Numerical Study of Enhanced Heat Transfer of Microchannel Heat Sink with Nanofluids

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Abstract: The electronic devices on the circuit board of the navigation device have the defects of high power and difficulty in heat dissipation, thereby restricting their performance. In view of the above defects, this paper proposes a miniature water-cooled radiator with a trapezoidal cross-section spoiler arranged at equal intervals. The numerical simulation of the three-dimensional laminar flow and heat transfer is carried out to study the performance of the micro water-cooled heat sink. The effects of Reynolds numbers and the types of nanofluids on the heat transfer performance of microchannel heat sink are also investigated. The results show that when the Reynolds number ranges from 100 to 500, the existence of the spoiler column makes the heat dissipation effect of the microchannel heat sink significantly improved, and the friction coefficient decreases with the increment of the Reynolds number. With the increasing of the volume fraction in the nano-particles, the heat transfer performance of microchannel heat sink is also improved. The heat transfer performance of SiC nanofluids is better than that of TiO$_2$ nanofluids under the same flow rate.

1. Introduction
With the rapid development of microelectronics technology, highly integrated miniaturized and high-power devices have become the main development trend, and the integration of devices is becoming higher and higher, resulting in the local heat flux density of devices increasing dramatically [1-3]. Studies have shown that chip-level heat flux can be as high as 100W/cm$^2$ and for component level can reach up to 25W/cm$^2$. The heat dissipation of high-performance modules and power devices of high-power navigation equipment, such as the satellites and radars, seriously affects the reliability of equipment and restrains the development of high technology. Therefore, there is a very high requirement for thermal design, and the problems of high heat generation and dissipation are urgently needed to be solved.

The microchannel heat sink has the advantages of compact structure, such as the weight and high heat exchange area, and also has a better heat dissipation performance than macroscopic heat dissipation method [4-7]. Despite of the significant advances in the microchannel heat sink system technology; more efficient coolants are needed to further increase cooling efficiency in some cases. By dispersing the nano-sized particles with a diameter about 1-100 nm into the base liquid, the heat transfer performance of the new heat conducting fluid can be improved obviously, and the heat conductivity is higher than that of the conventional liquid, thus it is also called as nanofluids [8-10]. As it can be uniformly distributed in the base solution and has a high thermal conductivity, it has been drawn a lot of attention by the researchers. However, the thermal conductivity properties of nanofluid can be determined by varied factors, such as solid particle (shape and size), the kinds of base solution, annexing agent, temperature and pondus hydrogenii values [11-14]. Especially for the SiC nanofluids and TiO$_2$. 

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which have a high thermal conductivity capacity [15-18]. However, there are no literatures reporting the enhance thermal by combing microchannel with nanofluids to cool the electronic devices.

Based on the above research, in this paper, the microchannel and nanofluid strengthening technology are combined to dissipate heat from a typical circuit boards in the navigation electronic devices. A new type of micro channel heat dissipating device for trapezoidal cross section spoiler is proposed and its heat transfer characteristics are numerically simulated. The average nussel number and average fanning friction coefficient of microchannel heat sink under different Reynolds numbers are studied. The effects of SiC nanofluid and TiO$_2$ nanofluid on the microchannel heat sink are compared in details.

2. Mathematical-physical model

2.1 Description of physical problems

The microchannel heat sink designed in this paper is based on a channel matrix made of copper, and water and nanofluids are cooling fluids. The specific physical model is shown in Fig. 1. As the spoiler column is arranged symmetrically in the channel, a unit in the microchannel heat sink is chosen for simulation study. Fig. 2 shows a detailed schematic diagram of calculation simulation region. The exact geometries of microchannel heat sink are shown in Fig. 3 and Table 1, respectively.

