Experimental evaluation of a compact two-phase cooling system for high heat flux electronic packages

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HIGHLIGHTS

• The system can dissipate over 380 W/cm\textsuperscript{2} while keeping chip temperature at 90 °C.
• Heat flux that the system could dissipate decreases with heat source area.
• The downstream super heater can improve the thermal performance.
• Increasing differential pressure can improve the heat dissipation capacity.

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Thermal test vehicle
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ABSTRACT

In this work, an experimental study on the aluminum plate fin evaporator based on a compact two-phase cooling system for high heat flux electronic packages is presented. Single-chip and multi-chip wire-bonded thermal test vehicles (TTVs) were fabricated and assembled in the PCB grooves designed to emulate high heat flux sources. The issue of heat dissipation was addressed by applying the evaporator to the TTVs, respectively, to evaluate their thermal characteristics. It is found that, the evaporator system could dissipate over 380 W/cm\textsuperscript{2} for the TTV1 while maintaining its temperature at about 90 °C. As the effective heat source area and thermal design power (TDP) increased, the maximum heat flux that the system could dissipate decreased given the same chip temperature rise. Furthermore, the addition of a second evaporator and heat source following the main evaporator, increased the dissipation of the system. As a result, an increase of 48 W/cm\textsuperscript{2} in heat removal capacity was observed in our test system. Finally, the effect of the differential pressure between the condenser and the evaporator was investigated. The increase in the differential pressure could improve the heat dissipation capacity of the two-phase cooling system. The temperature of the TTV2 dropped by 19 °C when the differential pressure increased by 2.7 bar. It can be concluded that the compact two-phase cooling system is a promising solution for removing heat from high heat flux electronic packages.

1. Introduction

With the ever-increasing need on power density of power semiconductors and power electronic systems, high heat flux is the development trend of electronic packages. For instance, it was projected that heat flux of insulated gate bipolar transistors for hybrid electric vehicles would be as high as 500 W/cm\textsuperscript{2}, leading to an increment of more than 3 times in comparison with the current level of 100–150 W/cm\textsuperscript{2} [1–4]. With the advent of wide bandgap (WBG) semiconductors such as silicon carbide (SiC) and gallium nitride devices, heat fluxes can be even higher as these devices tend to shrink to smaller form factor compared to their silicon counterparts with the same voltage blocking capability [5]. The demanding heat flux makes thermal management of electronic packages much more challenging than ever before.

Conventional cooling solutions for high heat flux electronic packages, e.g. heat sink [6–8], heat pipe [9], vapor chamber [10,11], and single-phase liquid cooling [12–14], have limited cooling capabilities. For example, vapor chamber in combination with forced

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Single-phase liquid cooling could reach 180 W/cm² [16,17]. Recently, micro-channels, etc. [23,24]. Among them, plate immersion pool boiling and liquid metal have been reported for high heat flux electronic packages. C. Falsetti et al. [2] presented an experimental investigation of flow boiling of R134a in a micro-pin fin heat sink to examine the operational maps, pressure drop, and heat transfer performances. The study showed that R134a exhibited a stable flow boiling behavior under a wide range of flow conditions. The heat transfer performance was improved significantly with the applied heat, whereas mass flux exhibited a minor influence. K. P. Drummond et al. [28] designed a hierarchical manifold microchannel heat sink array for intra-chip evaporative cooling with a dielectric fluid. The micro-channels were embedded into the heated substrate to reduce parasitic thermal resistances. Experimental results showed that the maximum heat flux dissipation increased with mass flux and channel depth.

In this work, an aluminum plate fin evaporator based two-phase cooling system for high heat flux electronic packages was fabricated, assembled, and experimentally examined. The cooling system mainly consisted of two evaporators with test chips, a condenser, a miniature compressor, a throttling device, and flowmeter. The thermal test chips (TTCs) were assembled in the printed circuit board (PCB) groove, wire-bonded to the PCB and protected with encapsulating material to emulate the high heat flux heat sources. Both the single-chip TTV1 and the

