Microchannel cold plate heat transfer and flow resistance characteristics calculation and structure optimization

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Abstract—This paper designs a microchannel liquid cold plate suitable for array multi-chip cooling. The influence of aspect ratio of rectangular cross section on heat transfer and flow performance of microchannel cold plate was studied by numerical analysis. The optimal ratio of height to width of rectangular section is selected and its structure is optimized. The results show that when the equivalent diameter of rectangular cross-section is fixed, the friction coefficient \(f\) and Nusselt number \(\text{Nu}\) increase with the increase of aspect ratio of rectangular cross-section; after optimizing the structure of the channel, the microchannel cold plate exhibits good heat transfer characteristics of low pressure drop, realizing effective heat dissipation and meeting the requirements of pressure drop.

1. INTRODUCTION
The microchannel liquid cold plate is a heat transfer micro-scale effect and a boundary layer enhanced heat transfer mechanism. It has the advantages of strong heat dissipation capability, low heat transfer resistance and compact size, but it has difficulty in forming cold plate structure and has large flow resistance\([1]\). When designing a microchannel cold plate, it must consider its heat transfer performance, flow resistance performance and molding difficulty. Many scholars have found that different structural features have a great impact on the heat transfer and flow resistance characteristics of microchannel cold plates.

In 2006, Wang found that the fractal microchannel heat exchanger has the advantages of pressure reduction and temperature uniformity\([2]\). In 2010, Hasan simulated the flow and heat transfer in a stable countercurrent chip microchannel heat exchanger\([3]\). It was found that the smaller the channel size, the better the heat transfer effect and the larger the pressure drop. In 2017, Zhu used a rectangular plate with small rectangular fins to form a microchannel liquid-cooled runner structure, and its ultimate heat dissipation capacity increases with the decrease of the fin size\([4]\). In 2018, Li et al. found that when the cooling water inlet position is in the middle of the hot end of the chip, the flow distribution of each channel is the most uniform; the narrow and high channel structure has good heat dissipation performance and large flow resistance\([5]\).

In this paper, the numerical simulation method is used to investigate the heat transfer and flow resistance characteristics of the microchannel cold plate. The main values of coolant pressure drop, chip temperature and temperature difference are evaluated, and the friction coefficient \(f\) and Nusselt number \(\text{Nu}\) are analyzed. The heat transfer and flow resistance characteristics of the microchannel cold plates with different aspect ratios were studied by using ethylene glycol solution as the cooling medium. The
Microchannel cold plate runners are structurally optimized to meet pressure drop and heat dissipation requirements.

2. MICROCHANNEL COLD PLATE STRUCTURAL DESIGN

2.1. Microchannel cold plate performance indicators

Microchannel cold plates usually use the heat transfer coefficient and inlet and outlet pressure difference as the main indicators to measure the performance of the cold plate. The heat transfer coefficient \( h \) is an index that measures the heat dissipation capacity of the cold plate to the heat source. The inlet and outlet differential pressure \( \Delta P \) can measure the pump power. Number equations consecutively.

\[
h = \frac{Q}{A(T_w - T_0)} \quad (1)
\]

\[
\Delta P = P_{\text{in}} - P_{\text{out}} \quad (2)
\]

Where \( A \) is the contact area of the heat source and the cold plate, \( m^2 \); \( T_w \) is the surface temperature of the heat source, \( T_0 \) is the average temperature of the inlet and outlet coolant, \( K \); \( Q \) is the total heat exchange amount, \( J/s \); \( P_{\text{in}} \) is the average pressure of the water inlet, and \( P_{\text{out}} \) is the average pressure of the water outlet.

In order to more accurately reflect the performance of the cold plate, this paper uses the dimensionless parameter Nusselt number \( Nu \) and the flow friction coefficient \( f \).

\[
Nu = \frac{h d}{\lambda} \quad (3)
\]

\[
f = \frac{2d \Delta P}{\rho u^2} \quad (4)
\]

Where \( d \) is the equivalent diameter of the heat transfer surface, \( m \); \( h \) is convective heat transfer coefficient, \( W/(m^2\cdot K) \); \( \lambda \) is the thermal conductivity of the stationary fluid, \( W/(m^2\cdot K) \); \( \rho \) is the fluid density, \( kg/m^3 \); \( l \) is the length of the flow channel, \( m \); \( u \) is the average velocity of the fluid, \( m/s \).