![Fig.1 Schematic diagram of microchannel heat sink](image1)

![Fig.2 Schematic diagram of simulation region](image2)

![Fig.3 Physical dimension](image3)

| Parameters | $W_s$ | $h_c$ | $t$ | $L_1$ | $W_2$ | $W_1$ | $\theta$ | $W_L$ |
|------------|-------|-------|-----|-------|-------|-------|---------|-------|
| Values     | 0.1   | 0.3   | 0.1 | 2.25  | 0.15  | 0.5   | 45      | 0.3   |

2.2 Mathematical modelling

Steady state and incompressible flow are adopted in this paper. Heat sink surface is smooth; the effects of radiation heat transfer and surface tension are ignored and the deposition of nanoparticles in the nanofluid solution is not considered. The thermophysical properties of the liquid have a great influence on the flow characteristics of the microchannel heat sink. All physical parameters are fixed values without internal heat source. Under the above assumptions, the continuity equation, momentum conservation equation and energy conservation equation of the fluid can be expressed as follows:
The continuity equation can be expressed as:
\[ \nabla \cdot (\mathbf{v}) = 0 \]  
(1)

The momentum conservation equation can be written as:
\[ \nabla \cdot (\rho \mathbf{v} \mathbf{v}) = \nabla P + \nabla \cdot (\mu \nabla \mathbf{v}) \]  
(2)

The energy conservation equation can be written as:
\[ \nabla \cdot (\rho \mathbf{v} c_p T) = \nabla \cdot (k \nabla T) \]  
(liquid)

\[ \nabla \cdot (k \nabla T) = 0 \]  
(solid)  
(3)

1) Boundary condition

Inlet:
\[ x = -2.25, \quad u = u_f, \quad v = w = 0, \]
For fluid: \( T_f = T_{in} = 300K \),
For solid: \( -k_s \left( \frac{\partial T_s}{\partial x} \right) = 0 \)
(4)

Outlet:
\[ x = 0 \quad P_f = P_{out} = 1 \text{atm}, \]
For fluid: \( -k_f \left( \frac{\partial T_f}{\partial x} \right) = 0 \),
For solid: \( -k_f \left( \frac{\partial T_f}{\partial x} \right) = 0 \)
(5)

Top:
\[ z = 0.4\text{mm} \quad u = v = w = 0 \]
\[ -k_s \left( \frac{\partial T_s}{\partial y} \right) = 0 \]  
(6)

Convection heat transfer contact surface can be expressed as:
fluid / solid surface: \( u = v = w = 0 \),
\[ -k_s \left( \frac{\partial T_s}{\partial n} \right) = -k_f \left( \frac{\partial T_f}{\partial n} \right) \]  
(7)

Periodic boundary:
\[ \text{At } y = 0\text{mm} \quad \text{symmetry} \]
\[ \text{At } y = 0.3\text{mm} \quad \text{symmetry} \]  
(8)

Bottom:
\[ z = 0\text{mm} \]
\[ -k_s \left( \frac{\partial T_s}{\partial y} \right) = q = 100W / \text{cm}^2 \]  
(9)

2) Parameter definition

The characteristic length of the channel is defined as:
\[ D_b = \frac{2W_c h_c}{W_c + h_c} \]  

(10)

The expression of Reynolds number is:

\[ \text{Re} = \frac{\rho u_m D_b}{\mu} \]  

(11)

where \( \rho \) is fluid density, \( u_m \) is average fluid velocity, \( \mu \) is hydrodynamic viscosity coefficient, \( W_c \) is channel width and \( h_c \) is channel height.

The density of nanofluids is defined as:

\[ \rho_{nf} = (1 - \varphi) \rho_b + \varphi \rho_p \]  

(12)

where \( \varphi \) is the volume fraction of nanoparticles in solution, \( \rho_b \) is the density of base solution, \( \rho_p \) is the density of nanoparticles.

The dynamic viscosity of nanofluids is defined as:

\[ \mu_{nf} = \frac{1}{(1 - \varphi)^{2.5}} \mu_f \]  

(13)

where \( \mu_f \) is the dynamic viscosity of the base liquid.