<table>
<thead>
<tr>
<th>Nomenclature</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>convective area ($m^2$)</td>
</tr>
<tr>
<td>$A_{\text{TTC}}$</td>
<td>area of TTC ($m^2$)</td>
</tr>
<tr>
<td>$h$</td>
<td>convective heat transfer coefficient ($W/m^2°C$)</td>
</tr>
<tr>
<td>$\Delta H$</td>
<td>total enthalpy change of R134a in the two evaporators and compressor (kJ/kg)</td>
</tr>
<tr>
<td>$\Delta H_1$</td>
<td>enthalpy change of fluid R134a in the first evaporator (kJ/kg)</td>
</tr>
<tr>
<td>$\Delta H_2$</td>
<td>enthalpy change of unevaporated R134a in the second evaporator (kJ/kg)</td>
</tr>
<tr>
<td>$\Delta H_3$</td>
<td>enthalpy change of R134a in the compressor (kJ/kg)</td>
</tr>
<tr>
<td>$I_{\text{TTV}}$</td>
<td>current applied to the TTV (A)</td>
</tr>
<tr>
<td>$k_{\text{TTC}}$</td>
<td>thermal conductivity of TTC ($W/m°C$)</td>
</tr>
<tr>
<td>$P$</td>
<td>pressure (bar)</td>
</tr>
<tr>
<td>$P_s$</td>
<td>saturation pressure (kPa)</td>
</tr>
<tr>
<td>$P_{TDP}$</td>
<td>TDP of a TTV applied to the evaporator (W)</td>
</tr>
<tr>
<td>$q^e$</td>
<td>heat flux of a TTV applied to the evaporator ($W/m^2$)</td>
</tr>
<tr>
<td>$R_1$</td>
<td>thermal resistance from junction to bottom ambient ($°C/W$)</td>
</tr>
<tr>
<td>$R_2$</td>
<td>thermal resistance of junction to top ambient ($°C/W$)</td>
</tr>
<tr>
<td>$R_{\text{UCA}}$</td>
<td>conduction resistance of UV curing adhesive ($°C/W$)</td>
</tr>
<tr>
<td>$R_{\text{NC}}$</td>
<td>natural convection resistance ($°C/W$)</td>
</tr>
<tr>
<td>$R_{\text{TTC}}$</td>
<td>conduction resistance of TTC ($°C/W$)</td>
</tr>
<tr>
<td>$R_{\text{TIM}}$</td>
<td>conduction resistance of TIM ($°C/W$)</td>
</tr>
<tr>
<td>$R_{\text{EVA}}$</td>
<td>thermal resistance of evaporator ($°C/W$)</td>
</tr>
<tr>
<td>$R_{\text{CON}}$</td>
<td>thermal resistance of condenser ($°C/W$)</td>
</tr>
<tr>
<td>$R_{\text{TV}}$</td>
<td>electrical resistance of TTV ($Ω$)</td>
</tr>
<tr>
<td>$t_{\text{TTC}}$</td>
<td>thickness of TTC (m)</td>
</tr>
</tbody>
</table>

$T_s$ | saturation temperature ($°C$) |
$U$ | voltage (V) |

**Subscripts**
- Con: condenser
- EB: evaporator bottom
- Eva: evaporator
- NC: natural convection
- $s$: saturation
- TB: TTV bottom
- UCA: ultraviolet curing adhesive

**Abbreviations**
- GWP: Global warming potential
- PCB: Printed circuit board
- PCM: Phase change material
- PH: High-pressure
- PL: Low-pressure
- PMMA: Polymethyl methacrylate
- SIC: Silicon carbide
- TDP: Thermal design power
- TTC: Thermal test chip
- TTV: Thermal test vehicle
- UV: Ultraviolet
- WBG: Wide bandgap
multi-chip TT2 were evaluated in the two-phase cooling system, respectively. The TTC surface was directly attached onto the bottom center surface of the evaporator, in between filled with a layer of thermal interface material (TIM) with high thermal conductivity. Moreover, the effects of a downstream super heater and differential pressure between the evaporator and the condenser on heat dissipation performance of the system were investigated. The potential of the two-phase cooling scheme was also discussed.

2. Two-phase cooling system

2.1. Description of cooling system

Fig. 1 shows schematic diagram of a two-phase cooling system for high heat flux electronic packages. The two-phase cooling system mainly consists of two evaporators, a miniature compressor, a condenser, a throttling device, and a flowmeter. The two evaporators and flowmeter are in the low-pressure (PL) region while the condenser lies in the high-pressure (PH) region. The two regions are separated by a miniature compressor and a throttling device. The miniature compressor can increase pressure of the refrigerant, while the throttling device can be adjusted to examine the effect of the differential pressure on the thermal performance.

In order to accurately analyze the phase change process in the two-phase cooling system, data acquisition of temperature and pressure in the system were conducted. Thermocouples T1-T6 were used to monitor temperature at different locations of the system. Two voltage output pressure transducers (PX309-200 G5V) from OMEGA were adopted to detect voltage of PH region (5) and PL region (1), respectively. In Ref. [29], experimental results showed the pressure changes in the evaporators were negligible compared to those in the throttling device and miniature compressor, so the two pressure transducers provided all the important data needed to assess system performance. The absolute pressure range of the transducer is 0–14 bar, and the static accuracy is 0.25% maximum. Fig. 2 displays the calibration curve of the pressure transducer. It is found that the pressure \( P \) is almost proportional to the voltage \( U \). The correlation can be fitted linearly as follows.

\[
P = 2.32 \times U + 0.565
\]  

(1)

(1) Evaporator

The evaporator is a component through which the fluid refrigerant absorbs heat from the hot chip and boils during the flow process. Compared with single-phase cooling solution, much more heat can be removed when the refrigerant works in the flow boiling regime in the evaporator. In order to ensure boiling heat transfer occurs in the evaporator, full advantage of latent heat during evaporation of the fluid refrigerant can be taken, and the refrigerant that enters the miniature compressor is all vapor, multiple evaporators can be designed and connected in series. Complete evaporation of refrigerant could occur at the last evaporator outlet.

