Experimental Investigation on the Heat Transfer Characteristics of Multi-Point Heating Microchannels for Simulating Solar Cell Cooling

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Abstract: Concentrating photovoltaic power generation technology is a highly efficient way of utilizing solar energy resources with the efficiency limited by cell cooling conditions. For the heat dissipation problem from multi-point solar cell cooling, a microchannel heat sink is used to resolve the issue. Ammonia is chosen as the working fluid and two diamond microchannel heat sinks in series for the 16 simulated solar cells cooling with typical size. The heat sink consists of 31 triangular microchannels, each with a hydraulic diameter of 237 \( \mu \)m and a flow path length of 40 mm. It is experimentally verified that the diamond microchannel heat sink has excellent multi-point heat source heat dissipation capability. The surface temperature of the heat source can be controlled below 65.9 \( ^\circ \)C under a heat flux of 351.5 W/cm\(^2\), and the maximum temperature difference between the multi-point heat sources is only 1.4 \( ^\circ \)C. The effects of heat flux, mass flux and inlet state on the flow boiling heat transfer capacity within the series heat sinks were investigated and the ranges of the operating conditions are as follows: heat flux 90.8–351.5 W/cm\(^2\), mass flux 108–611 kg/(m\(^2\)s), saturation temperature 15–23 \( ^\circ \)C and inlet temperature 15–21 \( ^\circ \)C. The results show that within the range of experimental conditions, the flow boiling heat transfer capacity of the series heat sink increases with the increase of heat flux and is less influenced by the mass flux, showing the typical two-phase heat transfer characteristics dominated by the nucleation boiling mechanism. Between the upstream and downstream heat sinks, the thermal resistance of the upstream heat sink is larger and the temperature uniformity of the downstream heat sink is poor because of the difference of the inlet state.

Keywords: solar cell cooling; multiple heat sources; microchannel; flow boiling; series heat sink

1. Introduction

With global energy resources becoming increasingly constrained and the use of traditional fossil energy sources causing a series of environmental problems, the development and efficient use of solar energy as one of the important renewable resources is of great importance to the national economy \[?\]. As one of the important renewable resources, the development and efficient use of solar energy for the national economy has a very important role. Concentrating photovoltaic (CPV) power generation technology is a highly efficient way of utilizing solar energy resources. The development of CPV technology as an efficient way to utilize solar energy resources has attracted much attention \[? \]. Since the 1970s, concentrating solar cells have made great progress and many researchers have conducted research on different types of concentrating solar cells, including hetero structured solar cells, concentrating silicon cells and multi-junction concentrating...
cells. At present, the conversion efficiency of concentrating solar cells has reached 44% [? ]. However, due to the forbidden band width of the cells, only a part of the solar energy is converted into electrical energy; most of it is absorbed by the cell and eventually becomes thermal energy, resulting in a significant increase in the cell junction temperature. If suitable cooling measures are not taken, the high junction temperature will increase significantly to reduce cell efficiency and shorten life. Research data show that the efficiency of solar crystalline silicon cells is almost linearly and inversely related to temperature, with the photovoltaic conversion efficiency decreasing by approximately 0.4% for every 1 °C rise in cell temperature [? ]. Agyekum et al. [? ? ] adopted active and passive cooling mechanisms to improve the electrical output of a PV module such as phase change material and humidifier, which are both helpful to reduce the temperature of the panel and improve the electrical efficiency. In addition, for every increase in cell temperature after the cell reaches its upper operating temperature limit, the aging rate of crystalline silicon cells doubles for every 10 °C increase [? ? ? ? ]. Therefore, cooling of solar cells under concentrated light conditions has become a key technology.

High concentrated photovoltaic (HCPV) systems can be divided into three categories according to the concentration spot, such as individual cell, columns of cells and large area cells. For individual spot cells, traditional passive cooling can be used, such as air flow cooling with fins to expand around. However, for line spots and dense array cell systems, active cooling is necessary depending on their structural characteristics, including jet impingement, microchannels and phase-change cooling. The main work on cell cooling techniques has been reviewed and evaluated by Royne [? ], and these cooling methods have heat transfer coefficients above $1 \times 10^4 \text{W/}(\text{m}^2\cdot\text{°C})$, which meet the performance requirements of cooling devices for dense-array cell systems with high light concentration ratios.