2.2. Structural design

Cold plate design requirements: the total heat that needs to be taken away by the entire cold plate is 400W; Liquid supply the temperature is 40℃, the maximum temperature of the chip surface is \( \leq 60^\circ C \), the temperature difference is \( \leq 10^\circ C \); the flow rate of the cooling medium must not exceed 1.5L/min, and the pressure loss of the cooling medium is less than 0.06 MPa.

According to the above requirements, the cold plate is designed to have a two-layer structure, a substrate with a microchannel flow path, and a cover plate with a position for mounting the chip. The cold plate structure is shown in Fig. 1, the flow channel shown in Fig. 2. The chip layout area is provided with a plurality of microchannels in parallel with the flow channel form, which enhances the heat dissipation capability and takes into account pressure loss characteristics.

Figure 1. Cold plate structure.
2.3. Mathematical model
To simplify the analysis, apply the following assumptions: The fluid flow is stable, incompressible, and the heat exchange process is steady state; The entire microchannel is dominated by laminar flow, and the laminar model is selected; Ignoring radiation, gravity, viscous dissipation and thermal contact resistance between components.

Mass conservation equation:
\[ \frac{\partial}{\partial x_i} (\rho u_i) = 0 \] (5)

In the inertial coordinate system, the momentum conservation equation in the i direction:
\[ \frac{\partial}{\partial x_j} (\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} + F_i \] (6)

Energy equation for calculating the heat transfer problem between a fluid and a solid region:
\[ \frac{\partial}{\partial x_i} \left[ u_i (\rho E + p) \right] = \frac{\partial}{\partial x_i} \left( k_{\text{eff}} \frac{\partial T}{\partial x_i} \right) \] (7)

In the above formula, \( u_i \) is the velocity vector in the i direction, \( \rho \) is the density of the fluid, \( p \) is the static pressure, \( F_i \) is the other volume force, \( \tau_{ij} \) is the stress tensor, \( E \) is the energy, \( k_{\text{eff}} \) is the effective thermal conductivity, and \( T \) is the temperature of the fluid.

3. Simulation results and analysis

3.1. Meshing
Each model uses a tetrahedral unit generated by Mesh. The fine mesh is concentrated near the wall, which solves the large velocity gradient and thermal boundary layer, while the other parts of the mesh are relatively sparse[6].

In order to verify the independence of the grid, a grid system of 435364 units, 1753644 units, 5906245 units and 12503259 units was generated in the cold plate. The results show that the relative error of the Nusselt number of the third grid system is 0.5% and less than 1%. Therefore, a third grid system was used for further simulation.

| Number of grids | 435364 | 1753644 | 5906245 | 12503259 |
|-----------------|--------|---------|---------|---------|
| Nu              | 19.11  | 19.51   | 19.79   | 19.89   |
| Relative error  | 3.9%   | 1.9%    | 0.5%    | 0       |
3.2. Material setting and boundary conditions
Starting energy equation; coupling boundary condition; outer surface convection boundary, convective heat transfer coefficient 10W/m²·K, high temperature ambient temperature 50°C, coolant inlet temperature 40°C; chip heating using energy source; speed inlet, pressure outlet.

The coupling of pressure and velocity adopts the Couple algorithm to achieve the stability of the solution convergence. The inlet volume flow rate is 1.5 L/min, and the fluid inlet velocity is determined to be 0.8846 m/s. Adiabatic and no slip boundary conditions are used. The calculation model fluid is No. 65 coolant, and the cold plate is aluminum alloy 6063.

3.3. Analysis of simulation results
The chip temperature and temperature difference, flow channel pressure drop were analyzed. The simulation results are shown in Fig. 3 and Fig. 4.

According to the above requirements, the cold plate is designed to have a two-layer structure, a substrate with a microchannel flow path, and a cover plate with a position for mounting the chip. The cold plate structure is shown in Fig. 1, the flow channel shown in Fig. 2. The chip layout area is provided with a plurality of microchannels in parallel with the flow channel form, which enhances the heat dissipation capability and takes into account pressure loss characteristics.

Figure 3. Chip temperature results.

Figure 4. Flow path pressure drop results.

The highest temperature is 50.26°C, the lowest temperature is 44.25°C, the temperature difference is 6.01°C, the chip temperature and temperature difference are all satisfactory. The inlet is 85.78 kPa, the outlet is 10 kPa; the pressure drop is 75.78 kPa > 60 kPa, and the pressure drop does not meet the requirements.