The specific heat capacity of the nanofluid is defined as:

\[ C_{pnf} = \frac{(1 - \varphi) \rho_b C_{pb} + \varphi \rho_p C_{pp}}{(1 - \varphi) \rho_b + \varphi \rho_p} \]  

(14)

where \( C_{pb} \) is the specific heat capacity of the base solution, and \( C_{pp} \) is the specific heat capacity of the nanoparticles.

The thermal conductivity of nanofluids is defined as:

\[ K_{nf} = K_b \left[ \frac{K_p + (n - 1) K_b - \varphi(n - 1)(K_b - K_p)}{K_p + (n - 1) K_b + \varphi(K_b - K_p)} \right] \]  

(15)

where \( n \) can be expressed as:

\[ n = \frac{3}{\psi} \]  

(16)

where \( K_b \) is the thermal conductivity of the base solution, \( K_p \) is the thermal conductivity of the nanoparticles, \( n \) is the shape factor of the nanoparticles, and \( \psi \) represents sphericity. For spherical nanoparticles, the usual value \( \psi \) is 1. The physical properties of other parameters are shown in Table 2 and Table 3.

### Table 2 Properties of SiC and TiO$_2$

| Thermophysical properties | H$_2$O | SiC | TiO$_2$ |
|---------------------------|-------|-----|--------|
| Density, \( \rho \) [kg m$^{-3}$] | 994.2 | 3160 | 4157 |
| Specific heat capacity, \( C_p \) [J kg$^{-1}$K$^{-1}$] | 4178 | 675  | 710   |
| Thermal conductivity, \( k \) [W m$^{-1}$K$^{-1}$] | 0.625 | 490  | 8.4   |
| Viscosity, \( \mu \) [N s m$^{-3}$] | 7.25×10$^{-4}$ | ...... | ...... |
### Table 3 physical parameters under the different percent of SiC and TiO₂ nanofluids

| Properties                  | SiC-H₂O  | TiO₂-H₂O |
|-----------------------------|----------|----------|
| \( \rho_{nf} \) [kg m\(^{-3}\)] | 1%       | 5%       | 9%       | 1%       | 5%       | 9%       |
| \( \mu_{nf} \times 10^{-3} \) [N s m\(^{-3}\)] | 7.43     | 8.23     | 9.17     | 7.43     | 8.23     | 9.17     |
| \( C_{p,nf} \times 10^{-4} \) [J kg\(^{-1}\) K\(^{-1}\)] | 4070     | 3670     | 3340     | 4040     | 3550     | 3161     |
| \( k_{nf} \) [W m\(^{-1}\) K\(^{-1}\)] | 0.644    | 0.723    | 0.809    | 0.67     | 0.703    | 0.771    |

The average fanning friction coefficient is defined as:

\[
\bar{f} = \frac{\Delta p D_h}{2 \rho L u_m^2}
\]

where \( L \) is the total length of the microchannel, \( \Delta p \) is the pressure drop along the length. \( \Delta p = \bar{p}_m - p_{out} \), where \( \bar{p}_m \) is weighted average pressure of import quality.

The average heat transfer coefficient is defined as:

\[
\bar{h} = \frac{q A_w}{A_{con} (\bar{T}_w - \bar{T}_f)}
\]

The average nussel number is defined as:

\[
\bar{Nu} = \frac{\bar{h} D_h}{k_f}
\]

where \( q \) is the heating source at the bottom surface, \( A_w \) is the area of the heating source at the bottom, \( A_{con} \) is the coupling area between solid and liquid, \( \bar{T}_w \) is the weighted average temperature at the bottom surface, \( \bar{T}_f \) is the weighted average temperature of the fluid mass, and \( k_f \) is the thermal conductivity of the fluid.

### 3. Independence check of grid

The continuity equation adopts the Standard format, and the SIMPLE algorithm is used to deal with the coupling problem of pressure and velocity. The momentum equation adopts the upwind format of second-order accuracy during the simulation process.

