Abstract—Microfluid cooling solution is one of the most effective techniques for thermal management of high heat fluxes. A jet-based Si micro cooler with multiple drainage microtrenches (MDMTs) has been developed for microelectronic thermal management. Integrated with MDMT in hybrid micro cooler, the negative cross-flow effect between nearby nozzles is eliminated, and thus fully developed jet impingement can be enabled for each nozzle. An inlet/outlet flow arrangement layer has been introduced to achieve uniform pressure distribution. The effects of three types of arrangement structures on the hydraulic and thermal performance of micro cooler have been analyzed and compared. Two different thermal/ fluid simulation models have been constructed for micro cooler design. The test vehicle with the new nozzle/trench layer is fabricated using double-side deep reactive-ion etching process. Assembly of the stacked micro cooler and Si thermal test chip is finished through two-steps optimized thermal compression bonding process. With 0.05-W pumping power for the micro cooler, the heat dissipation of 260 W/cm² has been demonstrated, and the chip temperature can be maintained under 51 °C. Excellent agreement has been obtained between experimental and simulation results. With the MDMT, enhanced-microjet array impinging has been achieved, and uniform chip temperature distribution is obtained.

Index Terms—Drainage microtrench, high heat flux cooling, microchannel flow, microjet impingement, Si micro cooler.

I. INTRODUCTION

With the increasing trend toward advancements and miniaturization in electronic device, the thermal management is becoming even more significant [1]. Microfluid cooling solution is one of the most effective techniques for thermal management of high heat fluxes [2]–[4]. Jet impingement is of the capability to achieve large heat transfer rates, especially in the stagnation zones. The arrays of micro jet can be utilized to maintain acceptable surface temperatures and thermal gradients [5]. Many studies investigating the performance of microjet impinging array are available in the literature [6]–[9]. Wang et al. [10] studied three different micro jet arrays with varying jet diameters, spacings, and configurations using water. Jet diameters ranged from 40 to 76 μm with a 200-μm standoff. Heat transfer coefficients (HTCs) of 0.72–4.4 × 10^4 W/m²K were measured. The maximum power dissipated in 1-cm² heater was 90 W. Fabbri and Dhir [11] studied single-phase heat transfer of impinging jet arrays with three different circular array patterns. Laser drilling was used to create ten arrays with diameters ranging from 69 to 250 μm. The HTCs reported were 0.6–6 × 10^4 W/m²K. Brunschwiler et al. [12] created a series of microjet array with branched hierarchical parallel fluid delivery and return architectures. The peak HTC measured was 8.7 × 10^4 W/m²K. Sung and Mudawar [13] combined the microchannel and jet impingement, and their copper heat sink facilitated the dissipation of 304.9 W/cm². Michna et al. [14] investigated the heat transfer performance of five submerged and confined microjet arrays. Two array arrangements of jet with diameter of 54 and 112 μm were considered, and the Reynolds numbers were in the range 50–3500 for water. The hybrid micro cooler, combining both microjet array and multiple microchannels, has been applied for cooling high power device [15]–[17]. However, the negative cross flow of the spent liquid and interference between adjacent jets results in weakened impingement of several nozzles, and then diminishes the cooling capability of microjet array [18]–[20].

In this paper, the Si-based hybrid micro cooler with multiple drainage microtrenches (MDMTs) has been developed, and high heat dissipation capability has been demonstrated in the experimental tests. Uniform cooling for high heat fluxes has been achieved by eliminating the cross-flow effect between the nearby nozzles. The microfluid in the cooler is operated in laminar configuration. Two different thermal/fluid simulation models are constructed for micro cooler design. Performance comparison is made between the developed hybrid structure and the normal jet-based hybrid structure. An inlet/outlet flow arrangement layer has been introduced to achieve uniform pressure distribution for each nozzle. The effect of three types of arrangement structures on the hydraulic and thermal performances of the micro cooler is analyzed and compared. Experimental tests have been conducted on the test vehicle with newly designed micro cooler, which is fabricated and assembled through the optimized process. Excellent agreement has been obtained between experimental and simulation.
results. Using the developed hybrid microcooler with MDMT, high heat removal capability can be achieved with low pumping power requirement.