In recent years, a number of new cooling devices have been developed with the objective of cooling dense arrays of cells. For example, Seepana et al. [? ] adopted a passive cooling approach using a discontinuous aluminum heat sink for a solar photovoltaic module and presented the effectiveness of low temperature. Royne et al. [? ] used jet impingement cooling and experimentally verified a number of different jet aperture designs, with an estimated heat transfer coefficient of $3.7 \times 10^4 \text{W/}(\text{m}^2\cdot\text{°C})$ for this jet impingement cooling device. Due to the highly effective performance of microchannel, Barrau et al. [? ? ? ] tested a combined jet-microchannel cooling device optimized for temperature uniformity. Due to the promising application of microchannel flow boiling heat dissipation technology, many researchers have been conducting experimental studies. Kuznetsov et al. [? ] experimentally investigated the flow boiling heat transfer characteristics of R134a in microchannels. The results showed that the heat flux has a significant effect on the heat transfer coefficient, which indicated that nucleate boiling is the dominant mechanism of heat transfer, but this dominant effect is suppressed at high vapor quality. Yang et al. [? ] carried out an experimental study of the saturated flow boiling heat transfer of ammonia using a triangular microchannel heat sink and visualized the flow pattern under different vapor qualities. Huang et al. [? ? ] investigated the effect of different parameters on the heat transfer coefficient of flow boiling in microchannels using R1233zd(E), R245fa and R236fa.

In practical heat dissipation applications, multiple solar cells are assembled on the substrate and are characterized by multiple hot spots and non-uniform heat flow distribution, posing a more severe challenge to the cell cooling system. Bogojevic et al. [? ] experimentally investigated the flow boiling instability of deionized water in microchannels under non-uniform heat flow boundary conditions. It was found that non-uniform heating conditions can affect temperature and pressure fluctuations, and non-uniform heat flow can lead to severe flow unevenness with different vaporization levels in each flow channel, which severely affects the temperature distribution. To solve the problem of heat dissipation from high heat flow hot spots, the DARPA ICECool project [? ] selected diamond material to prepare a microchannel heat sink, which can effectively reduce the hot spot temperature
and enhance temperature uniformity by using the extremely high thermal conductivity and heat expansion properties of diamond. Yang et al. [1] designed a diamond microchannel heat sink for the problem of heat dissipation from multi-point heat sources, which can dissipate heat from four heat sources simultaneously and maintain the stable operation of the heat dissipation system at the heat flux of 1000 W/cm², demonstrating that the high thermal conductivity diamond microchannel heat sink is an effective means to solve the multi-point heat source heat dissipation problem. For spatially dispersed solar cells, multiple heat sinks are needed to dissipate heat from multiple heat sources. For such a multiple heat source heat dissipation problem, Hong et al. [2] conducted an experimental study using two microchannel heat sinks in series, and the wall temperature of the downstream heat sink decreased slightly with the increase of the upstream heat sink heating. The increase of the inlet vapor quality of the downstream heat sink results in a slight increase in heat transfer capacity. Zhang et al. [3] found that because of manufacturing tolerances, impurity deposits and other factors, there are differences between the flow resistances of the parallel heat sinks, resulting in uneven flow distribution. Therefore, there is a certain temperature difference between the walls of two heat sinks, which is not conducive to the temperature uniformity of devices. He et al. [4] compared the flow boiling heat transfer performance of two heat sinks in series and in parallel, and the experimental comparison found that the wall temperatures of series heat sinks are basically the same and slightly lower than those in a parallel condition.

