PERGAMON

International Journal of Heat and Mass Transfer 45 (2002) 2823–2827

International Journal of HEAT and MASS TRANSFER

www.elsevier.com/locate/ijhmt

Technical Note

Numerical optimization of the thermal performance of a microchannel heat sink

J.H. Ryu, D.H. Choi *, S.J. Kim

Department of Mechanical Engineering, Korea Advanced Institute of Science and Technology, Taejon 305-701, South Korea

Received 13 June 2001; received in revised form 2 November 2001

1. Introduction

The recent trend in the electronic equipment industry toward denser and more powerful products requires higher thermal performance from a cooling technique. Many ideas for innovative cooling methods have been proposed including a microchannel heat sink. A microchannel heat sink, as first proposed by Tuckerman and Pease [1], is based on the idea that the heat transfer coefficient is inversely proportional to the hydraulic diameter of the channel. Even though the microchannel heat sink has brought significant improvements in the cooling performance, it has not been widely used because of two limitations: a high pressure drop due to a small hydraulic diameter of the channel and a significant temperature variation within the heat source between inlet and outlet.

The design of a microchannel heat sink and its performance in dissipating heat have been treated in many experimental and analytical studies [1–3]. These approaches, however, may be termed as only approximate as they rely on a simple one-dimensional model for the thermal conduction along the fin height. Others [4,5] modeled the microstructures of a device as a porous medium and used Darcy's law to describe the fluid flow.

Numerical works on the microchannel heat sink in the literature are mostly two dimensional and leave the three dimensionality largely untreated. The purpose of the present study is to develop a three-dimensional analysis procedure and examine the entrance effects on the thermal performance of a microchannel heat sink. The procedure is then incorporated in an optimization scheme to find the optimal design parameters of a microchannel heat sink that minimizes the thermal resistance subject to a specified pumping power.

2. Numerical analysis

The problem under consideration concerns the forced convection through a microchannel heat sink depicted in Fig. 1(a). A coolant passes through a number of microchannels and takes heat away from a heat dissipating electronic component attached below. In analyzing the problem, the flow is assumed to be laminar, incompressible, and hydrodynamically fully developed because the hydrodynamic entry length ($L_e \approx 0.05\, Re_{D_h} D_h$) is about 5% of the length of a heat sink. In addition, all thermophysical properties are assumed constant.

To predict the thermal performance of the microchannel heat sink more accurately, a three-dimensional numerical analysis is conducted. Taking advantage of geometric symmetry, the computational domain is simplified as shown in Fig. 1(a). The Navier–Stokes and energy equations that govern the flow may be expressed as

momentum equation:

$$\mu\left(\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right)=\frac{\mathrm{d}p}{\mathrm{d}x} \quad \text{(fluid region)}, \tag{1}$$

energy equation:

$$\rho_f c_p u\frac{\partial T}{\partial x}=k_f\nabla^2 T \quad \text{(fluid region)}, \tag{2}$$

$$0=k_s\nabla^2 T \quad \text{(solid region)}. \tag{3}$$

The velocity field for a given pressure gradient is known analytically [6]. As for the thermal boundary conditions, the central surfaces of the fin and the channel are

* Corresponding author. Tel.: +82-42-869-3018; fax : +82-42-869-3210.

E-mail address: dhchoi@mail.kaist.ac.kr (D.H. Choi).