II. HYBRID MICROCOOLER WITH MDMT

The developed hybrid microcooler combines the merits of microchannel flow, microjet array impingement, and microtrenches drainage, as is shown in Fig. 1. In normal hybrid microcooler, the impinging coolant will flow out in two directions along microchannel. The influences of surrounding nozzles, including the cumulative accumulation of cross flows from sequential rows of jets, cause the effectiveness of each jet to diminish by increasing amounts with streamwise development. The accumulated cross flow will cause each jet impingement concentration to deflect and bend in the downstream direction. In the designed hybrid microcooler with MDMT, no cumulative accumulation of cross flow exists to cause detrimental influence on the adjacent jet impingement for all individual nozzles. The jet flow from microjet plate impinges on the top wall, and is constrained to flow along the microchannel, then exit through the nearby drainage microtrenches. Therefore, fully developed impingement can be achieved for all individual nozzles, which will enable more uniform and higher cooling capability over the whole cooling area.

For the chip of size $7 \times 7 \text{ mm}^2$, the microjet array is designed to cover an area of $8 \times 8 \text{ mm}^2$ for cooling. Three-type hybrid structures are considered, as is shown in Fig. 2. Structure A is the normal hybrid microcooler. The nozzle and microchannel dimensions of this normal microcooler have been optimized considering the combined effect of heat convection and conduction [9]. The nozzle is of $100 \mu\text{m}$ diameter and $400 \mu\text{m}$ length, and the microchannel is of $250 \times 250 \mu\text{m}^2$ size. In one microchannel, there are 21 nozzles of pitch $350 \mu\text{m}$. Structure B is one type of new hybrid microcooler. Drainage microtrench is designed to replace the nozzle at some positions, as is shown in Fig. 2(2). The trench width is initially set to be the same as the nozzle diameter $100 \mu\text{m}$, and the nozzle-trench distance ($D_{nt}$) is $350 \mu\text{m}$. In Structure B, there is one nozzle between two trenches, and total nozzle and trench number in one channel are 11 and 10. There are 21 microchannels included in both microcoolers.

The solid-fluid coupling simulation model is constructed using COMSOL Multiphysics, which runs the finite-element analysis together with adaptive meshing and error control. The built-in fluid flow and heat transfer interfaces are used. Due to the symmetries in the cooling system, the structure along one microchannel with symmetrical boundary conditions is constructed, as is shown in Fig. 3. The dashed rectangle in the top-view image highlights the built model in the whole microcooler structure. The built model includes two microchannel fins of half thickness, and all the nozzles and microtrenches along the channel. The outside surfaces of the channel fins are set to be symmetric boundaries. The solution was tested for mesh independency by refining the mesh size. Velocities and temperature matched within 0.1% for both mesh sizes. The convergence criterion of the solution is $10^{-6}$. The element size of the fluid part is calibrated for fluid dynamics, while that of the solid part is calibrated for general physics. Heating power is applied in the chip directly attached with the microcooler.

No slip boundary conditions are applied for stationary walls. With flow rate 200–500 mL/min, the microfluid in the microcooler is operated in laminar configuration. Pressure difference is applied between jet inlet and trench outlet surfaces. Temperature-dependent thermal conductivity
of Si is considered, the expression of which is 

\[ k_{Si} = 152 \times \left( \frac{298}{T} \right)^{1.334} \]

Thermal conductivity of Au/Sn bonding material between Si chip and microcooler is 57 W/mK. Water is used as coolant. Performances of the developed microcoolers are investigated and compared to the normal microcooler. Simulation results for Structures A and B are shown in Figs. 4 and 5.

As is shown in Fig. 4, higher heat removal capability has been achieved by using Structure B microcooler with MDMT. The HTC is calculated using 

\[ h = \frac{Q_w}{T_w - T_{in}} \]

where \( Q_w \) is the heat flux, \( T_w \) is the wall temperature, and \( T_{in} \) is the water temperature at jet exit. With 0.2 W pumping power, the HTC of Structure B is as high as \( 17 \times 10^4 \) W/m²K, which is 88.9% larger than Structure A.