In this work, an experimental study of flow boiling within multiple heat sinks under non-uniform heat flow boundary conditions is carried out based on the above studies of microchannels. Two diamond microchannel heat sinks are connected in series using ammonia as a refrigerant to dissipate heat for 16 simulated heat sources with typical solar cell sizes. The heat dissipation capability of the series microchannel heat sink was evaluated in the working condition range of mass flow rate 108–611 kg/(m²·s) and heat flux 90.8–351.5 W/cm², using the heat source temperature, thermal resistance and temperature uniformity as indicators, and the effects of working conditions parameters such as mass flux, heat flux and inlet state on the flow boiling heat transfer performance within the series heat sink were analyzed.

2. Experimental Setup

2.1. Experimental System

As shown in Figure ??, a pumped two-phase fluid loop experimental system is built and the main components are pump, reservoir, regenerator, preheater, series microchannel heat sink, sight glass, condenser, water cooling unit and filter. To simulate the solar cells array, a multi-point heater is adopted with eight heaters on one microchannel heat sink.

As an inexpensive, renewable and environmentally friendly chemical with zero ozone depletion potential (ODP) and zero global warming potential (GWP), ammonia is chosen as the working fluid for the working environment without people because of its toxicity and peculiar odor. Most notably, ammonia has favorable thermodynamic properties as a refrigerant. Unlike water, which freezes below 0 °C, ammonia has a wide temperature range of phase transition (between −78 °C and 132 °C), preventing the risk of freezing in cold regions. Moreover, the latent heat of ammonia is large, which is 6.5 times that of R134a at 20 °C, indicating the potential to transfer high heat power with a limited flow rate. The dP/dT value of ammonia is also large, which is 1.55 times that of R134a at 20 °C, meaning a stronger capacity to maintain temperature uniformity under the same pressure drop in the flow process. In conclusion, ammonia is a promising medium temperature refrigerant in heat dissipation applications.

Driven by a gear pump, ammonia is heated by the regenerator and preheater, enters two series microchannel heat sinks and vaporizes, is then cooled to subcooled liquid by the regenerator and condenser, and flows to the pump to complete the next cycle. The line color represents the state of fluid, such as subcooled state of blue, near-saturated or two-phase fluid state of red. The use of the regenerator reduces the heat consumption of the preheater
and the condenser, thereby improving the energy efficiency of the system. The pipes in the system loop are made of stainless steel, and the components and pipes are wrapped with rubber insulation material to achieve thermal insulation from the environment.

Figure 1. Schematic diagram of pumped fluid loop system for solar cooling.

This experimental system is able to precisely control the mass flux, system pressure, heat sink inlet temperature and heat source heat flux. The gear pump used in this study is equipped with a brushless 24 V DC motor with integrated speed controller. Therefore, by gradually adjusting the pump speed using an external control analog signal, the flow rate can be gradually changed until a desired flow rate is precisely obtained with the adjustment accuracy of 0.1 L/h. The role of the reservoir is to control the system pressure by arranging the heating plates and semiconductor cooling plates on the reservoir, and the precise control of the temperature and pressure inside the reservoir is realized. The temperature of the working fluid at the inlet of the evaporator can be controlled by the preheater. The heat source of the evaporator is heated by the Agilent 6675A DC power supply, and the heat flux of the heat source can be controlled by adjusting the supply voltage.

In the data acquisition system, there are pressure sensors installed in the inlet and outlet of the heat sinks and pump, with a model of WP262-8S1BN-A whose range is 0–35 bar and measurement accuracy is ±0.1%. Temperature sensors Pt1000 are installed in the inlet and outlet of the main components with a measurement accuracy of ±0.3 °C, and the real-time monitoring and recording is realized by Panasonic FPXC38ATPLC. An ultrasonic flowmeter FLUXUS F601 is installed in the pipeline between the pump outlet and the preheater inlet, with a measurement accuracy of ±1%.

