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Technical Note

Effect of tip clearance on the cooling performance of a microchannel heat sink

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Abstract

In this paper, the effect of tip clearance on the cooling performance of the microchannel heat sink is presented under the fixed pumping power condition. The thermal resistance of a microchannel heat sink is defined for evaluating its cooling performance. The effect of tip clearance is numerically investigated by increasing tip clearance from zero under the fixed pumping power condition. From the numerical results, the optimized tip clearance is determined, for which the thermal resistance has a minimum value. Finally, we show that the presence of tip clearance can improve the cooling performance of a microchannel heat sink when tip clearance is smaller than a channel width.

Keywords: Microchannel heat sink; Tip clearance; Thermal resistance; Pumping power condition

1. Introduction

Does tip clearance have an adverse effect on the cooling performance of a heat sink? Tip clearance has two competing effects on heat transfer phenomena of a heat sink. The first effect is the promotion of the heat transfer rate because the heat transfer coefficient is remarkably increased at the tip surface [1]. The second effect is the restraint of the heat transfer rate due to the effect of flow bypass [1–4]. Because of the two competing effects, optimal tip clearance for the maximum heat transfer rate may exist when the pumping power is provided. So far, many investigators [2–4] have focused only on the effect of flow bypass. They experimentally and numerically have shown that tip clearance decreases the cooling performance of a heat sink. As a result, the optimization studies on minimizing the thermal resistance of microchannel heat sinks have been presented only when there is no tip clearance [5–7]. To the authors' knowledge, the optimization of a microchannel heat sink with tip clearance has not been presented under the fixed pumping power condition.

In this paper, the effect of tip clearance on the cooling performance of a microchannel heat sink is presented under the fixed pumping power condition. For this purpose, we define the thermal resistance of microchannel heat sinks for evaluating the cooling performance and numerically obtain its value by varying tip clearance. The optimized tip clearance is determined based on the results of the numerical analysis. Finally, we show that the presence of tip clearance can improve the cooling performance of a microchannel heat sink when tip clearance is smaller than a channel width.

2. Problem description and numerical model

The problem under consideration in this paper concerns forced convective flow through a microchannel heat sink with small tip clearance, as shown in Fig. 1(a). The flow direction is parallel to the x-axis. The top surface is insulated and the bottom surface is uniformly heated. A coolant passes through the microchannels and

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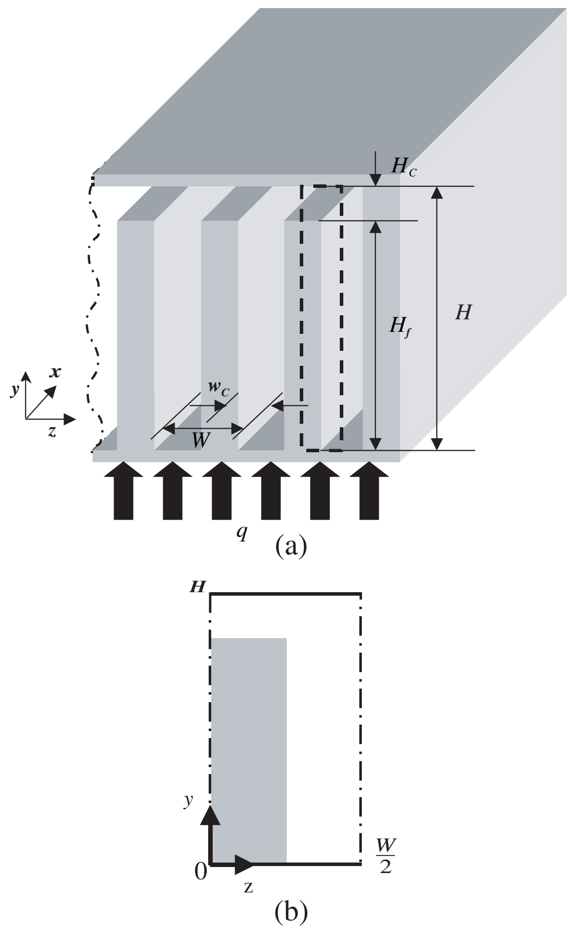

Fig. 1. Schematic of a microchannel heat sink with tip clearance. (a) microchannel heat sink, (b) computational domain.

takes heat away from a heat-dissipating component attached below. Water is used as a coolant, and the material of the microchannel heat sink is silicon. In analyzing the problem, the flow can be assumed to be laminar and both hydrodynamically and thermally fully-developed because the hydraulic diameter of the microchannel heat sink is sufficiently small. All thermophysical properties are assumed to be constant. In order to evaluate the effect of tip clearance on the cooling performance of the microchannel heat sink, the momentum for the fluid and the energy equations for both the fin and the fluid should be solved. The computational domain for this conjugate heat transfer problem is shown in Fig. 1(b). The governing momentum equation and boundary conditions are given as

$$-\frac{1}{\mu}\frac{\mathrm{d}P}{\mathrm{d}x}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}=0 \tag{1}$$