Larger coefficients are obtained due to fully developed impingement of all nozzles in the new microcooler, as is shown in Fig. 5. Local heat convection is quite strong in the stagnation zone, and cooling performance drops rapidly away from the impingement zone. In Structure A, the impingement jets located farther downstream are deflected by the streamwise flow and become less coherent, which results in a gradual HTC decrease. Near the channel exit, it is difficult for the jet flow to penetrate the cross flows and then impinge on the target wall.

To obtain 300 and 400 mL/min flow rate, the pressure drops for normal cooler Structure A are 14 and 24 kPa, while for the new cooler Structure B they are as high as 24 and 42 kPa. The pressure drop in Structure B of the initial trench design is much higher than in Structure A. Several parametric analyses have been performed to reduce the pressure drop of Structure B. By increasing the microtrench width from 100 to 150 μm, the pressure drop can be reduced by around 15%. Further increasing from 150 to 200 μm will have negligible influence. However, the widened microtrench in Structure B still cannot enable lower pressure than Structure A for the same flow rate. Another way is to reduce the nozzle-trench distance. In this case, more nozzles will be included in Structure B. By reducing the nozzle-to-trench distance \( (D_{nt}) \) from 350 to 250 μm, there will be 15 nozzles and 14 trenches along one microchannel. The dense microjet array can enable better thermal and hydraulic performances. To obtain 300 and 400 mL/min flow rate, the pressure drops are 12 and 20 kPa in Structure B of \( D_{nt} 250 \) μm and trench width 150 μm, which is smaller than in Structure A.

Structure C is a new type of hybrid microcooler with two nozzles between two trenches. Based on the above parametric analyses, in Structure C, the nozzle-trench distance \( D_{nt} \) is set to be 250 μm, trench width is 150 μm, and there are 16 nozzles and 9 trenches included in one channel. Simulations have been conducted to investigate the performances of the designed new microcoolers. The results for Structure B of 250 μm \( D_{nt} \) and Structure C are shown in Figs. 6 and 7.
Higher heat dissipation capability can be achieved by using Structure C than Structure B. With 0.2 W pumping power, the HTC of Structure C is as high as $20 \times 10^4$ W/m$^2$K, which is 12.1% larger than Structure B of 250 μm $D_{nt}$. To obtain 300 and 400 mL/min flow rate, the pressure drops for both microcoolers are similar. In Structure C, the two nozzles between two trenches can both achieve fully developed impingement without mutual flow interference.

The temperature distributions on the chip top surface are compared among the above-mentioned four microcoolers, as is shown in Fig. 8. The temperature variation rate $\alpha_T$ is calculated using $\alpha_T = \Delta T_{\text{max}} / T_{\text{avg}}$ to evaluate temperature uniformity of the chip, in which $\Delta T_{\text{max}}$ is the maximum temperature difference and $T_{\text{avg}}$ is the averaged chip temperature.

With the normal cooler Structure A, the lower chip temperature occurs at the center of the chip. With the new cooler Structures B and C, the lower chip temperature occurs at each nozzle-covering area, as is shown in Fig. 8. The rate $\alpha_T$ with Structure C is lower than 4%, while in Structure B is around 5%, and in the normal cooler is as high as 10%. The lowest chip temperature is achieved by using Structure C microcooler. Of high cooling capability and low pumping power requirement, Structure C is used in new hybrid microcooler.

### III. INLET/OUTLET FLOW ARRANGEMENT

An inlet/outlet flow arrangement layer has been introduced beneath the microjet/microtrench layer to achieve similar pressure distribution for each nozzle, as is shown in Fig. 1. This layer should be carefully designed for the package-level microfluid cooling solution. To eliminate leakage risk inside the microcooler, which may diminish cooling performance, the microjet/microtrench layer and inlet/outlet arrangement layer are combined into one piece of an Si plate as a two-sided layer, as is shown in Fig. 9.

