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Study on the flow and heat transfer in a thermal shielding radiator

Feiyang Sun, Yiqiang Wu, Fubo Xie, Peng Qian, Minghou Liu*

Department of Thermal Science and Energy Engineering, University of Science and Technology of China, Hefei, Anhui, China.

Email:mhliu@ustc.edu.cn

Abstract: A thermal shielding radiator, which has an extra cuboid cavity between channels and sealplate, is proposed to achieve a large temperature difference between the hot and cold surfaces and high temperature uniformity on the cold surface. To get uniform distribution of the flow inside each channel, the thermal shielding radiator with different inlet/outlet types (namely C-, Z-, Y- and I-type) are numerically simulated. It is found that I- type has the best performance in velocity uniformity. Due to the cavity, the temperature difference between two sides of the thermal shielding radiator improves significantly. Besides, the mini-channel with equidifferent fin thickness is also proposed, and more uniform flow and the temperature distribution on the cold surface are achieved. Moreover, effect of geometry, operating parameters on the flow and heat transfer performance for both the thermal shielding radiator and the heat radiator are studied and compared.

1. Introduction

Mini-channel and micro-channel heat radiators are always used to transfer heat from the rapid advanced electronics. Micro-channel which has an astonishing heat flux always come up with a high pressure drop[1] while mini-channel has intermediate level of pressure drop and heat flux. Usually, the topic micro-channels is ranged from 10 to 200 μm while the mini-channels is ranged from 200 μm to 3 mm[2]. On the other hand, Kew[3], Ong[4] used dimensionless parameters to definite the distinction of the channel. The confinement number of 0.5 and the Eötvös number around 0.2 are used to identify micro- or mini-channel flow.

Micro-channel was firstly raised up by Tuckerman and Pease. They reported the reduction of the hydraulic diameter of channel from macro-scale to micro-scale cause the increase of heat flux[5]. Gunnasegaran[6] and Xia[7] studied the effect of geometric structures on fluid flow and heat transfer performance. Three different shapes (rectangular, trapezoidal and triangular) of microchannel heat radiators, four kinds of header shapes (triangular, trapezoidal and rectangular) were numerically studied and found rectangular shaped microchannel heat radiators showed the highest heat transfer

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coefficient and Poiseuille number, while rectangular head shape provided the better velocity uniformity. Chen[8] performed various inlet/outlet arrangements to investigate the flow and temperature distribution. Zhou[9] installed an experiment on the transient heat transfer of mini-channel and found that the outlet temperature became higher with the increase of the cooling water velocity. Al-Neama[10] presented a study on the influence of chevron fin structures in the mini-channel. They found that, with the decreasing of the chevron fin oblique angle, the thermal resistance reduced furtherly. Ho[11] designed a divergent rectangular mini-channel and found it had a negligible impact on the heat transfer at low flow rate.

From the literature mentioned above, micro or mini-channel has done a great contribution in heat transfer. However, it is rarely used as a cold plate to achieve the thermal shielding function. In this paper, a thermal shielding radiator with a cavity in the middle of the sink is proposed to enlarge the temperature difference between the cold and hot side.

2. Numerical modeling of mini-channel thermal shielding radiator

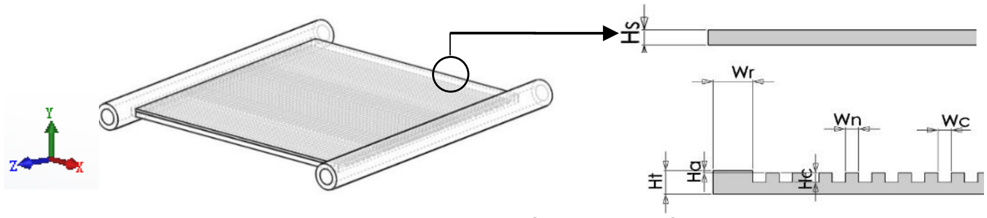
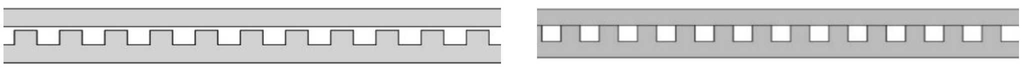


