2004). In the last several decades, many studies on the enhancement of the heat transfer coefficient have been experimentally and numerically conducted (Mudawar, 2001), which improved cooling applications. However, in the automotive industry, electronic devices require heat transfer coefficient in the range of $10^4$–$10^5$ W/(m$^2$K) for acceptable cooling performance. If fluorochemicals are utilized as refrigerants in vehicles, a heat transfer structure must be evolved to compensate for the poor latent heat.

Practically, cooling devices need a simple structure. A few simply structured cooling devices that use parallel microchannels have been proposed (Breuer et al., 2003) (Vaidyanathan et al., 2005). However, in the case of large heat transfer, these devices need significant improvement.

In recent studies, heat sinks have exhibited high heat transfer performance. Recinella and Kandlikar (2018) achieved a heat flux of $6.18 \times 10^4$ W/m$^2$ at a wall superheat of 20.1 K and a pressure drop of 13.8 kPa with distilled water using radial open microchannels with a manifold and offset strip fins. Shultz et al. (2015) achieved a heat flux of approximately $3.5 \times 10^4$ W/m$^2$ with a flow rate of $4.2 \times 10^{-3}$ kg/s and a stable pressure drop of 320 kPa with a radial array of channels using the dielectric coolant R1234ze(E). This coolant was injected into the center and spread through the channels. Finally, the coolant exited from the faces of the testing surface. However, these structures are still complicated in actual cooling applications. On the other hand, there are studies that have achieved high heat transfer performance using microchannels under subcooled boiling conditions. Wang and Cheng (2009) achieved a heat flux of $14.41 \times 10^4$ W/m$^2$ at a mass flux of 883.8 kg/(m$^2$-s) with the occurrence micro emission boiling in microchannel. However, in this heat transfer mode, there is an issue of dissipating a high heat flux maintaining a low superheat.

Today, much human activity depends on the automotive industry. Hence, we focused on electric vehicles (EVs), such as battery EVs, hybrid EVs, and plug-in hybrid EVs. By 2030, the use of EVs is expected to increase by over 50%. This trend will make the development of electronic devices more important. In EVs, motor and motor control devices have important functions. Motor control devices, such as power control units (PCUs), consist of many electronic devices using a large amount of current. Hence, they need to be protected from thermal issues. In addition, the miniaturization of these devices is needed. Therefore, cooling devices are required to dissipate large heat fluxes while maintaining a low temperature difference between the device and refrigerant. Further, these devices need to be used under actual harsh conditions. From a practical perspective, several severe problems, such as freezing and chemical stability of the refrigerant, need to be addressed. Therefore, we focused on the dissipation of high heat flux using a reliable refrigerant by enhancing the heat transfer performance using fluorochemicals.

The objective of this study was to evaluate the boiling heat transfer in microchannels and to enhance heat transfer performance. Furthermore, we achieved a high heat transfer coefficient using a simple structure and an appropriate refrigerant for application in vehicles. In this study, we applied forced convective boiling heat transfer in layered parallel microchannels for effective use in three-dimensional space; the heat transfer area was almost equal to the heat source area. Fluorochemicals, which have chemical stability and a low boiling temperature, were used as refrigerants.

It is well-known that the heat transfer performance of a heat sink using microchannels depends on a mass flow rate (Bowers and Mudawar, 1994) (Mudawar and Bowers, 1999). Hence, the effect of a mass flow rate on heat transfer performance. Furthermore, we achieved a high heat transfer coefficient using a simple structure and an appropriate refrigerant for application in vehicles. In this study, we applied forced convective boiling heat transfer in layered parallel microchannels for effective use in three-dimensional space; the heat transfer area was almost equal to the heat source area. Fluorochemicals, which have chemical stability and a low boiling temperature, were used as refrigerants.