As is shown in Fig. 10 top view and bottom view, the top side of this plate is the microjet/microtrench layer, including the nozzle array and drainage trenches, and the bottom side of this plate is inlet/outlet flow arrangement layer, including inlet/outlet trenches and flow channels. In this structure, the coolant flows from one side of the cooler into the nozzles for impinging, and then drops through drainage microtrenches and goes out from the other side of the cooler. This type of inlet/outlet arrangement with one side in and one side out is hereinafter referred to as Arrangement 1 (A1). Two other types of inlet/outlet arrangement designs for this layer are proposed: Arrangement 2 (A2) and Arrangement 3 (A3), as is shown in Fig. 11.

As for the two-sided plate for these three types of design, the top-side microjet/microtrench layer is the same, as is shown in Fig. 10 top view. The bottom view is different. In Arrangement 2, the coolant flows from the bottom into the nozzles, and then drops through drainage trenches and exits from the two sides of the cooler. In Arrangement 3, the coolant flows into nozzles from two sides of the cooler, and then goes outside through drainage trenches to the bottom.

A new simulation model, including one slice of the structure with symmetrical boundary conditions, is built to study and
compare the hydraulic and thermal performances of different arrangement, as is shown in Fig. 12. The dashed rectangle in the cooler top-view image highlights the built model in the whole microcooler structure. Pressure difference is applied between inlet and outlet port surfaces. This model consists of more than 1.8 million tetrahedral elements to be mesh-independent. Inlet/outlet port locations are different. The pressure distributions are shown in Fig. 13.

The hydraulic and thermal performances of microcoolers with different types of flow arrangement designs are obviously different, as is shown in Figs. 14 and 15. For arrangement A1, number 1 nozzle is the nozzle located near the outlet port, while number 21 is the one near the inlet port. The pressure across the nozzle drops gradually from number 1 to 21. Nozzle number 1 has the highest pressure drop of 19.6 kPa, and achieves the best heat convection at its impinging zone with local HTC of \(11.7 \times 10^4\) W/m²K. Similar to A1, the nozzle numbers 1 and 21 in arrangement A2, which are close to the outlet ports, have the highest pressure drop of around 20.5 kPa, and achieve largest local HTC of \(12.6 \times 10^4\) W/m²K. The overall pressure drop between the inlet and outlet ports is 30 kPa. With arrangements A1 and A2, even the largest nozzle pressure drop is much smaller than the overall pressure. Arrangement A3 enables the best performances as we pursed. The pressure drop across all nozzles is around 28 kPa, which is close to the overall pressure, and the highest local HTC is more than \(22 \times 10^4\) W/m²K. Similar performance was achieved in each jet impinging zone. In the cooler with A3, the spatially averaged HTC for the whole cooling area is \(20 \times 10^4\) W/m²K, which is 230.8% and 68.5% larger than that with A1 and A2, respectively. Therefore, A3 flow arrangement structure is used in the new hybrid microcooler.

### IV. Fabrication and Experiment

The hybrid microcooler of new design Structure C and arrangement A3 is fabricated by bonding two Si plates together, one plate with multiple microchannels and the other is the two-sided layer with microjet/microtrench design and inlet/outlet arrangement. The two-sided layer is fabricated using double-side deep reactive-ion etching process, as is shown in Fig. 16.

Si thermal chip, microchannel plate, and two-sided plate are metalized with 5-μm Au/Sn layer for chip-level thermal compression bonding (TCB) process. Low-temperature temporary tagging of the stacked microcooler is conducted at first, and then high-temperature bonding is implemented to complete
Fig. 16. Photograph of the fabricated two-sided layer. (a) Top-side view of nozzle/trench. (b) Bottom-side view of inlet/outlet.

Fig. 17. X-ray image of the bonded thermal test chip and hybrid microcooler.

Fig. 18. Photograph of the assembled test vehicle.

the bonding of thermal chip and microcooler, as is shown in Fig. 17. The bonding temperature and force are carefully chosen to maintain tight contact and avoid components crack. The soldering quality at the bonding interface was checked using both X-ray and scanning acoustic microscopy. No void was detected in the bonding layers. A commercial thermal test chip of size $7 \times 7$ mm$^2$ and thickness 100 $\mu$m is used. The resister array in test chip is $7 \times 7$, and each resister unit cell covers an area of $1 \times 1$ mm$^2$. The thermal diodes are calibrated in the air-convection oven with data taken from room temperature of 25 °C–150 °C to get the K-factor. Great correlation was obtained between voltage change with temperature, and the well-matching K-factor is 551 °C/V.

