Numerical study of the non-uniform channel width arrangement effect on mini-channel heat sink performance

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Abstract. The mini/micro-channel heat sinks have already been a successful innovation for cooling of electronic devices, and many factors could affect their thermal performance. However, the flow distribution is one of the fatal factors influencing the performance of the mini/micro-channel heat sinks. As we know, uniform flow strongly depends on the optimum structure design of the heat sinks. The particular focus of this work is the channel width arrangement effects on the flow uniformity and heat transfer inside mini-channel heat sinks (MCHSs). The mini-channel heat sinks with five non-uniform channel width arrangements were compared with a mini-channel heat sink with uniform channel width under four inlet flow rates and a fixed heat flux. Overall thermal resistance and pressure drop were used to evaluate the thermal performance of the heat sink. The simulation results show that the non-uniform channel width arrangement is significantly beneficial for achieving uniform flow distribution in the mini-channels and reduces the overall thermal resistance with up to 6.1%. It is found that the 5-type MCHS performs the best flow uniformity and the lowest thermal resistance with a moderate pressure drop. Besides, this study supplies a guideline to optimize the arrangement of channel width.

1. Introduction

Owing to significant heat transfer ability, small size, and small coolant requirement, micro-channel heat sink proposed by Tuckerman and Pease [1] and mini-channel heat sink, have been widely applied in electronic cooling. Over the past decades, a lot of investigations [2-8] have been conducted to enhance their thermal and hydraulic performance with experimental and numerical approaches, mainly focusing on structure optimization. Generally, a typical mini/micro-channel heat sink consists of many parallel channels. Therefore, the flow distribution among parallel channels also has a critical influence for the heat sink performance. A good manifold and channel structure design will optimize flow distribution significantly, which has been received amount of research attentions.

Some investigations [9-10] were conducted to evaluate the effect of inlet/outlet arrangement (IOA) on flow distribution uniformity in mini/micro-channel heat sinks. Results showed that the flow and temperature distribution inside these heat sinks were different among various IOAs.

Bifurcation of manifolds or channels was designed in Ref [11-12] for mini/micro-channel heat sinks. Results showed that such design was an efficient way to achieve uniform flow distribution, and the use of such design increased the total pressure drop, but the impact on flow distribution normally overcame the pressure loss impact.

Some researchers [13-15] found that it was positive for flow and temperature distribution uniformity to fill some resistance elements such as mini baffles, metal foam and pin fin arrays at the inlet of mini/micro-channel heat sink. However, the modifications introduced a higher pressure drop.
Some investigations were conducted to evaluate the effect of manifold design on flow distribution uniformity in mini/micro-channel heat sinks. Sohzabi et al. [16] found that flow distribution was relatively better for concave manifold and relatively poor for increasing manifold angle. Guo-Dong et al. [17] highlighted that flow distribution was relatively better for rectangular manifold and relatively poor for triangular one. Pistoresi et al. [18] conducted simulations on mini-channel heat sinks with an optimized discrete stairway shape and a continuous tapered shape manifold. They found that larger inclined angle in both shapes was favorable for more uniform flow distribution under higher flow-rate conditions. Saeed et al. [19] utilized their mathematical model to finalize a mini-channel heat sink with special header geometries. They found that flow maldistribution could be reduced up to 74% through proper design of manifold geometry.

Kumar et al. [20] numerically explored the influence of different inlet angles (such as $\theta = 90^\circ$, $\theta = 105^\circ$ and $\theta = 120^\circ$) on flow distribution and heat transfer performance in mini-channel heat sinks. They found that flow distribution changed with varying inlet angle. The results suggested that the performance of mini channel heat sinks could be improved if the coolant entered the distributor header with the flow inlet angle of $\theta = 105^\circ$ and exited from the middle of the collector header. Wei et al. [21] numerically and experimentally investigated the realization of target flow distribution among parallel mini-channels, using an optimized baffle insertion method. They found that different target distributions could be successfully achieved by the optimized baffle insertion method.

In a summary, most investigations about improving flow and temperature uniformities in mini/micro-channel heat sink which employs parallel channels with identical channel cross section. However, there is few studies with various channel cross section, such as various channel width. Therefore, it is necessary to assess the effect of non-uniform channel width on flow distribution in parallel channels. In this present study, a numerical analysis has been performed to obtain the performance of a water-cooled mini-channel heat sinks with 5 different mini-channel widths, which are compared with a heat sink with same channel width. The geometric dimensions of all the heat sinks and channels are same except the channel width for each simulation condition. Note that the mini-channel number is fixed so that the ratio of manifold cross-section area to total channel cross-sectional area are fixed too. Simulations under 4 inlet flow rates and further discussions will be presented and discussed in following section.