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### Table 1 Thermophysical properties under saturation conditions (Lemon et al., 2013)

| Fluid     | Temperature $T_{sat}$, °C | Pressure $p_{sat}$, kPa | Latent heat $i_{LV}$, kJ/kg |
|-----------|--------------------------|-------------------------|-----------------------------|
| HFC-245fa | 14.9                     | 100.7                   | $1.97 \times 10^2$          |
|           | 60.0                     | 462.5                   | $1.69 \times 10^2$          |
|           | 100.0                    | 1265                    | $1.36 \times 10^2$          |
| water     | 14.9                     | 1.695                   | $2.47 \times 10^3$          |
|           | 60.0                     | 19.95                   | $2.36 \times 10^3$          |
|           | 100.0                    | 101.4                   | $2.26 \times 10^3$          |


performance was evaluated in this study.

2. Methods and materials

To evaluate the heat transfer performance of our heat sink, a closed-loop flow circuit (Fig. 1) was set up using HFC-245fa as the refrigerant. The vapor flowed between the heating section and the condenser, and the liquid flowed in other connected tubes. To discharge the vapor effectively, microchannels were vertically arranged in parallel, and the refrigerant was pumped into the microchannel from the bottom to the top of the heat sink. At the heating section, the pressurized refrigerant was infused into the microchannels and then heated and boiled while receiving heat from the microchannel wall. The vapor was discharged from the heater section, and the residual refrigerant, which was not vaporized, returned to the pump via the sub-tank. At the condenser, the vapor was condensed and liquefied. Finally, the returned refrigerant mixed with the condensed refrigerant from the condenser in the sub-tank. This mixed liquid was pressurized again by the pump. To visualize the behavior of the discharged vapor from the microchannels, transparent polycarbonate was used as the boiler cover. It was confirmed that the vapor region in this system was under saturation conditions by measuring a temperature and a pressure at the exit of heating section. Thus, the experimental circuit was a saturated boiling cooling system.

From the viewpoint of application to a cooling device, it was important to dissipate a higher heat flux of a heat source while maintaining a lower temperature difference between the heat source and refrigerant. Therefore, we evaluated the relationship between the heat flux of the heat source and a superheat, which is similar to a boiling curve. Here, it should be noted that heat transfer characteristics using a heat sink with layered parallel microchannels highly depend on the shape and area of the heat source, unlike the case of general pool boiling, where the arrangement and size of microchannels are limited by these characteristics. In this study, the heat source size was to be 25.4 mm × 25.4 mm, which was taken into account in the context of an actual electric device size in an electric vehicle.

Details of the heating section are shown in Fig. 2. The cross-section of the heater block was 25.4 mm × 25.4 mm, which was the same for the aluminum heat sink. These dimensions were determined from the actual electronic device size of the PCU. 31 microchannels with diameters of 0.5 mm were bored for each layer and three layers were arranged in this heat sink as shown in Fig. 3. There were 93 microchannels in this heat sink.

The heat was applied from the cartridge heaters in the copper heater block and then transferred to the heat sink. The heat sink and aluminum plate were brazed, and the aluminum plate and heater block were soldered.

\[ q = k_c \frac{T_s - T_1}{L_{12}} \]  

(1) \[ \Delta T_{\text{sat}} = T_s - T_{\text{sat}} \]  

(2) \[ T_s = T_0 - \frac{qL_{\text{sat}}}{k_{\text{sol}}} \]  

(3)
\[ T_0 = T_1 - \frac{qL_{vol}}{k_c} = T_1 - \frac{L_{vol}}{L_{12}}(T_2 - T_1) \]  \hspace{1cm} (4)

Fig. 1 Schematic diagram of the experimental closed-flow loop using HFC-245fa as refrigerant.