2.2. Test Section

As shown in Figure ??a, the test section of the experimental system consists of heat sinks and simulating multi-point heat sources. The upstream #1 heat sink is connected in series with the downstream #2 heat sink. The heat sink is prepared by welding a diamond microchannel cold plate, a metal housing and an upper cover plate. The cross-sectional area of the simulated heat source is gradually reduced along the heat conduction direction to obtain high heat flux boundary conditions on the upper surface to simulate the conditions
of high heat flux solar cells. There are eight heating surfaces on the upper surface of the simulated heat source for one microchannel. The distribution of multi-point heat sources at the bottom surface of the cold plate is shown in Figure ??c. Each point heat source has a typical cell size (2 mm × 5 mm), with 10 mm horizontal spacing and 2 mm vertical spacing between point heat sources. For each heat sink, eight T-type thermocouples are installed below the heating surface, and all thermocouples are produced in the same batch. These thermocouples have been calibrated by metrological institute. The interface between the heating surface of the heat source and the bottom surface of the cold plate is filled with liquid metal to reduce the interface thermal resistance. Each heat sink with simulated heat source is placed inside a thermal insulation box wrapped with thermal insulation material to reduce the heat leakage, especially natural convection and radiation to the external environment. The thermal conductivity of the rubber insulation material is 0.34 W/(m·K).

Figure 2. Schematic diagram of the test zone. (a) Test area structure. (b) Microchannel size. (c) Heat sources distribution.

Figure ??b shows the dimensions of one microchannel unit of the diamond cold plate. The preparation process of the diamond microchannel cold plate is as follows: a high-quality diamond film with thermal conductivity up to 1500 W/(m·K) was obtained by the plasma jet CVD method for a deposition period of 400–500 h. The diamond cold plate was obtained by grinding, with dimensions of 40 mm in length, 20 mm in width and 3.5 mm in thickness. Then 31 parallel triangular microchannels were laser etched on the surface of the cold plate, with a slot width of 250 μm, a slot height of 2.4 mm and a substrate thickness of 1.1 mm. The hydrodynamic diameter of the microchannels dₙ is 237 μm and the height-to-width ratio is 9.6. The size of the diamond microchannel was illustrated using an optical microscope (OM, Zeiss) from different view directions, as shown in Figure ??a,b. Measured by DEKTAK150 surface profiler, the arithmetic average deviation value (roughness) of the profile of the microchannel inner wall is determined as 1422 nm. The measured surface profile curve over a sample length of 1000 μm is shown in Figure ??c.
where $\delta$ is the distance between the measuring point and the upper surface of the heat source. $T$ is the temperature of the heat source surface ($T_i$ ~ $T_O$). The temperature difference of the square columns of the heat source ($T_i$ ~ $T_O$) can be deduced from the measured temperatures using the one-dimensional thermal conductivity formula.

$T_{h,i} = T_i - \frac{q\delta}{\lambda}$  \hspace{1cm} (4)

$T_a = \frac{\sum_{i=1}^{8} T_{h,i}}{8}$  \hspace{1cm} (5)

where $\delta$ is the distance between the measuring point and the upper surface of the heat source with unit of m and $\lambda$ is the thermal conductivity of copper W/(m·K).

Figure 3. The details of the microchannel structure. (a) Top-view OM photography of the diamond microchannel. (b) Cross-section OM photography of the diamond microchannel. (c) Surface roughness of the microchannel inner wall.

3. Data reduction and Uncertainty Analysis

3.1. Data Reduction

Considering that there is heat leakage into the environment and the surrounding pipelines in the heat transfer process from the heat source to the working fluid, a set of single-phase flow heat transfer experiments were implemented for two heat sinks, respectively, to measure the heat transfer efficiency of the heat sink before conducting the two-phase flow experiments in this paper. In the single-phase flow experiment, the actual absorbed heat load of the working fluid $Q_{eff}$ is:

$$Q_{eff} = \dot{m}c_{pl}(T_{out} - T_{in})$$  \hspace{1cm} (1)

where $\dot{m}$ is the mass flux of the working fluid with unit of g/s, $c_{pl}$ is the specific heat capacity of liquid ammonia with unit of J/(g·K) and $T_{in}$ and $T_{out}$ are the inlet and outlet temperatures of the heat sink, respectively, with a unit of °C.