2. Mini-channel heat sink geometric configurations

The basic heat sink model to be studied contains 14 rectangular mini-channels defined as 0-type, and the geometric configuration of it is shown in figure 1 in detail. The thickness and height of the wall separating the channels are set to 1mm and 1mm respectively.

![Figure 1. Schematic diagram of the 0-type MCHS.](image)

To improve the flow distribution in the heat sink, the variable-channel-width arrangement is applied, where the channels become wider and wider from the middle to the edge of the heat sink.

Five types of non-uniform channel width arrangements are modeled based on 0-type with other geometries fixed. The channel widths of all the arrangements are illustrated in figure 2.
Due to symmetrical flow field, we take #1-#7 channels as discussed objects emphatically. For 1-type and 2-type arrangements, the channel width follows the arithmetic sequence, where the differences are 0.1mm and 0.2mm, respectively. For 3-type, the channel width varies from the side of the MCHS to the middle of the MCHS. For 4-type and 5-type, channel widths for #3 to #7 are the same, just channel widths for #7 and #8 are different.

3. Numerical procedures
The heat sink is designed to be attached to a chip which releases abundant heat during working. Across the base and channel walls of the heat sink, the heat is removed by the fluid. The features of the single-phase simulation include:

1. Fluid flow and heat transfer are in steady state and three dimensional.
2. The gravity effect is neglected, which is reasonable for mini-channel heat sink.
3. The flow is laminar.
4. All the outer surfaces of heat sink exposed to the surrounding are assumed to be adiabatic except the base plate area where constant heat flux is imposed.

The governing equations for fluid and energy transport are written as follows.

Continuity equation of fluid:
\[ \nabla \cdot \vec{V} = 0 \]  

Momentum conservation equation of fluid:
\[ \rho (\vec{V} \cdot \nabla \vec{V}) = -\nabla p + \mu \nabla^2 \vec{V} \]  

Energy conservation equation of fluid:
\[ \rho c_p (\vec{V} \cdot \nabla T) = k \nabla^2 T \]  

Energy conservation equation of solid
\[ k_s \nabla^2 T_s = 0 \]  

The boundary conditions are presented as follows.

Inlet:
\[ P = P_{in}, \quad T = T_{in} \]  

Outlet:
\[ P = P_{out}, \quad \frac{\partial T}{\partial n} = 0 \]
Fluid - solid interface:

\[ V = 0, \quad T = T_s, \quad k_s \frac{\partial T_s}{\partial n} = k \frac{\partial T}{\partial n} \]  

(7)

The base plate:

\[ q_b = -k_s \frac{\partial T_s}{\partial n} \]  

(8)

In the following simulation cases, the heat sink material and working fluid are chosen as copper and water respectively. The properties of water and solid are: \( \rho = 1000 \text{ kg/m}^3 \), \( c_p = 4182 \text{ J/(kg·K)} \), \( k = 0.6 \text{ W/(m·K)} \), \( \mu = 0.001 \text{ kg/(m·s)} \), \( k_s = 380 \text{ W/(m·K)} \). The inlet temperature of water is fixed at 25°C. Four inlet flow rates are simulated, including 200, 400, 600 and 800 g/min. The uniform heat applied at the base plate of the heat sink is fixed at 160 W.

The Reynolds number is defined as:

\[ Re_{Dh} = \frac{VD_h \rho}{\mu} \]  

(9)

With

\[ D_h = \frac{2H_c W_c}{H_c + W_c} \]  

(10)

Where \( V \) is the averaged velocity of fluid in the channel; \( D_h \) is the hydraulic diameter of the channel; \( H_c \) and \( W_c \) are height and width of the channel.

The maximum Reynolds number at the inlet flow rate of 800 g/min is below 1500, which indicates all simulated flow is laminar.

Equations (1-8) were solved numerically by a commercial finite volume CFD solver, COMSOL 5.3a.