Fig. 2 Schematic diagram of the heating section.
3. Experimental results

3.1 Evaluation of heat transfer performance

In this series of experiments, the heat sink with layered parallel microchannel dissipated a heat flux of \(3.04 \times 10^{6}\) W/m\(^2\), as shown in Fig. 4, and achieved a heat transfer coefficient of \(5.20 \times 10^{4}\) W/(m\(^2\)\(\cdot\)K), as shown in Fig. 5, at a pressure drop of 48.4 kPa, as shown in Fig. 6, under mass flow rate of \(3.33 \times 10^{-2}\) kg/s, using HFC-245fa as refrigerant. Large pressure drops in the microchannels are a well-known problem. Owing to the size limitation of the heat sink, which was equal to the heat source size, the microchannel length shortened, preventing the increasing pressure drop. A heat transfer coefficient \(h\) was estimated using Newton’s law of cooling, which is shown as:

\[
\Delta T = \frac{Q}{h A}
\]

where \(\Delta T\) is the temperature difference, \(Q\) is the heat flux, and \(A\) is the heat transfer area.
\[
h = \frac{q}{\Delta T_{\text{sat}}}. \tag{5}
\]

Although the measured heat flux approached \(3.04 \times 10^6\) W/m\(^2\), as shown in Fig. 4, the heat flux increased almost linearly. This indicated that maximum heat flux, which is the critical heat flux, was larger than \(3.04 \times 10^6\) W/m\(^2\).

In Fig. 4, the gradient of heat flux increased with the increasing superheat; however, in case of the mass flow rate of 0.50 \(\times 10^{-2}\) kg/s and 1.11 \(\times 10^{-2}\) kg/s, after superheat exceeded a certain value, the slope of the heat flux changed to decreasing.

In the entire superheat range, the heat flux increased with increasing mass flow rate of refrigerant, as shown in Fig. 7. However, when the mass flow rate exceeded a certain value, the improvement in heat flux due to increasing mass flow rate was reduced.

In addition, at all mass flow rate conditions, the pressure at the exit of heating section increased with increasing heat flux, ranging from 166 kPa to 356 kPa as shown in Fig. 8.

### 3.2 Heat leakage from an aluminum plate

Heat leakage from the aluminum plate between the microchannel block and heater block was estimated using Eq. (6), assuming the aluminum plate was an annular fin. Here, \(q_{\text{fin}}\) is the fin heat transfer rate; \(I_0\) and \(K_0\) are modified, zero-
order Bessel functions of the first and second kinds, respectively; \( I_1 \) and \( K_1 \) are modified, first-order Bessel functions of the first and second kinds, respectively; \( r_1 \) is the radius of an annular fin root, which was assumed to be the equivalent radius of the sectional area of the microchannel block; \( r_2 \) is the outside radius of the aluminum plate; \( t \) is the thickness of the aluminum plate; and \( T_\infty \) is temperature of ambient air. The schematic diagram of annular fin model is shown in Fig. 9. Here, the heat transfer coefficient between the aluminum plate and the ambient air was assumed to be 26 W/(m\(^2\)·K). The heat transfer coefficient between the aluminum plate and refrigerant was obtained using Kutateladze’s equation (Kutateladze, 1952) in the region where the heat flux was less than the critical heat flux obtained by Zuber’s equation, and using the critical heat flux divided by the superheat in the region where the heat flux was larger than the critical heat flux. Figure 10 shows the relationship between an applied heat transfer rate and a calculated heat leakage. As shown in Fig. 10, in the region of low heat transfer performance, heat leakage increased due to the increase in microchannel block temperature:

\[
q_{\text{fin}} = 2\pi k r_1 t \theta_b m \left[ \frac{I_1(m r_2) - I_1(m r_1)}{K_0(m r_2) + I_0(m r_2)} \right],
\]

\[
\theta_b = \frac{T_b - T_\infty}{m^2 = \frac{2h}{kt}}.
\]

3.3 Observation of outlet flow
At a heat flux of 1.2×10\(^6\) W/m\(^2\), a superheat of approximately 35 K, and a mass flow rate of 1.11×10\(^{-2}\) kg/s, the refrigerant of the two-phase state if from the channels near the heat source was not observed, as shown in Fig. 11. The
ejection of two-phase flow from the channels was unsteady. Figure 11 shows the phenomenon at a certain moment. The space on the left of the aluminum plate was filled with vapor. Therefore, we suggest that only vapor was discharged near the heat source. This phenomenon indicated that dryout occurred in the channels near the heat source. Under these conditions, the gradient of the heat transfer coefficient started to decrease, as shown in Fig. 5.