In this work, two \( 50 \times 50 \times 9 \text{ mm}^3 \) aluminum plate fin evaporators were designed and fabricated, whose thermal resistance was about 0.04 °C/W. The evaporator was a highly compact and lightweight evaporator. Its interior structure, of which the overall size is \( 30 \times 30 \times 7 \text{ mm}^3 \), is shown in Fig. 3. 30 plate fins with a dimension of \( 20 \times 0.5 \times 6 \text{ mm}^3 \) are placed with a channel of 1 mm. Originally, in order to observe the vaporization phenomena of refrigerant, the evaporator was sealed by a visible polymethyl methacrylate (PMMA) plate using plastic sealing ring and glue. However, when heat flux of heat source increased to about 200 W/cm², the PMMA plate was damaged due to huge energy absorption during vaporization process of the refrigerant. In order to withstand higher vapor pressure, the chamber is sealed by an aluminum plate through bolt connection and glue sealing. The evaporator has an aluminum tube inlet and an aluminum tube outlet. The inner and outer diameter of the aluminum tube is 5.35 mm and 6.35 mm, respectively. The size is the same as that of copper tube, which facilitates the connection to the standard copper tubes with straight connectors in the system.

(2) Compressor

The compressor is a core component in the two-phase cooling system. It can generate a pressure rise in the refrigerant and drive the flow of refrigerant in the system. In this work, an Aspen miniature rotary compressor (14-24-000X) was used in the system, which was driven by a 24 V DC power unit. The miniature compressor is the smallest and lightest among commercially available compressors as far as we know. Its suction volume is 1.4 cm³, and the maximum rotational speed is up to 6500 RPM. The miniature compressor is originally designed for R134a refrigerant compression to be used in thermal management of high-power electronics [30].

Compared with the pump, the compressor not only can move the fluid, but also increase the potential energy of fluid by compressing it in a closed system, thus control phase change of refrigerant. The rotational speed of the compressor was adjusted through a variable resistor, which can be used to adjust the voltage to the compressor. Through adjusting the compressor, the pressure of refrigerant will reach its peak value. Meanwhile, the temperature of refrigerant also goes up with pressure.

(3) Condenser

The condenser is an opposite component with evaporator. When refrigerant passes through it, heat absorbed in the evaporators and compressor can be dissipated to the ambient. In this work, an aluminum mini-channel heat exchanger was adopted, which was fabricated with the mini end milling process. In order to decrease the convection resistance of the condenser, a fan was applied onto the condenser.

(4) Throttling device

The throttling device is an important component in the two-phase cooling system, where temperature and pressure of the refrigerant can be reduced drastically. The maximum operating pressure is 343 bar at 37 °C. The decrease in the opening of throttling valve can reduce the pressure of the refrigerant to a low level such that vaporization of the refrigerant occurs by absorbing a large amount of heat. In other words,
the differential pressure of refrigerant between the condenser and the evaporator can be varied by adjusting the opening of the throttling device besides the adjustment of the compressor.

(5) Flowmeter

The flowmeter is an optional component. The digital mass flow meter (FMA-4312) from OMEGA is used to monitor flow rate of the refrigerant. The normal flowrate is 10 L/min.

(6) Refrigerant

With the selection of refrigerant material, boiling point of refrigerant should be low enough at low pressure to ensure the boiling in the evaporator but high enough at high pressure for liquefaction in the condenser to dissipate the heat to the ambient [23]. In this work, low boiling point refrigerant R134a was selected as the working fluid and filled into the closed loop. R134a is a chlorine-free and non-ozone-depleting hydrofluorocarbon refrigerant. The ozone depletion potential is 0. It has relatively low global warming potential (GWP) of 1430, compared to other refrigerants such as R-12 with GWP of 10900. It flows from the two evaporators, the flowmeter and the compressor to the condenser and comes back to the first evaporator again through the throttling device. In both the first evaporator and condenser, phase changes of refrigerant R134a occur. The liquid R134a evaporates to vapor in the evaporators. The vapor R134a changes to liquid in the condenser. Through adjustment of throttling device, the pressure and temperature of R134a decrease to a lower value, at which it easily vaporizes and a large amount of heat is removed. Fig. 4 displays fitted curve of saturation temperature of R134a dependent on pressure. The fit function is expressed in Eq. (2).

\[
T_s = -53.2 + 43.6 \times \left(1 - e^{-\frac{P_s}{1959.3}}\right) + 121.3 \times \left(1 - e^{-\frac{P_s}{159.8}}\right)
\]

where \(T_s\) is the saturation temperature, \(P_s\) is the saturation pressure of R134a. From the figure, it can be seen the saturation temperature of R134a rises with pressure. Saturation temperature of R134a rises quickly at lower pressures. As the pressure keeps increasing, the saturation temperature rise begins to become slow and levels off. The saturation temperature of R134a is about \(-26.5^\circ\text{C}\) at one bar. When the pressure reaches 10 bar, its saturation temperature is about \(39.5^\circ\text{C}\).