The heat transfer efficiency of the heat sink $\alpha$ is:

$$\alpha = \frac{Q_{eff}}{Q_i}$$  \hspace{1cm} (2)

where $Q_i$ is the total heat load imposed on the heat sink with a unit of W.

Based on the results of the single-phase flow heat transfer experiment, the heat transfer efficiency of the heat sink $\alpha$ can be obtained, which can be used in the following two-phase flow boiling experiment to calculate the heat flux of the heat source.

$$q = \frac{\alpha Q_i}{A_h}$$  \hspace{1cm} (3)

where $A_h$ is the direct heating area of the simulated heat source with unit of m$^2$. There are eight raised square columns at the top of the simulated heat source and the temperature measuring points are arranged at the bottom center of each square column. The upper surface temperatures of the square columns of the heat source ($T_{h,1}$ ~ $T_{h,8}$) and the average temperature of the heat source surface ($T_a$) can be deduced from the measured temperatures ($T_i$ ~ $T_O$) using the one-dimensional thermal conductivity formula.

$$T_{h,i} = T_i - \frac{q\delta}{\lambda}$$  \hspace{1cm} (4)

$$T_a = \frac{\sum_{i=1}^{8} T_{h,i}}{8}$$  \hspace{1cm} (5)
Under the condition of uniform heating, the wall temperature can be calculated based on the one-dimensional thermal conductivity equation, and thus the two-phase heat transfer coefficient at the solid-fluid interface can be obtained. In contrast, under the condition of non-uniform heating, this calculation method is no longer applicable because of the lateral diffusion of the heat flux at the base of the cold plate, and therefore the heat transfer coefficient cannot be calculated. Most studies under non-uniform heating conditions directly used the measured temperature as an indicator of the cooling performance of the microchannel cold plate. In multi-point heat dissipation problems, temperature uniformity is another important indicator to consider the heat exchanger capacity, and, in this paper, the difference between the maximum and minimum values of the surface temperature of the multi-point heat source, $\Delta T_{\text{max}}$, is calculated to characterize the surface temperature uniformity of the heat source, and the higher the temperature uniformity, the smaller the value of $\Delta T_{\text{max}}$.

$$\Delta T_{\text{max}} = \max(T_{h,i}) - \min(T_{h,i})$$

To avoid the occurrence of outliers such as extremely different values and their deteriorating effect on the accuracy of temperature uniformity calculation results, the readings of all thermocouples were observed and analyzed during the entire experiment process. Before turning on the heat source and after turning off, the readings show that the difference between the maximum and minimum temperature is less than 0.1 $^\circ$C when a steady state flow is reached. During the experiment, the readings show that the distribution of local temperatures is reasonable and no outlier value is observed. Therefore, Equation (6) can be adopted to calculate $\Delta T_{\text{max}}$ as an index of temperature uniformity.

The total thermal resistance $R_a$ from the surface of the heat source to the working fluid inside the heat sink contains three layers of thermal resistance: the contact thermal resistance of the liquid metal layer between the heat source and the heat sink, the thermal resistance of the diamond substrate, and the two-phase heat transfer resistance at the solid-fluid interface. The changes of working parameters such as heat flux, mass flux and inlet temperature cannot change the contact thermal resistance and thermal conductivity of the diamond substrate, but only affect the two-phase heat transfer resistance at the solid-fluid interface. Therefore, the change of $R_a$ intuitively reflects the change of the two-phase heat transfer coefficient at the solid-fluid interface, and the increase in $R_a$ indicates the decrease in the two-phase heat transfer coefficient.

$$R_a = \frac{T_h - T_f}{Q_{\text{eff}}}$$

where $T_f$ is the qualitative temperature of the working fluid with a unit of $^\circ$C.

In this paper, the heat source temperature $T_h$, the maximum temperature difference $\Delta T_{\text{max}}$ and the total thermal resistance $R_a$ are used as indicators to evaluate the heat transfer performance of the microchannel heat sink.