**Table 1.** Grid independence examination, 0-type, inlet flow rate=800 ml/min.

| Mesh   | Grid size | Maximum temperature (°C) | Difference |
|--------|-----------|--------------------------|------------|
| Mesh 1 | 1.2×10^5  | 42.81                    | 0.9%       |
| Mesh 2 | 4.2×10^5  | 42.41                    | 0.04%      |
| Mesh 3 | 6.8×10^5  | 42.39                    | baseline   |

In order to ensure the accuracy and reliability for the simulation, the basic heat sink model, 0-type is used for verification. Three meshes of different size are used to check the grid dependence of the solution. Table 1 shows the deviation of calculated results with Mesh 3 as the baseline. Considering the accuracy and computation load comprehensively, Mesh 2 is adopted for the simulation in this paper.

In this study, the heat sink performances are characterized by the following two parameters. A standard deviation of mass flow rates in all channels is used for characterizing the flow uniformity, which is defined as:

\[ S = \left[ \frac{\sum_{i=1}^{n} (m_i - m_{\text{avg}})}{n} \right]^{1/2}, \quad i = 1, 2, \ldots, 14, \quad n = 14 \]  

(11)

Where \( m_i \) and \( m_{\text{avg}} \) are local mass flow rate of the No. \( i \) channel and average mass flow rate of all the mini-channels, respectively. A small \( S \) represents good flow uniformity.

The overall thermal resistance, which is defined as:

\[ R = \frac{T_{w,\text{max}} - T_{\text{in}}}{Q} \]  

(12)

Where \( T_{w,\text{max}} \) and \( T_{\text{in}} \) are the maximum temperature on the heat sink base plate and fluid inlet temperature, respectively; \( Q \) is the total heat transfer rate, equal to 160 W, in the present simulation.

Whether the heat transfer in the channel is the thermally developing region can be verified through the calculation of the thermal entrance length \( x_d \) using the following approximation [22].
\[ x_d = 0.005D_h R_e D_h P_r \] (13)

Where \( R_e D_h \) is the Reynolds number based on \( D_h \) of channel \#7, \( P_r \) is the Prandtl number.

Based on the given condition, \( x_d \) is 61 mm which is larger than the channel length of 30 mm employed in this study. So, the heat transfer in channel \#7 is in the thermally developing region.

The local Nusselt numbers along the channel are calculated according the numerically obtained data of temperature and heat transfer rate, as

\[ N_u z = \frac{h_z D_h}{k} = \frac{q_w D_h}{k(T_{w,z} - T_{f,z})} \] (14)

Where \( h_z \) is the local heat transfer coefficient, \( q_w \) is the local heat flux, \( T_{w,z} \) is the local averaged channel wall temperature, \( T_{f,z} \) is the local averaged fluid temperature.

Meanwhile, an analytical simulation has been also conducted according Shah and London’s correlation. Figure 3 demonstrates the comparison of local Nusselt numbers between numerical and analytical results. As seen, the numerical simulation agrees quite well with the analytical solution.

Figure 3. Local Nusselt number along channel \#7 of 0-type heat sink at inlet flow rate of 800 g/min.

4. Results and discussion

4.1 Flow distribution

There is a significantly uneven velocity profile in manifold causing larger fluid velocities in the middle channels near the entrance. The specific mass flow rate in each channel is illustrated in figure 4 (a), directly displaying the non-uniformity of flow. As the inlet flow rate increases, the non-uniformity becomes more remarkable. Figure 4 (b) shows the S value which increases with the inlet flow rate.

Figure 4. (a) Mass flow rate in each channel of 0-type MCHS with different inlet mass flows; (b) S with inlet mass flow rate for 0-type MCHS.

When non-uniform channel width arrangement is applied to the MCHS, a significant change takes place in the flow distribution which is shown in figure 5 (a), where the inlet flow rate is 800 g/min.
In other types of MCHSs, the mass flow rates in the channels near the entrance are smaller and the mass flow rates far away from the entrance are bigger compared with 0-type MCHS. The entire flow distribution presents like a “M-curve”. The minimum mass flow rates occur in #7 and #8 channels for 2-type MCHS which has the smallest channel widths of #7 and #8 channels among 1-type, 2-type and 3-type MCHSs. The mass flow rates in the channels near the outer sides of MCHS are relatively bigger for 3-type MCHS which has larger channel widths there. 3-type MCHS has a better flow distribution compared with 1-type and 2-type MCHSs whose channel widths follow arithmetic sequences. In 3-type MCHS, the width difference between adjacent channels increases from middle to outer regions.