2.2. Thermodynamic analysis of cooling system

Fig. 5 shows states of R134a at different stages in its pressure-enthalpy diagram. 1~5 in the figure is corresponding to the locations as marked in Fig. 1. The diagram was divided into three regions by saturated liquid and saturated vapor curves: subcooled liquid, liquid-vapor mixture, and superheated vapor.

At low pressure, the fluid refrigerant in the first evaporator can remove majority of heat generated by the high heat flux heat source.
The bubble grows and departs at a high frequency, and bubble flow and slug flow are dominant in the channels between the plate fin of the first evaporator, in which the fluid refrigerant changes into two-phase mixtures (process 1–2). Then, the mixtures flow into the second evaporator and continues absorbing heat and vaporizes fully. The liquid–vapor mixtures change into vapor. In this evaporator, slug flow and annular flow dominates (process 2–3). Thirdly, the superheated vapor from the second evaporator outlet enters the compressor through flowmeter and is compressed to a higher vapor pressure, resulting in a higher temperature as well (process 3–4). Fourthly, the compressed vapor flows into the condenser and heat transferred in previous processes (process 1–4) is dissipated to the ambient (process 4–5). Lastly, the subcooled liquid refrigerant returns into the first evaporator through the throttling device where pressure drops drastically (process 5–1).

From the figure, a large amount of heat is removed in the two evaporators (process 1–3), and the fluid refrigerant fully evaporates into the vapor. The total enthalpy change \( \Delta H \) of R134a in the two evaporators and compressor can be expressed by

\[
\Delta H = \Delta H_1 + \Delta H_2 + \Delta H_3
\]

where \( \Delta H_1 \) is the enthalpy change of fluid R134a in the first evaporator, \( \Delta H_2 \) is the enthalpy change of unevaporated R134a in the second evaporator, and \( \Delta H_3 \) is the enthalpy change of R134a in the compressor.

3. Thermal test vehicles

In order to evaluate the heat dissipation capacity of the two-phase cooling system, wire-bond thermal test chips (TTCs) from Thermal Engineering Associates Inc. are used to emulate the high heat-flux heat source. TTC is a semiconductor device that contains one or more resistors and one or more diodes [31]. Electric layout of the TTC is shown in Fig. 6(a). The size of TTC used in the work was 2.5 × 2.5 × 0.625 mm³, which had two resistors and four diodes. Two resistors cover more than 85% of the chip. One diode is located in the chip center, and two diodes are located at the opposing corners and one at the middle of the chip edge. The resistance of each resistor is about 7.6 Ω.

In order to make TTC work normally and reduce thermal resistance from chip junction to evaporator, a single-chip wire-bonded TTV (TTV1) assembled in the PCB groove with the same thickness was presented and fabricated. A wire-bond TTC was firstly placed in the groove of a PCB. Then, two resistors in the TTC were connected in series and bonded to the PCB through gold wires. The total electric resistance of the TTV1 is about 15.2 Ω. Thirdly, the wire-bond TTC was encapsulated by ultraviolet (UV) curing adhesive to protect from damage.

Lastly, to achieve better heat dissipation effect and reduce the thermal contact resistance between the heat source and the evaporator, the TTV1 was directly attached onto the bottom center surface of the first evaporator through a layer of TIM with high thermal conductivity. The structure schematic of the TTV1 assembled on the evaporator is shown in Fig. 6(b).

In order to study the effect of heat dissipation area, wire-bond TTCs in a 2 × 2 array were packaged together to form a multi-chip TTV (TTV2). The pitch of the TTC is 0. The schematic of the multi-chip TTV2 assembled on the evaporator is depicted in Fig. 6(c). Four wire-bond TTCs were firstly placed in the groove of a PCB. Then, eight resistors of the four TTCs were connected in series and bonded to the PCB with gold wires. The total resistance of TTV2 is about 60.8 Ω. The area of four TTCs is about 0.25 cm², in comparison with the single TTC with the area of 0.0625 cm². When current of 1 A is applied on the TTV, the total TDP is about 60.8 W, and the heat flux will achieve about 243.2 W/cm².