In order to ensure the quality of flow field, the domain decomposition method is often used for practical devices with complex geometric shapes. Due to the complex flow field in the trapezoidal spoiler region, the microchannel model is divided into two regions, the inlet and outlet region and the spoiler region. The two regions are divided into grids, which can ensure the accuracy of calculation and save time. In this paper, ICEM software is used to grid the 3D model. For the near wall surface and the spoiler column area, due to the large velocity and pressure gradient, a dense grid is adopted. The meshing results are shown in Figs. 4 (a) and (b).
4. Results and discussion

4.1 The effect of Reynolds number

With the increase of Reynolds number, the convective heat transfer capacity of the micro-channel increases, thus the average nussel number increases and the average surface temperature decreases as shown in Fig. 6 and Fig. 7 (a). This is because the secondary flow is generated in the micro-channel radiator by introducing an inclined channel, which leads to the destruction of the velocity boundary layer, thus reducing the thickness of the average thermal boundary layer and achieving the purpose of strengthening heat dissipation. But with the increase of Reynolds number, the increase of average nussel number slows down, and the decrease of surface temperature also slows down. With the increase of Reynolds number, fanning friction coefficient of the channel shows a downward trend as shown in Figs. 6 and 7(b). Therefore, with the increase of Reynolds number, heat transfer is further enhanced and system work loss is reduced.
Fig. 6 Diagram of the change of the average nussel number and fanning friction coefficient under different Reynolds numbers

Fig. 7 Surface average temperature and interface velocity and downstreamer distribution with different Reynolds numbers

On the above basis, \( z = 0.2 \text{mm} \) is taken as an example to study the streamlines distribution of inclined channels under different Reynolds Numbers, as shown in Fig. 7(c). At the entrance of the secondary circulation channel close to the wall, there is an obvious vortex generation, and the size and influence area of the vortex increase with the increase of the Reynolds number. When the Reynolds number is 500, the vortex is the largest and most numerous. At the vortex, the large temperature gradient changes from the wall to the boundary between the vortex and the main flow, indicating that the vortex can enhance heat transfer.

4.2 The effects of nanofluids on heat transfer
On this basis, nanofluids are introduced to further enhance heat transfer through the microchannel heat sink. As shown in Fig. 8 and Fig. 9, both nanofluids have an enhanced effect on heat transfer of the microchannel heat sink, and it can be seen that with the increase in the concentration of nanofluids, the maximum temperature on the surface of the radiator gradually decreases, indicating that the heat transfer ability is gradually enhanced. And as the flow rate of the liquid increases, the temperature drop gradually slows down. Compared Fig. 8 with Fig. 9, when two kinds of nanofluids with the same concentration are selected for heat transfer comparison, the heat transfer capacity of the two kinds of nanofluids increases with the increase of refrigerant flow rate, which is mainly due to the enhanced convective heat transfer capacity when the flow rate increases. At the same flow rate, the maximum temperature drops
to 2.5 K. At the same time, it can be found that the heat transfer effect of SiC nanofluid is better than that of TiO$_2$ nanofluid, which is mainly due to the fact that the thermal conductivity of SiC nanofluid is better than that of TiO$_2$ nanofluid when the content of nanofluid is 9%.

5. Conclusions

In this paper, the numerical study of three-dimensional laminar flow and heat transfer of a miniature water-cooling radiator with trapezoidal section and evenly spaced spoiler columns designed for the circuit board of navigation equipment is carried out. The results show that:

(1) In the range of 100-500 Reynolds number, the larger the Reynolds number is, the smaller the friction resistance is, and the decreasing trend gradually becomes gentle. For the micro-channel radiator studied in this paper, the Nu value increases with the increase of Reynolds number.

(2) As for the microchannel radiator studied in this paper, secondary flows is generated due to the trapezoidal section structure of the spoiler column, which destroys the velocity boundary layer, reduces the thickness of the thermal boundary layer, and improves the heat transfer capacity of the microchannel.

(3) Both SiC nanofluids and TiO$_2$ nanofluids have enhanced heat transfer of the radiator, and with the increase of flow rate, the enhanced heat transfer effect gradually slows down, and the former has better heat transfer effect than the latter.

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