4. Discussion

A heat flux of \(3.04 \times 10^6\) W/m\(^2\), which was the maximum heat flux reached in this study as shown in Fig. 4, was acceptable in a simple configuration for heat transfer enhancement using a refrigerant with poor latent heat dissipation compared with the critical heat flux of water of \(1.1 \times 10^6\) W/m\(^2\), estimated using Zuber’s equation (Zuber, 1958). The maximum heat flux exceeded the HFC-245fa critical heat flux of \(2.2 \times 10^5\) W/m\(^2\) estimated using Zuber’s equation (Zuber, 1958).

A heat transfer coefficient of \(5.20 \times 10^4\) W/(m\(^2\)∙K), as shown in Fig. 5, was also comparable with results obtained in past studies (Mudawar, 2001). It was considered that high performance depended mainly on the arrangement of channels in the heat sink. Despite being confined to a limited area, due to the arrangement of the channels in the layers, the heat sink had a large effective heat transfer area. However, this performance was still not satisfactory for industrial applications. Hence, to further improve heat transfer performance, heat transfer at the layered parallel microchannel configuration under high heat flux should be closely analyzed.

4.1 Comparison between experimental and numerical value

To evaluate the heat transfer performance of boiling flow in microchannel, the numerical simulation is one of useful tools. Several works focused the bubble expansion process in microchannel rigorously (Okajima and Stephan, 2019); however, it is difficult to explore the heat transfer characteristics under various inlet and boundary condition exhaustively. Hence, to evaluate the boiling flow in microchannel heat sink, the model which can cover a wide range of condition is required. As shown in previous studies (Okajima et al., 2018, Okajima and Komiya, 2018), we calculated the heat transfer rate of the heat sink with the following assumptions and simplifications:

(i) Gravity force was negligible.
(ii) The flow of working fluid was homogeneous and one-dimensional. The velocities of both phases were identical.
(iii) Thermodynamic equilibrium was locally achieved. Hence, the vapor quality was identical to the thermodynamic equilibrium quality.
(iv) The average thermophysical properties of the two-phase flow were calculated based on the vapor quality.
Figure 12 shows the analysis configuration of this calculation. To simulate the microchannel as an element of the heat sink, an isothermal wall was assumed. The channel length and diameter were fixed at 25.4 mm and 0.5 mm, respectively, which were the actual heat sink channel dimensions. HFC-245fa was used as the working fluid. In the calculation, the experimental values of mass flow rate and pressure were applied.

Mass conservation and the first law of thermodynamics are written, respectively, in the general form as:

\[
\dot{m} = \rho u A = GA, \tag{7}
\]

\[
\Delta \left( e + \frac{p}{\rho} + \frac{1}{2} u^2 \right) = \frac{\dot{Q}(z)}{\dot{m}}. \tag{8}
\]

In Eq. (7), \( \dot{m} \) is mass flow rate; \( u \) is velocity; \( A \) is area; \( G \) is a mass flux, and in Eq. (8), \( e \) is internal energy; \( p \) is pressure; \( \dot{Q} \) is heat transfer rate; \( z \) is the coordinate. The heat transfer rate from the isothermal wall was calculated using the following equation:

\[
\dot{Q}(z) = h(z) \left[ T_w (z) - T_{\text{sat}} (z) \right] \pi D \Delta z. \tag{9}
\]

In Eq. (9), \( h \) is the heat transfer coefficient; \( T_w \) is wall temperature; \( D \) is a diameter of a microchannel. The heat transfer coefficient was calculated using Li and Wu’s correlation (2010), which is expressed as

\[
Nu_{TP} = 334Bl^{0.3} \left( Bo \cdot Re_L^{0.36} \right)^{0.4}, \tag{10}
\]

where,

\[
Nu_{TP} = \frac{hD}{k_L}, \quad Bl = \frac{q}{G_i_{L,V}}, \quad Bo = \frac{g \left( \rho_L - \rho_v \right) D^2}{\sigma}, \quad Re_L = \frac{G \left( 1 - x_{eq} \right) D}{\mu_L}. \tag{11}
\]

\[
x_{eq} = \frac{e - H_v}{H_v - H_L}
\]