To ensure the refrigerant vaporizes completely in the evaporator and to take full advantage of latent heat of refrigerant, two evaporators were connected by an aluminum tube through vacuum brazing technique, and two identical multi-chip TTVs were attached onto the bottom surface of the first and the second evaporator, respectively, as shown in Fig. 6(d). The second multi-chip TTV (TTV3) was used as a downstream super heater. After passing through the two evaporators, vaporization of refrigerant is expected to be complete at the outlet of the second evaporator. The refrigerant that flows into the compressor is all superheated vapor, so that the compressor cannot be damaged by

![Structure schematic of wire-bonded TTVs assembled on the evaporator](image-url)
liquid refrigerant.

In order to accurately evaluate the heat dissipation of TTVs, the electric resistances of TTVs using Digit Multimeter from Keysight were measured, and TDP of a TTV applied to the evaporator \( P_{TDP} \) is calculated as

\[
P_{TDP} = \frac{I_{TTV}^2}{R_{TTV}}
\]

(4)

Correspondingly, the heat flux of the TTV \( q \) is expressed below,

\[
q = \frac{P_{TDP}}{A_{TTC}}
\]

(5)

where \( I_{TTV} \) is the current applied to the TTV, \( R_{TTV} \) is the electrical resistance of the TTV, and \( A_{TTC} \) is the area of the TTC.

Fig. 7 shows a compact thermal resistance network of a single-chip TTV assembled on the evaporator. Among them, the \( R_{UCA}, R_{TIM}, R_{EVA} \) are the conduction thermal resistance which can be obtained based on the dimensions of the part and its thermal conductivity. For example, the \( R_{TTC} \) can be calculated as Eq. (6).

\[
R_{TTC} = \frac{t_{TTC}}{k_{TTC} \cdot A_{TTC}}
\]

(6)

The natural convection resistance \( R_{NC} \) is expressed as:

\[
R_{NC} = \frac{1}{h \cdot A}
\]

(7)

It can be seen from Fig. 7, the thermal resistance \( R_1 \) from junction to bottom ambient is given below

\[
R_1 = R_{UCA} + R_{NC}
\]

(8)

The thermal resistance \( R_2 \) from junction to top ambient is expressed as

\[
R_2 = R_{TIM} + R_{EVA} + R_{CON}
\]

(9)

where \( t_{TTC} \) is the thickness of TTC, \( k_{TTC} \) is the thermal conductivity of TTC, \( A_{TTC} \) is the cross-sectional area of TTC, \( h \) is the convective heat transfer coefficient, and \( A \) is the convective area, \( R_{UCA} \) is the conduction resistance of UV curing adhesive, \( R_{NC} \) is the natural convection resistance, \( R_{TIM} \) is the conduction resistance of TTC, \( R_{EVA} \) is the thermal resistance of evaporator, and \( R_{CON} \) is the thermal resistance of condenser.

According to Eqs. (6)-(9), \( R_1 \) is approximately 33600 °C/W which corresponds to a specific thermal resistance of \( R_1^\prime = 2100 \, \text{°C} \cdot \text{cm}^2/\text{W} \) based on the TTC area, while \( R_2 \) is only about 3.2 °C/W or \( R_2^\prime = 0.2 \, \text{°C} \cdot \text{cm}^2/\text{W} \). As a result, the heat loss to the ambient from the chip is negligibly small and neglected in the present analysis.

4. Experimental evaluation

4.1. Description of measurement methods

Fig. 8 illustrates experimental apparatus of a compact two-phase cooling system. The system was fixed on an aluminum positioning board. Two identical aluminum evaporators with aluminum tubes were connected by copper tube fittings. Other components were connected through copper tube and fittings.

Before evaluating the heat dissipation capacity of the two-phase cooling system, the sealing performance of the system was checked and improved. Vacuum pump was firstly used to remove the gas within the system. Then, liquid nitrogen was filled into the system and experiment on leakage was performed. For the purpose of real-time monitoring the sealing performance of the system, pressures at the throttling device inlet and outlet tested by two voltage output pressure transducers, were connected to a digit multimeter. Voltages of the high- and low-pressure regions could be monitored in real time. When the voltages of the two regions did not change, indicating the system was well sealed without observable leakage.

In the following, the refrigerant R134a was filled into the closed loop. Six high precision type K thermocouples (T1-T6) were attached at different locations to measure the temperature changes for better understanding of temperature distribution of the system. Location of thermocouples T1-T6 are shown in Fig. 1 and Table 1. Thermocouples T1, T2, T5, and T6 were installed at the copper tube outer walls, on which measured temperature was higher than those of refrigerant in the inner walls. Thermocouple T3 was mounted onto the top surface center of TTV on the first evaporator to monitor the temperature change of TTV. Thermocouple T4 was attached onto the top surface center of the first evaporator or top surface center of TTV on the second evaporator. Before testing, thermocouples signals were calibrated and reference temperature of thermocouples was set to be 20 °C. A high resolution data acquisition (DAQ) card with maximum sixteen channels was applied to acquire the thermocouple temperatures [32]. Six channels were utilized in this work. The DAQ card was connected to a computer and LabVIEW was used to acquire the temperatures of the six areas in real time.