To prepare the test vehicle, the bottom side of the microcooler is attached on the printed circuit board using epoxy, and the chip on the top is wire-bonded for electrical connection, as is shown in Fig. 18.

Fig. 19. Schematic of the experimental setup.

Fig. 20. Pressure drop as a function of flow rate from experiments and simulations for different microcoolers.

The experimental apparatus is shown in Fig. 19. The dc power is supplied to the resistor array of the thermal test chip. Water coolant is driven through the flow loop using a microgear pump. This pump forces water through a 15-$\mu$m filter and a flow meter before entering the microcooler. Inlet water and ambient temperature are around 25 °C. The heated water from the microcooler is cooled down by a water bath to ambient temperature, and then flows back to the cooling system. A differential pressure transmitter is attached to the manifold to measure the pressure drop. The thermal diode at steady state is monitored and recorded at different locations of the chip. The microcoolers with arrangements A1 and A2 are also fabricated and tested for performance comparison.

V. RESULTS AND DISCUSSION

A stable flow motion inside the heat sink can be obtained. In order to maintain 300-mL/min flow rate, the pressure drops are around 43, 19, and 10 kPa for arrangements A1, A2, and A3 structure, respectively. To obtain the same flow rate, the pressure drops in A2 and A3 are much lower than in A1. Pressure drop comparison between A2 and A3 is made, as is shown in Fig. 20. The hydraulic tests have been performed before and after 1000 cycles of thermal cycling reliability test (−40 °C–125 °C). Consistent results have been obtained, and no leakage was observed.

The cooling capability is shown in Fig. 21, and the pumping power for all cases is set to be 0.05 W. The simulation results show excellent agreement with the experimental results. The temperature shown in Fig. 21 is the tested maximum chip temperature ($T_{\text{max}}$). For A2, to dissipate 100 W, $T_{\text{max}}$ (65 °C) occurs near the chip center, while the minimum temperature $T_{\text{min}}$ (48 °C) is near the chip edge. A1 was also tested, and the temperature variation on the chip is quite large due to
the uneven pressure distribution inside the cooler. With only 30 W heating power, $T_{\text{max}}$ of the chip occurring near the inlet port already reached as high as 85 °C, while $T_{\text{min}}$ near the outlet port is only around 30 °C. The best performances are demonstrated by using A3. To dissipate the same heating power, even chip $T_{\text{min}}$ with A1 and A2 is still higher than $T_{\text{max}}$ with A3. In addition, lowest pressure is required for A3 to achieve such high cooling capability. The hybrid microcooler with A3 can dissipate 260 W/cm² while maintaining chip temperature under 51 °C. Several thermal diodes at different locations in the thermal test chip are measured. The maximum temperature difference between each monitor point is less than 2 °C. More uniform cooling capability has been achieved by using the designed Si hybrid microcooler with arrangement A3.

VI. CONCLUSION

A new jet-based microcooler with MDMT has been developed, and high heat dissipation capability has been demonstrated. The negative cross-flow effect between the adjacent jets is eliminated by integrating the microtrenches. Fully developed impingement can be achieved for all individual nozzles in the new microcooler, resulting in larger spatially average HTC than the normal hybrid cooler with the same pumping power. Microjet/microtrench design Structure C that can achieve high heat dissipation capability with low pumping power requirement is used in the new microcooler. Three types of inlet/outlet arrangements have been studied. A3 with side in and bottom out can enable similar pressure drop for each individual nozzle and requires lowest pressure drop compared to A1 and A2. The new hybrid microcooler with A3 is fabricated and bonded with the optimized two-steps TCB process. A1 and A2 are also fabricated and tested for performance comparison. With 0.05 W pumping power for the microcooler with arrangement A3, 260 W/cm² has been dissipated, and the chip temperature is kept under 51 °C. The on-chip temperature variation <2 °C has been obtained, suggesting uniform cooling capability is achieved. The developed hybrid Si microcooler with MDMT shows promise in improving the performances of microelectronic devices.

REFERENCES

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