0017-9310/02/$ - see front matter
PII: S0017-9310(02)00006-6

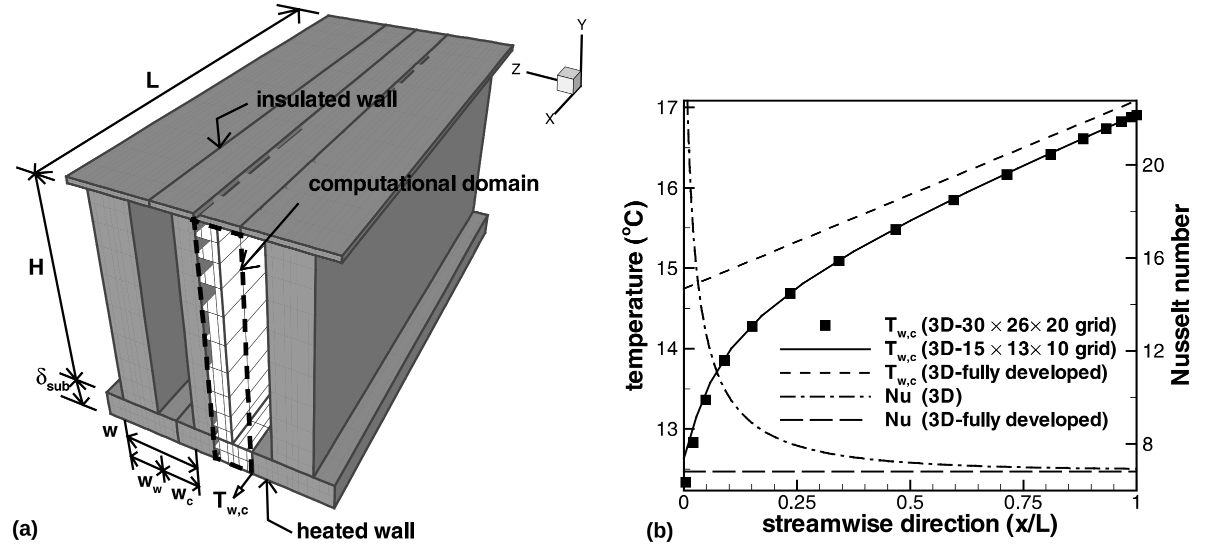

Fig. 1. Thermal characteristics of a microchannel heat sink for $P_p = 2.56$ W, $H = 453$ μm, $w_c = 45.3$ μm, $w_w = 35.3$ μm, $\delta_{sub} = 100$ μm: (a) schematic of a microchannel heat sink and computational domain; (b) wall temperature distribution along the channel centerline and section Nusselt-number distribution in the streamwise direction.

adiabatic due to symmetry. The top surface of the domain can also be considered adiabatic because it is usually covered by a thermally insulated material. A uniform heat flux is applied to the bottom surface of the substrate which is attached to the heat generating component. The coolant enters each of the microchannels with a prescribed temperature ($T_{f,in} = 10$ °C) and is assumed to reach the thermally fully developed state, that the rate of temperature increase is constant, at some distance downstream. The latter allows us to impose the condition of vanishing streamwise diffusion at the channel exit. Likewise, the streamwise diffusion in the solid region is set equal to zero at both ends.

The governing Eqs. (2) and (3) are then solved by an ADI-type finite-volume method. Here the derivatives are discretized by the central differencing except for the convective terms, which are done by the upwind scheme. The process is iterative and the solution is considered converged when the sum of the variations between the two successive iterations for the entire region becomes less than 10^{-3}. A nonuniformly distributed $15 \times 13 \times 10$ grid is found to be adequate, as the result obtained with a $30 \times 26 \times 20$ grid differs by less than 1% as will be seen later in Fig. 1(b), and has been used throughout the study.

3. Optimization technique

To maximize the thermal performance of a microchannel heat sink, we optimized the channel shape using the random search technique [7]. The objective function to be minimized is the thermal resistance, a commonly used quantity in measuring the heat-sink performance,

$$\theta = \frac{\Delta T_{\max}}{\dot{q}} = \frac{(T_{w,\max} - T_{f,in})}{\dot{q}}, \tag{4}$$

where $T_{w,\max}$ and $T_{f,in}$ are the maximum wall temperature and the inlet fluid temperature, respectively, and $\dot{q}$ is the total heat-flow rate. The geometric parameters of a microchannel heat sink such as the channel depth H, the channel width w_c, and the channel number N are chosen as the design variables. The procedure to find the optimal vector $\mathbf{X}$, which is composed of design variables, is briefly described below.