4.2. Thermal evaluation of a single-chip wire-bonded TTV1

In this section, the high heat flux single-chip TTV1 were assessed in this work. The measured electrical resistance of the TTV1 was 15.2 Ω through Digit Multimeter. The area of the heat source was only 0.065 cm², which was much smaller than the effective area of evaporator of 9 cm². The backside of TTC was directly attached onto the bottom surface center of the first evaporator through high thermal conductivity TIM. One type K thermocouple (T3) was mounted onto the top surface of UV curing adhesive. A 72 V/1.2 A DC power supply was adopted to provide the required power to the TTV1. By applying the current range from 0.8 A to 1.25 A, the TDP of the TTV1 increased from 9.73 W to 23.8 W, and consequentially, its heat flux rose from 155.6 W/cm² to 380 W/cm².

Fig. 9 shows the pressures of high- and low-pressure regions dependent on heat flux of TTV1. The pressure of R134a at the condenser

Fig. 7. A compact thermal resistance network of a single-chip TTV assembled on the evaporator.
outlet (high-pressure region) maintained about 7 bar, which was high enough for R134a liquefaction in the condenser to dissipate the heat to the ambient. After passing through the throttling device, the pressure of R134a at the evaporator inlet (low-pressure region) dropped to about 3.6 bar. Combining Eq. (2) with Fig. 4, the saturation temperature of R134a was about 6.17 °C at the pressure, which was low enough in the evaporator chamber to guarantee the boiling of refrigerant again.

Fig. 10 illustrates the temperature at different locations of the system dependent on heat flux of TTV1. It can be seen from the figure, the temperature at other locations almost kept unchanged except for that of TTV1. The temperature of TTV1 increased linearly with its heat flux approximately. When the heat flux of TTV1 reached 380 W/cm², the temperature of TTV1 was about 90 °C, which is below 125 °C that is allowable for normal operation of silicon-based electronic devices. The temperature at the TCON_OUT maintained about 25.4 °C, which was close to the ambient temperature, which means that the largest portion of heat generated by TTV1 and further heat accumulation due to vapor compression were dissipated into the ambient in the condenser. Therefore, for high heat-flux single-chip TTV1, the system can dissipate over 380 W/cm² while maintaining the TTV1 temperature at about 90 °C.

4.3. Thermal evaluation of a multi-chip wire-bonded TTV2

In this section, the high heat flux and high TDP multi-chip TTV2 were evaluated. The measured electrical resistance of the TTV2 was 60.8 Ω through Digit Multimeter. The effective heat source area is 0.25 cm², which is 4 times of single-chip TTV. By applying the current from 0.6 A to 0.95 A, the TDP of the TTV2 increased from 21.9 W to 54.9 W, corresponding to the heat flux from 87.6 W/cm² to 219.5 W/cm².

Fig. 11 illustrates the pressures of high- and low-pressure regions dependent on heat flux of TTV2. From the figure, it can be seen the pressure of high-pressure region was about 7 bar. Through throttling device, the pressure of R134a at the evaporator inlet dropped to 3.3 bar. From Eq. (2), it can be concluded that the saturation temperature of R134a was about 3.67 °C at the pressure.

Fig. 12 shows the temperature at different locations of the system dependent on heat flux of TTV2. As shown in the figure, as heat flux of TTV2 increased, the temperature at the TCON_OUT rose slightly and was still close to the ambient temperature, so most of heat transferred from the evaporator and compressor was dissipated into the ambient in the condenser. The temperature of TTV2 increased linearly with its heat flux approximately. When the heat flux of TTV2 was only about 185 W/cm², its temperature reached about 90 °C. Compared with the system with a single-chip TTV, the heat dissipation capacity of the system was weakened given the same chip temperature rise. It is also found that the temperatures at the TTH1, TTV2, TTEVA1,TOP or TTTV3, TTEVA2,OUT increased evidently. Therefore, as effective heat source area and TDP increase, the maximum heat flux that the system can dissipate decreases given the same chip temperature rise.