In order to study the effect of channel aspect ratio on the heat transfer and flow properties of the microchannel cold plate, the equivalent diameter Dh of the fixed microchannel section, change the aspect ratio, and select the cold plate with the best heat transfer and flow resistance characteristics.
TABLE 2. PARAMETERS OF RECTANGULAR CROSS-SECTION MICROCHANNELS

| Model | Width W (mm) | Height H (mm) | Equivalent diameter Dh (mm) | Aspect ratio |
|-------|--------------|---------------|----------------------------|--------------|
| Case1 | 0.6          | 6.67          | 2.257                      | 11.11        |
| Case2 | 0.8          | 5             | 2.257                      | 6.25         |
| Case3 | 1            | 4             | 2.257                      | 4            |
| Case4 | 1.2          | 3.33          | 2.257                      | 2.78         |
| Case5 | 1.4          | 2.86          | 2.257                      | 2.04         |

The temperature and pressure drop of the cold plate structure of the five models were simulated. The summary of the simulation calculation results is shown in Table 2.

TABLE 3. SUMMARY OF COLD PLATE CALCULATION RESULTS

| Model | Chip Maximum Temperature(℃) | Temperature Difference(℃) | Pressure Drop(kPa) | f   | Nu  |
|-------|------------------------------|---------------------------|--------------------|-----|-----|
| Case1 | 49.04                        | 5.72                      | 140.50             | 0.376| 22.96|
| Case2 | 49.75                        | 5.7                       | 100.37             | 0.269| 21.06|
| Case3 | 50.26                        | 6.01                      | 75.78              | 0.203| 19.79|
| Case4 | 50.71                        | 6.06                      | 63.57              | 0.170| 18.96|
| Case5 | 51.05                        | 6.14                      | 60.25              | 0.161| 18.55|

Fig. 5 shows the variation of the friction coefficient and the Nusselt number of different aspect ratios in the laminar flow state of the microchannel. As the aspect ratio becomes larger, the friction coefficient and the Nusselt number of the liquid flow are also larger.

Considering the cold plate chip temperature, temperature difference and pressure drop, combined with the friction coefficient and the Nusselt number change rule, the Case 4 cold plate is selected for structural design, and the flow channel is optimized to reduce the pressure loss.
3.4. Cold plate flow path optimization calculation and analysis
In the right-angled position, the pressure drop caused by the flow path communication position and the flow path exit is large, and it is necessary to improve these positional structures. Increase the radius of the right-angled corner, change the inside of the right angle to a corner, and change the outlet of the flow path from a wide tube to a straight tube.

| Parameter | flow path height | flow path width | angle bend width | angle corner radius | Inner corner radius |
|-----------|------------------|-----------------|------------------|---------------------|-------------------|
| Size(mm)  | 5                | 1.2             | 8                | 5                   | 3                 |

The optimized cold plate structure is simulated again. The maximum temperature of the chip is 50.67℃, and the temperature difference is 6.07℃. The chip temperature and temperature difference are all satisfactory. The inlet pressure is 68.80kPa, the pressure drop is 58.80kPa < 60kPa, and the pressure drop meets the requirements.

After optimizing the design, the temperature and temperature difference are basically unchanged; the overall pressure drop of the flow channel is obviously improved to meet the requirements. Both the corner radius and the outlet line have a significant effect on the pressure drop performance.

4. Simulation results and analysis.
Numerical simulation of the microchannel cold plate heat transfer process using a three-dimensional coupled flow heat transfer model. The geometrical structure was analyzed by parameters such as temperature difference and flow path pressure drop. The influence of the aspect ratio of the microchannel rectangular section on the flow friction coefficient $f$ and Nusselt number $Nu$ was studied. The track structure is optimized to reduce its flow resistance. Simulation results show:

1. When the microchannel equivalent diameter is fixed, the frictional resistance coefficient and the Nusselt number increase as the aspect ratio of the rectangular section becomes larger. When the aspect ratio is 2.78, the cold plate has the best heat transfer and flow performance;
2. The initial design of the runner structure and the optimization scheme of the runner structure can achieve effective heat transfer and temperature reduction of the chip;
3. By adjusting the structure of the flow channel, the pressure drop characteristics of the flow channel can be changed, and the radius of the corner and the form of the outlet flow channel have a significant influence on the pressure drop performance.

Acknowledgment
Thanks to Qing Zhang and Chao Zhang for their help in the research.

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