Here, \( Nu_{TP} \) is a two-phase Nusselt number; \( k_L \) is liquid thermal conductivity; \( Bl \) is Boiling number; \( Bo \) is Bond number; \( x_{eq} \) is vapor quality. Vapor quality is the ratio of vapor mass-flow rate to total mass-flow rate, and estimated using the enthalpy of saturation vapor (\( H_v \)) and that of saturation liquid (\( H_L \)) as shown in Eq. (11). The frictional pressure drop in the microchannel was calculated using the separated flow model suggested by Lockhart and Martinelli (1949), which is expressed as:

\[
\Delta p_{TP} = \phi_L \Delta p_L + \phi_v \Delta p_v. \tag{12}
\]
Chisholm and Laird (1958) reinforced the theoretical background of Lockhart and Martinelli’s study by theoretically analyzing the two-phase pressure drop of annular flow. Chisholm and Laird derived the function form of $\Phi_L$ as:

$$\Phi_L = 1 + \frac{C}{X} + \frac{1}{X^2}.$$  \hspace{1cm} (13)

Here, $X$ is the Lockhart-Martinelli parameter. Lee and Mudawar (2005) also proposed a new parameter, $C$, using experimental data from the boiling flow in the microchannel:

$$C = \begin{cases} 2.16 Re_{LO}^{0.047} We_{LO}^{0.60} & \text{(Laminar liquid - Laminar vapor)} \\ 1.45 Re_{LO}^{0.25} We_{LO}^{0.23} & \text{(Laminar liquid - Turbulent vapor)} \end{cases} \hspace{1cm} (14)$$

where $Re_{LO}$ and $We_{LO}$ are the Reynolds and Weber numbers, respectively, assuming that the liquid phase flows alone, contributing to the total mass flux. The heat flux through the heat sink was calculated by integrating the local heat transfer rate, expressed as:

$$\dot{Q}_{\text{tot}} = N_{\text{tube}} \int_{0}^{L_{\text{tube}}} \dot{Q}(z) \, dz = \pi D N_{\text{tube}} \int_{0}^{L_{\text{tube}}} h(z) \left[ T_w - T_{\text{sat}}(z) \right] \, dz,$$  \hspace{1cm} (15)

$$q_{\text{heat sink}} = \frac{\dot{Q}_{\text{tot}}}{A},$$  \hspace{1cm} (16)

where wall temperature $T_w$ was assumed to be uniform as a heat sink. Here, $N_{\text{tube}}$ is the number of microchannels; $L_{\text{tube}}$ is the length of the microchannel.

![Graph](image1)

![Graph](image2)

![Graph](image3)

![Graph](image4)

Fig. 13 Comparison between experimental and calculated values for different mass flow rate.

The comparison between experimental and numerical values is shown in Fig. 13 (a)–(d). It was confirmed that
The experimental heat flux under the same superheat conditions was lower than the calculated value in the entire mass flow rate examined in this study. These calculated results were obtained on the assumption that all microchannel walls were at the same temperature; however, the comparative results suggested that this might not be actually the case.

4.2 Boiling heat transfer phenomena occurring in multi-layered microchannels

The scheme of heat transfer as shown in Fig. 14 was considered. When the heat transfer rate $Q_{\text{total}}$ was applied to the heat sink, the boiling flow from the first, second, and third layer channels caused a heat transfer rate of $Q_{\text{1st}}$, $Q_{\text{2nd}}$, and $Q_{\text{3rd}}$, respectively. Because thermal resistance existed between each layer channel, the wall temperature at the channel decreased in order of distance from the heat source. Therefore, the wall superheat at the channel far from the heat source was lower than the wall superheat at the near channel. Consequently, the relationship between the heat transfer rate of each layer is given as $Q_{\text{1st}} > Q_{\text{2nd}} > Q_{\text{3rd}}$. The calculated value, considering the uniform wall temperature, did not correspond with the experimental value and was higher than the experimental value. Hence, in the actual heat sink with layered microchannels, the entire microchannel set was unable to dissipate heat at the ideal transfer rate. This tendency increased with increasing heat flux.