3.2. Uncertainty Analysis

The uncertainties of the experimental parameters are listed in Table 2. For the direct measurement parameters, the uncertainties can be derived from the error calibration of the measuring instrument [1]. For the indirect measurement parameter $R$, the uncertainties can be found by the error calculation method proposed by Moffat [2], where $R$ is expressed as a function of several unrelated direct measurement parameters $x_i$, then the uncertainty of $R$, that is $\delta R$, can be obtained from the uncertainties of these parameters $\delta x_i$:

$$R = f(x_1, x_2, \ldots, x_n)$$

$$\delta R = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial R}{\partial x_i} \delta x_i \right)^2}$$
Table 1. Uncertainty of parameters.

| Direct Measurement Parameters     | Uncertainty   | Indirect Measurement Parameters               | Uncertainty |
|-----------------------------------|---------------|-----------------------------------------------|-------------|
| Pressure, P/kPa                   | ±0.1%         | Heat flux of heat source, q/W/cm²             | ±2.9%       |
| Temperature, T/°C                 | ±0.3 °C       | Surface temperature of heat source, T_R/°C    | ±1.4%       |
| Volume flux, V/m³/s               | ±1%           | Surface temperature difference of heat source, ΔT_max/°C | ±2.0%       |
| Hydraulic diameter, d_h/mm        | ±0.005 mm     | Thermal resistance, R_a/K/W                   | ±3.3%       |

4. Results and Discussion

4.1. Heat Transfer Efficiency

Due to the heat transfer efficiency being difficult to directly measure or calculate in the two-phase flow experiment, we have to conduct a single-phase flow experiment to estimate the heat transfer efficiency α under different heat loads. Single-phase flow experiments were conducted for two heat sinks with mass flux of 1620 kg/(m²·s) and electric powers of 40 W, 72 W and 112 W, and the heat transfer efficiencies α were obtained by Equation (2) for each operating condition. Mass flux G refers to the mass flow rate per unit flow area of the microchannels. Figure ?? shows the results of the single-phase flow heat transfer experiments. The horizontal coordinate is the electric power and the vertical coordinate is the heat transfer efficiency, i.e., the ratio of the effective heat absorption of the working fluid to the electric power. The upstream heat sink is denoted as the #1 heat sink and the downstream heat sink is denoted as the #2 heat sink. It can be seen that the heat transfer efficiencies of the two heat sinks range from 90.9% to 97.3%, and the heat transfer efficiency of heat sink varies with the heat load increasing.

![Figure 4. Heat transfer efficiency.](image_url)

In single-phase and two-phase experiments, each heat sink with heat source is placed inside an insulation box wrapped with thermal insulation material, which can reduce heat leakage through natural convection and radiation heat transfer to reduce the difference in the heat transfer efficiency for different flow regimes to some extent. Therefore, based on the results obtained in single-phase flow experiments, interpolation method is employed to estimate the effective heat transfer under different heat loads for two-phase flow experiments, and this calculation method is widely used in many literatures related to flow boiling [??].

4.2. Effect of Heat Flux

To investigate the effect of heat flux on the heat dissipation performance of the series heat sink, a set of flow boiling experiments were carried out in this section. In this set of experiments, the inlet temperature of #1 heat sink T_in, the outlet saturation temperature of #2 heat sink T_sat, and the mass flux G were kept constant, and the heat flux of all point heat sources were continuously adjusted. Figure ?? shows the trend of the surface temperature...
and temperature uniformity of the multi-point heat source on each heat sink during the process of changing heat flux from 90.8 to 351.5 W/cm² under the working conditions of mass flux of 335 kg/(m²·s), inlet temperature of #1 heat sink at 15 °C and outlet saturation temperature of #2 heat sink at 15 °C.

Figure 5. Heat dissipation performance of heat sink at different heat fluxes. (a) #1 heat sink (b) #2 heat sink.

In Figure ??a, the hot spot temperature of #1 heat sink increases with the increase of heat flux, and maintains at a low temperature level. The surface temperature of the heat source is maintained below 65.9 °C when the heat flux of the heat source is 351.5 W/cm², corresponding to the thermal resistance of 0.172 K/W and unit thermal resistance of 0.138 (K·cm²)/W, which is 1.31 times that obtained in this study. The lower thermal resistance is due to the fact that the microchannel cold plate material used in this study is high thermal conductivity diamond, which not only can effectively reduce the hot spot temperature, but also can improve the temperature uniformity of the multi-point heat source because of its excellent lateral thermal conductivity.