Table 1

<table>
<thead>
<tr>
<th>Thermocouple</th>
<th>Location</th>
<th>Denotes</th>
</tr>
</thead>
<tbody>
<tr>
<td>T1</td>
<td>Between the condenser outlet and the throttling device inlet</td>
<td>TCON.OUT</td>
</tr>
<tr>
<td>T2</td>
<td>Between the throttling device outlet and the first evaporator inlet</td>
<td>THR.OUT</td>
</tr>
<tr>
<td>T3</td>
<td>Top surface center of TTV on the first evaporator</td>
<td>TTV1, TTV2</td>
</tr>
<tr>
<td>T4</td>
<td>Top surface center of the first evaporator or top surface center of TTV on the second evaporator</td>
<td>TTEVA1,TOP or TTTV3</td>
</tr>
<tr>
<td>T5</td>
<td>Between the second evaporator outlet and flowmeter inlet</td>
<td>TTEVA2,OUT</td>
</tr>
<tr>
<td>T6</td>
<td>Between the compressor outlet and the condenser inlet</td>
<td>TCON.IN</td>
</tr>
</tbody>
</table>
_kept 155.6 W/cm\(^2\). Pressure regions slowly increased with heat super heater TT\(V_2\). As suggested by the system with one downstream super heater dependent on heat flux of R134a is about 35.3 °C at the pressure.

of R134a is changed to liquid.

The temperature at the \(T_{\text{CON,OUT}}\) increased slowly with the heat flux of the TT\(V_2\). When heat flux of the TT\(V_2\) reached 294.3 W/cm\(^2\), the temperature at the \(T_{\text{CON,OUT}}\) was about 31.9 °C, which was still lower than the saturating temperature of R134a was about 35.3 °C at the pressure of 9.1 bar. Therefore, after passing through the condenser, R134a is changed to liquid.

For the downstream super heater, when the heat flux increased from 119.2 W/cm\(^2\) to 155.6 W/cm\(^2\), the temperature of the TT\(V_3\) rose from 55.4 °C to 77.5 °C. The temperature of the downstream super heater kept on rising with the heat flux of the TT\(V_2\). It rose from 77.5 °C to 97.7 °C. It is also found that the temperature at the \(T_{\text{EV2,OUT}}\) gradually increased and stabilized at 31.5 °C. Compared with the system with a multi-chip TT\(V\), the temperature was higher than 7.4 °C, which indicated liquid-to-vapor conversion rate of refrigerant was higher and the heat dissipation potential of the two-phase was improved. And therefore, heat dissipation capacity was greatly improved by applying the downstream super heater on the second evaporator, and heat flux increased by about 48 W/cm\(^2\) while maintaining the TT\(V_2\) temperature at about 90 °C. The downstream super heater can not only improve the thermal performance of the two-phase cooling system, but also protect the compressor since the refrigerant that flows the compressor is all superheated vapor.

4.5. Effect of differential pressure on the heat dissipation capacity of the system

In order to further improve the heat dissipation capacity of the two-phase cooling system, the effect of differential pressure between the condenser and the evaporator on the heat dissipation performance of the system was studied. The differential pressure was adjusted by throttling device and variable resistor of compressor. The differential pressure between the condenser and the evaporator increased from 3.2 bar to 6.2 bar. Heat flux of the TT\(V_2\) was kept as 294.3 W/cm\(^2\) while that of the TT\(V_3\) was firstly kept as 155.6 W/cm\(^2\), then increased to 243.2 W/cm\(^2\).

The temperature at different locations of the system with one downstream super heater depend on differential pressure between the condenser and the evaporator. As can be seen from the figure, the temperatures of the TT\(V_2\) and the TT\(V_3\) decreased evidently after increasing differential pressure between condenser and evaporator, with a decrement of about 19 °C and 18 °C, respectively. The

4.4. Effect of a downstream super heater on the heat dissipation performance of the system

Unlike the placement of thermocouple T4 in previous two experiments, thermocouple T4 was mounted onto the top surface of the second TT\(V\) (TT\(V_3\)) sample on the second evaporator to monitor its temperature changes. The TT\(V_3\) was adopted as a downstream super heater. The measured resistance was also 60.8 Ω. Two 72 V/1.2 A DC power supplies were used to drive the two TT\(V\)s, respectively. By applying current swings from 0.7 A to 1.1 A and 0.7 A to 0.8 A on the TT\(V_2\) and TT\(V_3\), respectively, the total TDP of the two TT\(V\)s increased from 59.6 W to 112.5 W, the heat flux of the TT\(V_2\) increased from 119.2 W/cm\(^2\) to 294.3 W/cm\(^2\), and the heat flux of the downstream super heater firstly increased from 119.2 W/cm\(^2\) to 155.6 W/cm\(^2\), then kept 155.6 W/cm\(^2\).

Fig. 13 illustrates the pressure of high- and low-pressure regions of the system with one downstream super heater dependent on heat flux of TT\(V_2\). It can be seen from the figure the pressure of high- and low-pressure regions slowly increased with heat flux of TT\(V_2\). When heat flux of TT\(V_2\) reached about 294.3 W/cm\(^2\), the pressure of high-pressure region rose to 9.1 bar. According to Eq. (2), the saturating temperature of R134a is about 35.3 °C at the pressure.