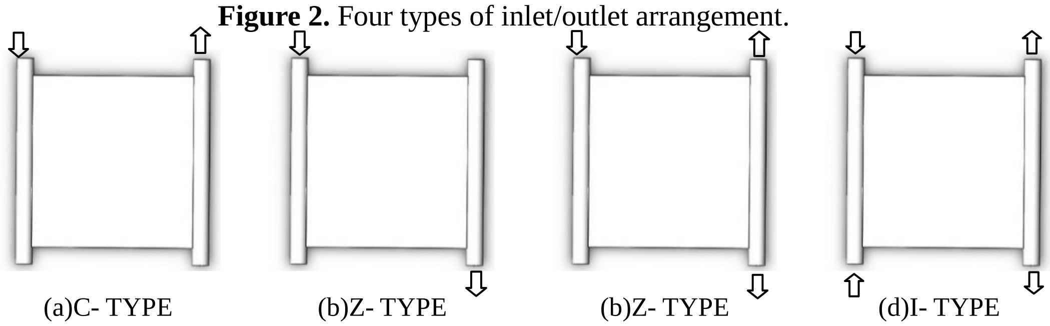
Figure 1. Geometric configuration of HIR and EHIR.

The heat insulation radiator(HIR), which has 48 channels with a cavity between the channel and the sealplate, is 100mm in length, 101mm in width and 3mm in thickness(figure 1). Cross section of traditional heat radiator and heat insulation radiator are displayed in figure 2. The equidifferent fins heat insulation radiator(EHIR) has the same size as HIR in the geometry. However, the fin width is growing as 1.05 times with the first fine width of 0.55mm. In present study, the heat insulation radiator is likely a mini-channel heat radiator which has enough height to arrange the inlet /outlet on the side wall. Four kinds of mini-channels(I、Z、Y、C) are proposed to study the influence of inlet/outlet position. The details are presented in figure 3. Geometrical dimensions are summarized in table 1.



(a)HIR. (b)Mini-channel.

Figure 2. Four types of inlet/outlet arrangement.



(a)C- TYPE (b)Z- TYPE (b)Z- TYPE (d)I- TYPE

Figure 3. Four types of inlet/outlet arrangement.

Table 1. Dimensions for HIR and EHIR thermal shielding radiator.

Characteristic	Size(mm)
Geometry width×length×height	100×101×3
Top plate height H_s	1
Bottom plate height H_t	2
Width between channel and side wall H_w	HIR:3; EHIR:1.7;
Fins width W_n	HIR:1; EHIR: W_n=$W_1 \cdot q^n$ [a]
Channel width W_c	1
Channel height H_c	1
Cavity height H_a	0.2

[a] W_1 =0.55, q^n1.05, n is positive integer

3. Governing equations

In present numerical study, the following assumptions are made: 1) Fluid flow and heat transfer are steady-state and three-dimensional. 2) The flow is laminar; 3) Properties of fluid and the radiator are temperature-independent. 4) Except the top plate where connected with the heat source, all the other surfaces exposed to the surroundings are heat insulated.

Based on above assumptions, the continuity, the momentum and the energy equation can be written as followings:

$$\frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} + \frac{\partial W}{\partial z} = 0 \tag{1}$$

$$\rho(\vec{V} \cdot \nabla\vec{V}) = -\nabla p + \mu\nabla^2\vec{V} \tag{2}$$

$$\rho c_p(\vec{V} \cdot \nabla T) = k\nabla^2 T \tag{3}$$

We assemble the entrance of the insulation at z=0, and sign the channel from 1 to 48. The radiator is made by aluminum, with a rubber layer between the top plate and the bottom plate for thermal resistance. The bottom/top plate is connected with the heat source which is taken as 400 K. In the mini-channel section, the inlet velocity is 2 m/s and the inlet water temperature is 300 K. The outlet is set as pressure out.

The flow fluid is deionized water and stays 300K by thermostatic water bath. The properties of solid material are as follows: $k_s = 148\, W/(m \cdot K)$, $c_{ps} = 712\, J/(kg \cdot K)$, $\rho_s = 2329\, kg/m^3$ respectively. The properties of water in $k_f = 0.612\, W/(m \cdot K)$, $\rho = 996.5\, kg/m^3$,$c_p = 4177\, J/(kg \cdot K)$, $\mu = 943.25 \times 10^6\, Pa \cdot s$.