During the increase of heat flux from 90.8 to 351.5 W/cm², the hot spot temperature uniformity of #1 heat sink only increased slightly from 0.6 °C to 1.4 °C, and the uncertainty of temperature uniformity is ±2%. This is because the microchannel cold plate material used in this paper is high thermal conductivity diamond, which not only can effectively reduce the hot spot temperature, but also can improve the temperature uniformity of the multi-point heat source because of its excellent lateral thermal conductivity.

In Figure ??b, the hot spot temperature level and trend of #2 heat sink are basically the same as that of #1 heat sink, but the temperature uniformity performance is different. The hot spot temperature uniformity of #2 heat sink increases significantly from 0.4 to 3.5 °C during the increase of heat flux from 90.8 to 351.5 W/cm², which is inferior to that of #1 heat sink. This is probably due to the fact that the inlet fluid of #2 heat sink, which is located downstream, is in a gas–liquid two-phase state. The gas-liquid two-phase flow produces significant uneven distribution of flow and vapor quality in each channel after divided by the flow path [? ], which makes the two-phase flow more complex. There is too
much liquid phase in some channels and the boiling is not sufficient. But in some channels, there is too much gas phase, which may cause early drying out, deteriorate the temperature distribution uniformity and have an impact on the heat dissipation performance of the heat sink.

Figure ?? shows the trend of thermal resistance of the two heat sinks with heat flux. It can be seen that the thermal resistances of the upstream and downstream heat sinks have the same trend; both decrease with the increase of the heat flux of the heat source. This may be because with the increase of heat flux, the frequency of bubble generation and detachment at the microchannel wall increases. The latent heat transfer during bubble generation and growth, and the micro convection heat transfer between hot and cold liquid during bubble detachment enhance the heat transfer strength. As a result, the heat transfer capacity rises rapidly, manifested as a decrease in average thermal resistance $R_a$. The experimental results of Zhuan [??], Alam [??] and Balasubramanian [??] also showed similar heat transfer performance. In addition, due to the existence of flow resistance within the microchannel heat sink, the inlet pressure of #1 heat sink is higher than the outlet pressure of #2 heat sink. In this group of experiments, the inlet temperature of #1 heat sink and the outlet saturation temperature of #2 heat sink are controlled to be 15 °C. Therefore, the inlet temperature of #1 heat sink is lower than the inlet saturation temperature, and the inlet fluid is supercooled. The wall needs a certain degree of superheat to generate bubbles, which has a negative effect on the heat transfer capacity of the heat sink. The #2 heat sink is located downstream and the inlet fluid is in two-phase state, thus the heat transfer capacity of #2 heat sink is stronger than that of #1 heat sink and the thermal resistance is smaller.

Figure 6. Trend of thermal resistance under different heat flux.

### 4.3. Effect of Mass Flux

???? show the trends of the surface temperature, temperature uniformity and thermal resistance of the multi-point heat source on each heat sink during the process of changing the mass flux from 108 to 611 kg/(m²s) under the working conditions of heat flux of 308.7 W/cm², inlet temperature of #1 heat sink at 21 °C and outlet saturation temperature of #2 heat sink at 23 °C. It can be seen that, with the increase of mass flux, the hot spot temperature uniformity of #1 heat sink always remains at about 1 °C, and the thermal resistance is also basically unchanged, but the hot spot temperature rises slightly. This is because the outlet pressure of experimental #2 heat sink remains unchanged, but with the increase of mass flux, the flow resistance within the heat sink increases, so the import and export pressures of #1 heat sink increase, and the saturation temperature corresponding to pressure increases, and the fluid temperature within #1 heat sink increases. Therefore, the hot spot temperature rises slightly when the thermal resistance is basically unchanged, and the hot spot temperature increment and the qualitative temperature increment of the fluid within the heat sink is basically the same.
The heat source surface temperature, temperature uniformity and thermal resistance trends of #2 heat sink are consistent with that of #1 heat sink. By comparing the relative magnitude of the two heat sink indicators, it can be seen that the temperature uniformity of the downstream heat sink is inferior to that of the upstream heat sink, which may be because the inlet working fluid in downstream heat sink is in the gas–liquid two-phase state, and the flow and vapor quality distributions are more uneven, worsening the temperature distribution uniformity. The upstream heat sink temperature and thermal resistance are inferior to those of the downstream heat sink, which may be because the inlet fluid of #1 heat sink is supercooled, and the wall needs a certain degree of superheat to generate bubbles. It has a negative impact on the overall heat transfer capacity of the heat sink.