Fig. 14 illustrates the temperature at different locations of the system with one downstream super heater dependent on heat flux of TT\(V_2\). As suggested by the figure, the temperature of TT\(V_2\) increased linearly with its heat flux approximately. Compared with the system with one multi-chip TT\(V\), heat dissipation capacity of the two-phase system with one additional downstream super heater was improved. The heat flux of the TT\(V_2\) was increased to 233 W/cm\(^2\) while maintaining its temperature at about 90 °C.

![Fig. 11. Pressures of high- and low-pressure regions dependent on heat flux of TT\(V_2\).](image1)

![Fig. 12. Temperature at different locations of the system dependent on heat flux of TT\(V_2\).](image2)

![Fig. 13. Pressure of high- and low-pressure regions of the system with one downstream super heater dependent on heat flux of TT\(V_2\).](image3)
The TTV3 is increased to 243.2 W/cm², the highest temperature of the amounts of heat generated by high heat dissipation capacity of the two-phase cooling system. The temperature of the TTV3 increased with its heat flux. When heat flux of the TTV3 is increased to 243.2 W/cm², the highest temperature of the TTV3 is about 121 °C. And thus, increasing the differential pressure between the condenser and the evaporator is helpful for improving the heat dissipation capacity of the two-phase cooling system.

4.6. Discussion on heat transfer enhancement

According to the experimental results in this work, far greater amounts of heat generated by high heat flux devices can be removed in the evaporator and dissipated into the ambient in comparison with the 180 W/cm² typically for single phase cooling. Due to the limitation of test chip and maximum operating pressure of the system, the maximum heat dissipation capacity of the two-phase boiling system was not analyzed. It is expected that the system can dissipate high heat-flux over 500 W/cm² while maintaining the chip temperature below 120 °C. The heat dissipation capacity of the two-phase cooling scheme can be further improved by introducing a downstream super heater, enlarging differential pressure between the condenser and the evaporator.

More work can be conducted based on the present two-phase cooling system. The effects of evaporator structure factors on the thermal performance of the two-phase cooling system will be optimized. The sub-cooling and over-heating of the refrigerant, low GWP refrigerant, influence of the flow rate on the heat transfer performance will be investigated in the future work.

The SiC device as a promising WBG technology has many advantages, such as higher breakdown voltage, faster switching speed, higher operation temperature compared to silicon counterparts [33–35]. Although SiC devices can operate theoretically at high temperatures up to 600 °C thanks to wide bandgap energy and low intrinsic carrier concentrations of the material, the highest junction temperature of commercial SiC power modules are confined to 175 °C at this moment, limited mainly by the suitable package materials [33]. Most of package materials adopted in the SiC power module, such as die attach and encapsulation materials, cannot withstand temperature beyond 175 °C for long time. Moreover, power loss of SiC power module increases with switching frequency and its heat flux is higher due to smaller form factor compared to silicon counterparts. Consequently, thermal management of SiC power module is more challenging. Two-phase cooling scheme developed in this work has great potential for solving its heat dissipation issue while maintaining the chip temperature at a low temperature. When power loss and heat flux of a SiC power module at their peak values, the SiC chip temperature is still below 125 °C.

5. Conclusions

In this work, an aluminum plate fin evaporator based two-phase cooling system for high heat flux power electronics was presented. Wire-bonded TTCs assembled in the PCB groove were presented and fabricated to emulate the high heat flux heat sources. The TTC was directly attached onto the bottom center surface of the evaporator through a layer of TIM with high thermal conductivity. Heat dissipations of high heat flux single-chip TT1V and high heat flux and high TDP multi-chip TTV2 were evaluated in the two-phase cooling system, respectively. In order to improve heat dissipation capacity of the system, effects of a downstream super heater and differential pressure on the heat dissipation capacity of the system were investigated. Some conclusions are drawn as follows.

1. For high heat-flux single-chip TT1V, the designed system can dissipate over 380 W/cm² while maintaining its temperature at about 90 °C.
2. As effective heat source area and TDP increase, the maximum heat flux that the system could dissipate decreases given the same chip temperature rise. For the multi-chip TTV2, the maximum heat flux that the system can dissipate is only 185 W/cm² while its temperature has reached about 90 °C.
3. Heat dissipation capacity of the two-phase cooling system is greatly improved by applying one heat source on the second evaporator as a downstream super heater. Compared with the system with only one TTV2, and heat flux that the system can dissipate increases by about 48 W/cm². The downstream super heater can not only improve the thermal performance of the two-phase cooling system, but also protect the compressor since the refrigerant that flows the compressor is all superheated vapor.
4. Increasing the differential pressure between the condenser and the evaporator can improve the heat dissipation capacity of the two-phase cooling system. When differential pressure between the condenser and the evaporator increases to about 5.9 bar, the temperatures of the both TTVs drop by 19 °C.

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