Starting with the specified set of design variables, $\mathbf{X}^0$, a new set of values is obtained at each iterative step:

$$\mathbf{X}^n = \mathbf{X}^{n-1} + l\mathbf{S}^n, \tag{5}$$

where n is the iteration number and $\mathbf{S}$ is the directional vector in the design space. The scalar length l defines the distance that we wish to move in the direction of $\mathbf{S}$. The direction $\mathbf{S}$ at the new iteration (n^{th}) is randomly chosen while the distance l is determined to yield the minimum objective function in that direction. A local $l - \theta$ relation is obtained by fitting a cubic polynomial through four sets of (l, θ) in the direction of S. The value l that makes θ minimum can then be calculated from the polynomial. The process is repeated until the objective function stays within the specified limit between the two successive iterations. This optimization is performed under two constraints, i.e., (1) the pumping power is constant and (2) the channel aspect ratio is less than 10 as the microchannel heat sink with a large aspect ratio is difficult to fabricate.

4. Results and discussion

We consider the heat sink which has a square horizontal surface ($L \times W$), 1 cm × 1 cm. The water ($\rho = 1000$ kg/m^3, $\mu = 0.001$ kg/ms, $k_f = 0.613$ W/m K, $c_p = 4179$ J/kg K) and the silicon ($k_s = 148$ W/m K) are used as the coolant and the heat-sink material, respectively. To validate the numerical procedure, the thermal resistance is compared with the experimental data [1] for various channel geometries and pressure drops, in Table 1. The two are in good agreement as the discrepancy is less than 4%; this confirms that the procedure is adequate for thermal analysis.

Before carrying out the optimization procedure, the calculation is performed for the typical shape and operating conditions to see the flow characteristics. The channel with the aspect ratio of 10 is given the width of 45.3 μm and the fin thickness of 35.3 m. This amounts to 124 fins on the component. The substrate thickness is 100 μm and the uniform heat flux of 10^2 W/cm^2 is applied at the bottom surface. The pumping power is 2.56 W and results in the volume flow rate of 10.02 cm^3/s and the Reynolds number of 364, which is based on the hydraulic diameter of the channel.

The wall temperature distribution, $T_{w,c}$, along the channel centerplane which is monotonically increasing is presented in Fig. 1(b). Also compared in the figure are the result obtained by the grid which is twice as dense in each direction, and that for the thermally fully developed flow, in which the normalized temperature profile across the channel stays unchanged in the streamwise direction. The agreement of the results for two different grids is excellent and thus confirms that the present grid is fine enough to resolve the thermal field. The discrepancy between the three-dimensional result and the fully developed one signifies the entrance effects of the thermal field. The discrepancy is larger in the inlet region, where the thin thermal boundary layer is still in the developing stage. The temperature field can be considered fully developed when the rate of temperature rise becomes nearly constant at $x/L \approx 0.5$. The three-dimensional characteristic of the entry region is also observed in the section Nusselt number ($Nu = q_w D_h/(k_f(T_w - T_m))$) plot in Fig. 1(b). Due to the thin thermal boundary layer in the developing region, the heat transfer is more active, i.e., higher Nu, and the substrate temperature becomes lower than that for the thermally fully developed flow. These results show that the thermal entrance effects should not be neglected when the working fluid is water ($Pr \approx 7$). If neglected, this effect would result in an error of 15% for the averaged Nusselt number for the case shown in Fig. 1(b).

The procedure is then used to obtain the optimal shape of a fin–channel heat sink. The convergence history for H, w_c, and N is given in Fig. 2. The procedure is fairly efficient as the computing time on Pentium III (800 MHz) is less than 30 min. It is seen that the converged solution can be obtained within 20 iterations, despite the initial geometry being far from the final one. In the initial stage of computation, the number of the channel is allowed to assume a floating-point number. Once the calculation reaches the point when this number does not vary much, the number is rounded to the nearest integer and fixed for further iteration to obtain the optimal shape. The optimal values for the design variables turn out to be $N = 124$, $w_c = 45.3$ μm, and $H = 10w_c$ for $\delta_{sub} = 100$ μm. The corresponding thermal resistance is 0.069 as shown in Fig. 2.