Figure 5. (a) Mass flow rate in each channel of 0-, 1-, 2- and 3-type MCHSs at inlet flow rate of 800g/min; (b) S with inlet mass flow rate for 0-, 1-, 2- and 3-type MCHSs.

Figure 5 (b) shows that non-uniform channel width arrangement is conducive to flow uniformity apparently. There is still room for optimization.

Figure 6 (a) shows that the flow distribution after the channel width arrangement is optimized. There is a clear improvement in flow uniformity in 4-type and 5-type MCHSs, which have the same channel width from #3 to #12 channel, while the latter has a larger size in #1 and #14 channels. Figure 6 (b) shows that 5-type MCHS exhibits the best flow uniformity.

Figure 6. (a) Mass flow rate in each channel of 0-, 4- and 5-type MCHSs at inlet flow rate of 800g/min; (b) S with inlet mass flow rate for 0-, 4- and 5-type MCHSs.

4.2 Thermal performance and pressure drop

A uniform flow distribution in the MCHS is expected to enhance its thermal performance. It can be found from simulation results that 5-type MCHS has more low temperature area than 0-type. Besides, the highest temperatures of 5-type and 0-type MCHSs are 41.23 °C and 42.43 °C, and the averaged base plate temperatures of 5-type and 0-type MCHSs are 35.35 °C and 35.88 °C, respectively.

In order to further demonstrate the heat transfer enhancement effect, the heat transfer coefficient enhancement ratio, defined as \( \frac{h_{5\text{-type}}}{h_{0\text{-type}}} \) for each channel, is introduced. The heat transfer coefficient for each channel is characterized as:
\[ h_i = \frac{q_i}{T_{W,i} - T_{f,i}}, \quad i = 1, 2, \ldots, 14 \]  \hspace{0.5cm} (15)

Where \( q_i, T_{b,i}, T_{w,i} \) are the averaged heat flux, the averaged channel wall temperature and the averaged water temperature for \( i \) channel, successively.

**Figure 7.** The enhancement ratio of heat transfer coefficient for each channel at different inlet flow rates.

The calculated results are shown in figure 7. The plots indicate that the heat transfer coefficients of most channels are augmented by using non-uniform channel width arrangement which increases the velocities in those channels. It is noted that as inlet flow rate increases the heat transfer coefficients of the middle channels change little, while the heat transfer coefficients of the outer channels increases obviously.

**Figure 8.** Thermal resistance of five MCHSs with inlet flow rate.  
**Figure 9.** Pressure drop of five MCHS with inlet flow rate.

The analysis above has demonstrated the improvement of heat transfer in local regions as a result of the improvement of flow uniformity. The following content will present the heat transfer enhancement of the entire heat sink. Using Eq. (12), the lumped thermal resistance of the five MCHSs are calculated, as shown in figure 8. It can be seen that in the laminar region, \( R \) of each MCHS decreases with the inlet flow rate increasing. The thermal resistance of 5-type MCHS is the lowest attributed to the best flow uniformity as discussed above, 6.1% lower than 0-type MCHS in average. It is concluded that the non-uniform channel width arrangement brings about the improvement of flow uniformity and therefore the heat transfer enhancement.

Figure 9 shows that the pressure drops in five types of MCHSs over the inlet flow rate. It indicates
that the pressure drops in all MCHSs increase with inlet flow rate increasing. The pressure drops for 2-type and 3-type MCHSs are larger than other types of MCHSs due to their smaller hydraulic diameters of the channels at the region covered by mainstream. It is noted that 5-type MCHS has the best thermal performance but the moderate pressure drop.

5. Conclusions
In this study, non-uniform channel width arrangements are applied to the mini-channel heat sinks to optimize the flow distribution. Numerical simulation on the fluid flow and heat transfer characteristics in five types of MCHSs is conducted under the same operating conditions. 5-type MCHS performs the best flow uniformity and the lowest thermal resistance with a moderate pressure drop. This study offers a fast and efficient approach to optimize the heat sink structure, by which the flow uniformity can be improved and therefore the heat transfer can be enhanced. At present, it is necessary to consider the two factors, channel width and inlet/outlet manifold design, for uniform flow distribution. Thus, it is expected that a mathematical model considering the two factors can be proposed to achieve the best flow distribution uniformity in mini/micro-channel heat sinks.

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