$$u=0 \quad \text{at } y=0,\ y=H$$

$$\frac{\partial u}{\partial z}=0 \quad \text{at } z=0,\ z=\frac{W}{2}$$

The no-slip condition is satisfied along the fin surface as well as on the top and bottom walls depicted using a solid line in Fig. 1(b). The symmetric condition is satisfied along the boundaries denoted using a dashed line. Similarly, the governing energy equation and boundary conditions are given as

$$\rho C_p u\frac{\partial T}{\partial x}=k\left(\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \tag{2}$$

$$T=T_{\mathrm{W}} \text{ at } y=0, \quad \frac{\partial T}{\partial y}=0 \text{ at } y=H$$

$$\frac{\partial T}{\partial z}=0 \quad \text{at } z=0,\ z=\frac{W}{2}$$

where u, P, μ, ρ, C_p, T, T_{W}, q'' and u_{m} are x-velocity, pressure, viscosity, density, specific heat, temperature, wall temperature and mean velocity, respectively. From the energy balance, $\partial T/\partial x = q''/\rho C_p u_{\mathrm{m}} H$, where q'' denotes the uniform wall heat flux. The governing equations are solved by the control-volume-based finite difference method. A non-uniform grid is used to employ a fine grid in a region near the interface between the solid phase and the fluid phase. The discontinuity of the thermal conductivity at the interface is treated by placing a control surface at that location and utilizing the harmonic mean formulation for the thermal conductivity. The numerical solutions for the velocity and temperature distributions are obtained for the computational domain shown in Fig. 1(b). Based on these numerical results, the thermal resistance can be obtained as tip clearance is increased from zero under the fixed pumping power condition. The thermal resistance is expressed by

$$\theta=\frac{T_{\max}-T_{\mathrm{in}}}{q} \tag{3}$$

where θ, $T_{\max}$, T_{in} and q are thermal resistance, maximum temperature at the base of the microchannel heat sink, inlet temperature and total amount of heat, respectively. In addition, the pumping power is defined as

$$\text{Pumping power}=\Delta P\cdot Q \tag{4}$$

where ΔP and Q are pressure drop across a system and volume flow rate, respectively.

The thermal resistance is numerically calculated under the fixed pumping power condition in the present study, while other previous studies have adopted the fixed pressure difference condition as a constraint. The condition of the fixed pressure difference physically indicates that the power required to supply the fluid through the channel between the fins and tip clearance changes whenever the geometry is varied. However, the condition of the fixed pumping power used in this paper physically means that the power required to drive the fluid through the heat sink is fixed. Therefore, the fixed pumping power condition as a constraint for evaluating cooling performance of heat sinks is physically more practical than the fixed pressure difference condition.

3. Results and discussion

Before investigating the effect of tip clearance on the cooling performance, we calculated velocity and temperature distributions of the microchannel heat sink without tip clearance. Velocity and temperature profiles in the microchannel heat sink without tip clearance are compared with the exact solution by Shah and London [8] and the analytical solution by Kim and Kim [7], respectively. They are in close agreement with each other within 0.5%. The thermal resistance is expressed as the sum of the thermal resistance of the fin and that of the flow [5].

$$\theta = \theta_{\text{fin}} + \theta_{\text{flow}} \tag{5}$$

The thermal resistance of the fin means the resistance associated with both conduction through and convection between the fins and can be represented by

$$\theta_{\text{fin}} = \frac{T_{\max} - T_{\text{b}}}{q} \tag{6}$$

where $T_{\max}$ and T_{b} are maximum temperature at the base of the heat sink and bulk mean temperature, respectively. The thermal resistance for the flow is responsible for the temperature rise of the coolant from the inlet to the exit. From the energy balance, this resistance can be expressed as

$$\theta_{\text{flow}} = \frac{T_{\text{b}} - T_{\text{in}}}{q} = \frac{1}{\dot{m}C_p} \tag{7}$$

where T_{in}, C_p and $\dot{m}$ are inlet temperature and specific heat of a coolant and mass flow rate, respectively. In addition, the thermal resistance can be expressed with number of transfer units (NTU) terminology, which is familiarly dealt with in analysis related to heat exchangers.

$$\theta = \theta_{\text{flow}} + \theta_{\text{fin}} = \frac{1}{\dot{m}C_p}\left(1 + \frac{1}{\text{NTU}}\right) \tag{8}$$