Considering the effects of heat flux and mass flux on the flow boiling heat transfer capacity in series heat sinks, it can be seen that the two-phase heat transfer capacity at the solid–fluid interface is enhanced with increasing heat flux within the range of experimental conditions, and is basically independent of the mass flux, which is the typical two-phase heat transfer characteristics dominated by the nucleation boiling mechanism, and is in agreement with the findings in other literatures [? ?]. The temperature uniformity of the upstream heat sink remains around 1 °C under different heat fluxes and mass fluxes, while the temperature uniformity of the downstream heat sink is poor and becomes more severe with the increasing heat flux that is due to the two-phase inlet conditions.

5. Conclusions

For the heat dissipation problem of multi-point heat sources, such as a simulated solar cell array, this paper carried out an experimental study of flow boiling within series mi-
crochannel heat sinks with ammonia as the working fluid, and the following experimental conclusions were drawn in the operating condition ranges of mass flow 108–611 kg/(m²s) and heat flux 90.8–351.5 W/cm²:

- The microchannel heat sink prepared with diamond is an effective means to solve the heat dissipation problem of multi-point heat sources. The surface temperature of the heat source is maintained below 65.9 °C under the heat flux of 351.5 W/cm², and the maximum temperature difference between multi-point heat sources of the upstream heat sink is only 1.4 °C.
- Heat transfer at the solid–fluid interface of the heat sink is dominated by the nucleation boiling mechanism. The heat transfer capacity increases with the increase of heat flux, which is less affected by the mass flux. The surface temperature of the heat sources and thermal resistance of the series heat sink varies in the same trend.
- The inlet states of upstream and downstream heat sinks are different, resulting in the difference in heat transfer capacities of the two heat sinks. The working fluid at the inlet of upstream heat sink has a small subcooling degree, leading to a high thermal resistance. While the two-phase inlet state of the downstream heat sink brings a worse temperature uniformity.

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Nomenclature

| Symbol | Description |
|--------|-------------|
| α      | heat transfer efficiency |
| A_h    | direct heating area m² |
| C_p,l  | specific heat capacity of liquid J/g K |
| δ      | distance m |
| d_h    | hydraulic diameter mm |
| G      | mass flux kg/(m²s) |
| λ      | thermal conductivity W/(m·K) |
| m      | mass flux g/s |
| P      | Pressure kPa |
| Q      | heat load W |
| q      | heat flux W/cm² |
| R_s    | thermal resistance K/W |
| T      | temperature °C |
| T_a    | average temperature °C |
| T_h    | upper surface temperature °C |
| ΔT_max | surface temperature uniformity °C |
| V      | Volume flux m³/s |

Subscripts

| Subscript | Description |
|-----------|-------------|
| eff       | effective value |
| f         | fluid |
| in        | inlet |
| out       | outlet |
| t         | total value |
Abbreviations

| Abbreviation | Description |
|--------------|-------------|
| CPV          | Concentrating photovoltaic |
| CV           | Chemical vapor deposition |
| DARPA        | Defence Advanced Research Projects Agency |
| GWP          | Global warming potential |
| HCPV         | High concentrated photovoltaic |
| ODP          | Ozone depletion potential |
| OM           | Optical microscope |

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