The detail of parameter of the mini-channel is displayed in table 1. The hydraulic diameter D_h of the fluid flow channel is defined as $D_h = 4(W_c + H_c)/(2W_cH_c)$ and the Reynolds number Re is defined as $Re = \rho UD_h/\mu$, where U is the average velocity ranging from 0.68-1.37 m/s, μ is dynamic viscosity of fluid. The Re based on the hydraulic diameter and water velocity ranges from 718-1447.

To verify the independence of gird, three sets of grids (1.96× 10^6, 3.49× 10^6, 4.49× 10^6) are tested. The average and maximum temperature of top plate are316.14/314.28 K, 313.74/325.77 K, 320.63/321.98 K, respectively. It is clear that almost identical results are predicted for different grid systems. On account of the result from figure 4, the computation model with 1.96× $10^6$is used to obtain fluid flow characteristics accurately.

4. Results and discussion

4.1 flow uniformity

The thermal performance of the heat insulation is characterized based on the average top plate temperature while flow uniformity affects the temperature uniformity directly. Subsequently, Average flow flux of each channel is presented in figure 4 to show the flow uniformity. The cross section flow flux at the inlet of channel 1 to 48 revels that the inlet/outlet location of C-, Z-, Y-, I-type affects the fluid velocity. For I-type, the inlet/outlet location has higher average velocity as expected. Fundamentally, flow velocity is reassigned due to the opposite fluid flows that mix eventually. Figure 5 showed the pressure drop of four type mini-channels. It is found that the pressure drops of I-, Y-type are much smaller compared to C-, Z- type. When the fluid flows into the insulation by one side, recirculation zone can be obviously found near the center of the channel. It is because that the local pressure is smaller than surrounding pressure. Due to the uniformity of fluid flow, the I-type has been chosen for further study.

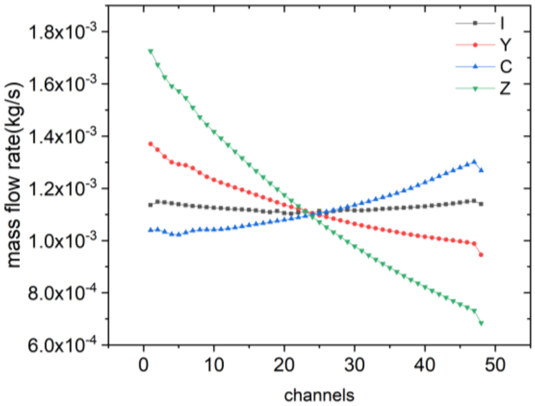


Figure 4. Comparison of the flow uniformity.

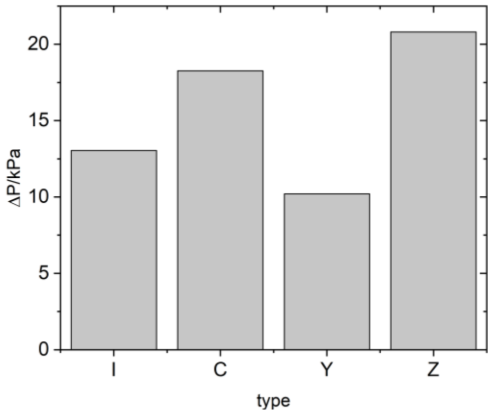
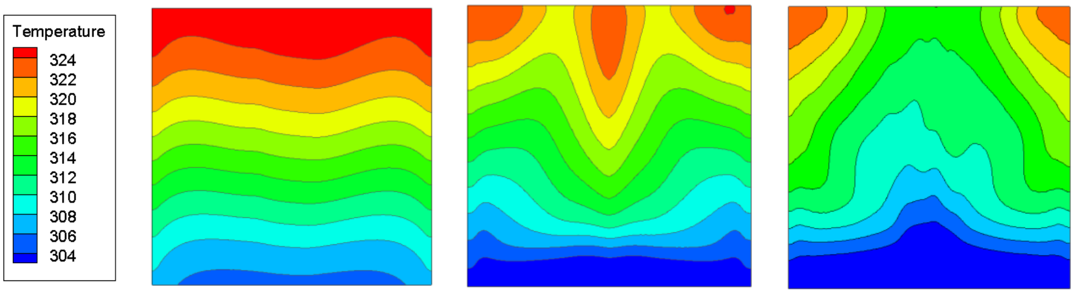


Figure 5. Comparison of pressure drop.