In addition to heat dissipation, the applied heat transfer rate variation was considered. If the heat transfer rate applied to the heat sink was small and the heat transfer rate at the first and second layer channels was too large, the wall temperature of the third layer channel significantly decreased, i.e., the temperature difference between the channel wall and refrigerant decreased, suppressing the boiling in the third layer channel, as shown in Fig. 15 (a). At this time, the flow resistance in the third layer channel became smaller than those in the other channels, as boiling was suppressed and the mass flow rate increased. At the same time, in the first and second layer channels, the phenomenon in the third layer channel caused the mass flow rate and local performance of heat transfer to decrease. Eventually, these phenomena caused the heat transfer performance of the heat sink to decrease. On the contrary, had the above imbalance of flow rate existed, and the applied heat transfer rate were too large, dryout readily occurred easily at the first layer channel, as shown in Fig. 15 (b). This caused a deterioration of heat transfer performance. The visualized result as shown in Fig. 11 might demonstrate the case of the large applied heat transfer rate as shown in Fig. 15 (b).

Figure 16 shows the relationship between a superheat and vapor quality at the outlet of the microchannel obtained by the above calculations. As shown in Fig. 16, the calculated vapor quality increased with increasing superheat. This indicated that the increase in superheat intensified the boiling. Further, it was confirmed that the calculated vapor-quality decreased with increased mass flow rate. This indicated that the increase in mass flow rate caused the decrease in flow resistance. Therefore, the increase in the mass flow rate might have improved the above flow-resistance balance problem. This may be the reason that the difference between experimental and numerical values decreased with increasing mass flow rate. Furthermore, in Fig. 16, at a superheat of about 40 K under the mass flow rate of $0.50 \times 10^{-2}$ kg/s, the changing point of the vapor quality slope was confirmed, and in Fig. 4, at the same superheat, the changing point of heat flux slope.
was also confirmed. This indicated that the heat transfer performance decreased when vapor quality increased to near 1.0. Thus, it was also important for the heat transfer enhancement to control vapor quality so that it did not approach 1.0.

Assuming dissipation of heat flux under actual conditions, to prevent dryout in the first layer channel, some means need to be adopted, e.g., decreasing the channel diameter (Bowers and Mudawar, 1994, Mudawar and Bowers, 1999). Furthermore, to reach the maximum potential of the heat sink, the wall temperature of the far channel had to be prevented from decreasing using such methods as reduction of thermal resistance, reduction of the distance between each layer channel, and changing the heat sink material.

5. Conclusions

In this study, a heat sink configured as layered, parallel microchannels was evaluated for different mass flow rate using HFC-245fa as the refrigerant. We conclude the following:

1. A heat flux of $3.04 \times 10^6 \text{ W/m}^2$ and a heat transfer coefficient of $5.20 \times 10^4 \text{ W/(m}^2\text{K)}$ was achieved at a pressure drop of 48.4 kPa and a heating-section pressure of $3.56 \times 10^5 \text{ kPa}$ under a refrigerant mass flow rate of $3.33 \times 10^{-2} \text{ kg/s}$ with layered parallel microchannels using fluorochemicals of HFC-245fa as a refrigerant. Despite the poor latent heat of the fluorochemicals, this heat flux exceeded the critical heat flux of water, estimated using Zuber’s equation.

2. The prevention of a temperature decrease at layer channels far from the heat source may effectively contribute to improving the flow resistance balance between each layer channel. Consequently, this may affect the heat transfer performance of the entire set of microchannels.

3. The layered parallel microchannel configuration could enhance the heat transfer performance.

To achieve maximum potential of the heat sink, we explored the precise predictive methods of the phenomena in the microchannels under several sets of conditions and established appropriate heat transfer balance methods between each channel.
References

Bowers, M. B. and Mudawar, I., High flux boiling in low flow rate, low pressure drop mini-channel and micro-channel heat sink, International Journal of Heat and Mass Transfer, Vol.37, No.2(1994), pp.321–332.