Under the condition of the fixed mass flow rate, the maximum value of NTU equals the minimum value of the thermal resistance. However, the condition of the fixed pumping power is more widely used for determining an optimum geometry of the microchannel heat sink. Hence the present study uses the fixed pumping power condition as a constraint. Under this condition, the mass flow rate is not fixed but changed according to a system impedance. Therefore, the thermal resistance is used in estimating the cooling performance of the microchannel heat sink. For this the thermal resistance of the microchannel heat sink is calculated by increasing tip clearance from zero with the fixed fin thickness and the channel width. The present work focuses on the effect of tip clearance when it is smaller than the channel width, while previous investigators [1–4] have focused on the effect of flow bypass with a tip clearance larger than the channel width. Fig. 2 shows the trend of both the thermal resistance of the flow and that of the fin with dimensionless tip clearance as increasing under the fixed pumping power condition. As tip clearance becomes larger, the thermal resistance of the flow is smaller. This is due to the fact that the pressure drop is decreased and the mass flow rate is in turn increased as tip clearance is increased under the fixed pumping power condition. Regarding the thermal resistance of the fin, it is shown to be nearly constant in the range of $0 < H_C/w_C < 0.6$ while it is substantially increased for $H_C/w_C > 0.6$. This phenomenon can be explained as follows: as tip clearance is increased, the heat transfer coefficient near the fin sidewall is decreased and that near the fin tip surface is increased. This is because the mass flow rate between the channels is decreased while that through tip clearance is increased when the pumping power is fixed. Based on the results, the decrement of the heat transfer rate near the fin sidewall is nearly offset by the increment of that near the fin tip surface for $0 < H_C/w_C < 0.6$. Therefore, the thermal resistance of the fin is nearly constant. On the other hand, for $H_C/w_C > 0.6$ the decrement of the heat transfer rate near the fin sidewall overshadows the increment of that near the tip surface. So, in the range of $H_C/w_C > 0.6$, the thermal resistance of the fin is remarkably increased. Based on the numerical results, we can obtain optimal tip clearance for which the thermal

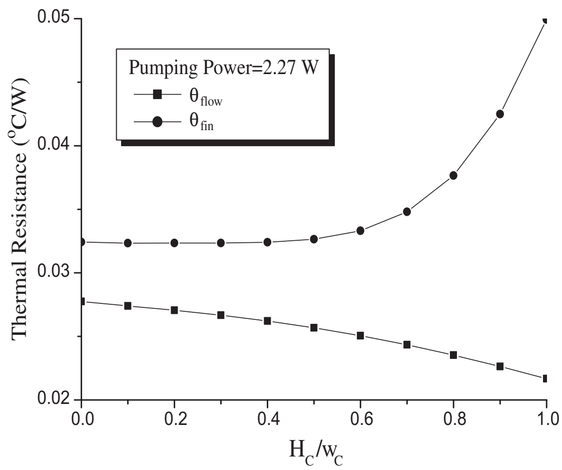

Fig. 2. Thermal resistance of the flow and that of the fin.

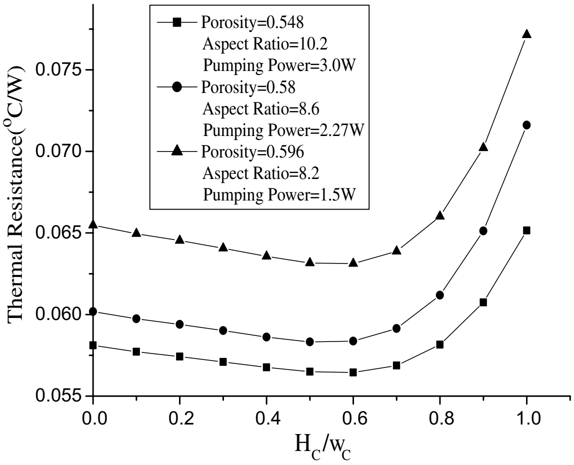

Fig. 3. Thermal resistance vs. H_C/w_C.

resistance is minimized, because the thermal resistance of the flow is decreased and that of the fin increased as tip clearance is increased. As shown in Fig. 3, there exists an optimal value for tip clearance of a microchannel heat sink with the various values of aspect ratio and porosity (the porosity is defined as w_C/W in this study) under the various pumping power conditions when H_C/w_C is about 0.6. Under the pumping power of 2.27 W, the minimum thermal resistance is 0.058 °C/W at $H_C/w_C = 0.6$, as shown in Fig. 3. The enhancement of the cooling performance of the microchannel heat sink with $H_C/w_C = 0.6$ is about 3.5% compared with that of the optimized microchannel heat sink without tip clearance. The result obtained in this study is of practical importance because the presence of optimal tip clearance can improve the cooling performance of the heat sink. In other words, the microchannel heat sink does not need to be fully shrouded to achieve maximum cooling performance. This is quite a surprising result when compared with what thermal engineers have believed so far.

4. Conclusion

In this paper, the effect of tip clearance on the cooling performance of the microchannel heat sink is presented under the fixed pumping power condition. For this purpose, we define the thermal resistance of the microchannel heat sinks for evaluating the cooling performance and numerically investigate the effect of tip clearance on the cooling performance by increasing tip clearance. From the numerical results, we have shown that the thermal resistance of the flow is decreased while that of the fin is increased as H_C/w_C is increased. As a result the overall thermal resistance, which is a combination of the two, attains a minimum value when H_C/w_C is about 0.6. The enhancement of the cooling performance of the microchannel heat sink with $H_C/w_C = 0.6$ under the pumping power of 2.27 W is about 3.5% compared with that of the optimized microchannel heat sink without tip clearance. The result obtained in this study is of practical importance because the result shows that the presence of optimal tip clearance can improve the cooling performance of the heat sink. In other words, the microchannel heat sink does not need to be fully shrouded to achieve maximum cooling performance. This is quite a surprising result when compared with what thermal engineers have believed so far.

Acknowledgements

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