4.2 Temperature distribution

The top plate temperature contours of HIR, EHIR and traditional mini-channel heat radiator are shown in figure 6. The maximum, minimum, average temperature and the temperature difference between the top plate and the bottom plate are presented in table 2.



(a)mini-channel (b) HIR (c) EHIR

Figure 6. Temperature contours at the top plate.

According to figure 6, it is found the high temperature region occurs at the outlet for three type thermal shielding radiators due to temperature increasing of the fluid. The low temperature region occurs at the inlet due to the low inlet fluid temperature and the high local heat transfer coefficient. In figure 6(b), a high temperature located at the middle of the outlet region is found. This is due to the low mass flow rate at the middle channels. Figure 6(c) discloses that the equidifferent fins make the flow redistribution and eliminates the local high temperature. The table 2 showed that, compared with mini-channel, the HIR and EHIR get the lower average temperature because of the cavity used as a huge thermal resistance between the heat source and the surroundings. It can reduce the heat transfer to the surroundings and enlarge the temperature difference. In addition, the maximum temperature of HIR is higher than others due to the velocity reduces when the cross section of water area increase.

Table 2. Heat transfer performance of the threeheat radiators.

Type	T_{max}(K)	T_{min}(K)	T_{ave}(K)	ΔT(K)
Mini-channel	325.77	305.35	316.14	83.86
HIR	324.10	301.11	313.10	86.80
EHIR	323.48	301.05	312.32	87.68

4.3 Effect of heat source location

For HIR or EHIR, the heat source can be located either in seal plate side (top) or in channel side (bottom) which caused different location of the cavity. The heat dissipation, pressure drop and thermal shielding performance, characterized by the minimum, maximum or average temperature in the cold surface are shown table 3.

Table 3. The thermal shielding performance of HIR and EHIR.

Heat source	T_{max}(K)	T_{min}(K)	T_{ave}(K)	$Q(W/m^2)$	ΔP(Kpa)
HIR(top)	324.10	301.1	313.40	5.1×10^5	4.71
HIR(bottom)	335.70	301.24	322.47	7.8×10^5	4.71
EHIR(top)	323.48	301.05	312.32	4.3×10^5	4.69
EHIR(bottom)	329.26	302.29	317.62	6.3×10^5	4.69

For HIR or EHIR thermal shielding radiators, the heat dissipation is about 52.94% or 46.51% augment if heat source is located in channel side (bottom). However, the maximum temperature for HIT or EHIR increases about 11.60 K or 5.78 K when heat source is located in the seal plate side (top), respectively. For both thermal shielding radiators, the pressure drops are nearly the same. It is suggested that, for higher thermal shielding purpose, the heat source should be located in seal plate side, which can achieve larger temperature difference between the hot and the cold side. And it can get higher heat dissipation by changing the location of the heat source to the channel side.

5. Conclusions

To achieve a large temperature difference between the hot and cold surfaces and high temperature uniformity on the cold surface, a heat insulation radiator is proposed. To get uniform distribution of the flow inside each channel, the mini-channel thermal shielding radiator with different inlet/outlet types (namely C-,Z-,Y-, I types) are numerically simulated. The conclusions are summarized as:

(1) The temperature uniformity of the thermal shielding radiator cold face is affected by inlet/outlet

type due to changing of the flow distribution in the channel. Among C-, Z-, Y-, I-type of the heat radiator, the I-type shows uniform mass flow rate. For Y-, I-type, however, they can reduce the pressure drop.

(2) For higher thermal shielding purpose, the heat source should be located in seal plate side, which can achieve larger temperature difference between two sides for both HIR and EHIR. For heat dissipation purpose, however, the heat source should be located in the channel side.

(3) Due to the equidifferent fins, the flow and temperature are more uniform in EHIR.

Acknowledgments

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