Breuer, N., Leuthner, S., Hohl, R. and Satzger, P., Cooling device for cooling components of the power electronics, said device comprising a micro-heat exchanger, United States, Publication, No.: US 2003/0178178 A1(2003).

Chisholm, D. and Laird, A. D. K., Two-phase flow in rough tubes, Transactions of the American Society of Mechanical Engineers, Vol.80, No.2(1958), pp.276–286.

Kutateladze, S. S., Heat transfer in condensation and boiling, 2nd edition, Mashgiz (1952); AEC translation 3770, U.S. AEC Tech. Info. Service.

Lee, J. and Mudawar, I., Two-phase flow in high-heat-flux micro-channel heat sink for refrigeration cooling applications: Part I—pressure drop characteristics, International Journal of Heat and Mass Transfer, Vol.48, No.5(2005), pp.928–940.

Lemon, E. W., Bell, I. H., Huber, M. L. and McLinden, M. O., NIST standard reference database 23: Reference fluid thermodynamic and transport properties-REFPROP, version 9.1, NIST standard reference database (2013).

Li, W. and Wu, Z., A general correlation for evaporative heat transfer in micro/mini-channels, International Journal of Heat and Mass Transfer, Vol.53, No.9–10(2010), pp.1778–1787.

Lockhart, R. W. and Martinelli, R. C., Proposed correlation of data for isothermal two-phase, two-component flow in pipes, Chemical Engineering Progress, Vol.45, No.1(1949), pp.39–48.

Mudawar, I., Assessment of high-heat-flux thermal management schemes, IEEE Transactions on Components and Packaging Technologies, Vol.24, No.2(2001), pp.122–141.

Mudawar, I. and Bowers, M. B., Ultra-high critical heat flux (CHF) for subcooled water flow boiling –I: CHF data and parametric effects for small diameter tubes, International Journal of Heat and Mass Transfer, Vol.42, No.8(1999), pp.1405–1428.

Okajima, J., Jeong, S. and Maruyama, S., Evaluation of cooling performance of ultrafine cryoprobes: Effect of probe structure on thermodynamic properties of refrigerant, International Journal of Air-Conditioning and Refrigeration, Vol.26, No.2(2018), pp.1850020-1–1850020-11.

Okajima, J. and Komiya, A., Numerical evaluation of boiling heat transfer performance in isothermal microchannels, Proceedings of the 11th Australasian Heat and Mass Transfer Conference, AHMTC11(2018), 11AHMTC18–22.

Okajima, J. and Stephan, P., Numerical simulation of liquid film formation and its heat transfer through vapor bubble expansion in a microchannel, International Journal of Heat and Mass Transfer, Vol.136(2019), pp.1241–1249.

Recinella, A. and Kandlikar, S. G., Enhanced flow boiling using radial open microchannels with manifold and offset strip fins, Journal of Heat Transfer, Vol.140(2018), pp.021502-1–021502-9.

Schultz, M., Yang, F., Colgan, E., Polastre, R., Dang, B., Tsang, C., Gaynes, M., Parida, P., Knickerbocker, J. and Chainer, T., Embedded two-phase cooling of large 3D compatible chips with radial channels, ASME Paper, No. IPACK2015-48348 (2015).

Thome, J. R., Boiling in microchannels: a review of experiment and theory, International Journal of Heat and Fluid Flow, Vol.25, No.2(2004), pp.128–139.

Vaidyanathan, K. R., Ortega, A., Platero, M., Skandakumaran, P and Bower, C., Method of manufacturing microchannel heat transfer exchangers, United States, Patent, No.: US 7,360,309 B2(2008).

Wang, G. and Cheng, P, Subcooled flow boiling and microbubble emission boiling phenomena in a partially heated microchannel, International Journal of Heat and Mass Transfer, Vol.52, No.1–2(2009), pp.79–91.

Zuber, N., On the stability of boiling heat transfer, Transactions of the American Society of Mechanical Engineers, Vol.80(1958), pp.711–720.

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