We varied the number of channels and obtained the associated optimal design variables and the thermal resistance. This is to see the sensitivity of the thermal resistance to the channel number and the results are given in Fig. 3. As seen in Fig. 3(a), if the channel number is cut in half, the thermal resistance increases about 15% from the optimal value. The optimal design variables for various N are shown in Fig. 3(b) and exhibit some interesting features. Generally, these increase as the channel number decreases. The fin thickness appears to be more sensitive than the channel width to the change in N. The fact that the optimal channel width varies only mildly prompts us to examine the behavior of the thermal resistance for various N but fixing either the fin

Table 1
Thermal resistance for various geometric shapes and operating conditions

	w_c (μm)	w_w (μm)	H (μm)	Δp (kPa)	θ (°C/W)		
					Experimental data [1]	Numerical solution	Deviation (%)
1	56	44	320	103.42	0.110	0.111	1.0
2	55	45	287	117.21	0.113	0.117	3.4
3	50	50	302	213.73	0.090	0.092	2.2

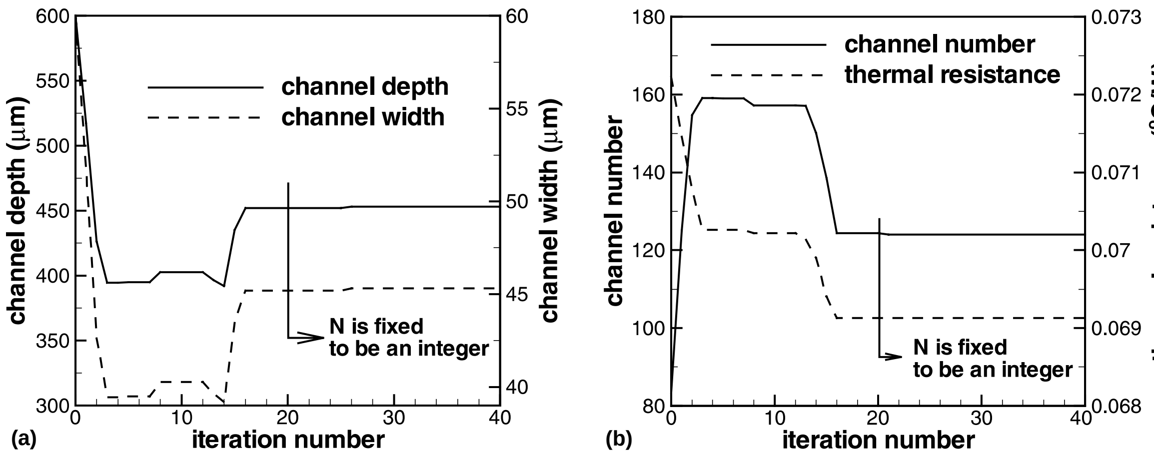

Fig. 2. Convergence history toward the optimal values: (a) geometric variables; (b) channel number and thermal resistance.

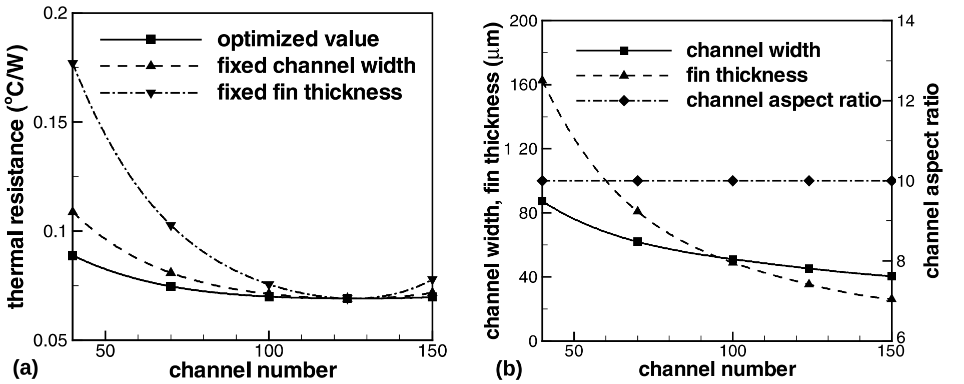

Fig. 3. Effect of the channel number on the thermal performance of a microchannel heat sink: (a) thermal resistance; (b) optimal geometric variables for various channel numbers.

thickness or the channel width to the optimal values for $N = 124$. The channel depth and the volume flow rate of the coolant are allowed to vary as the channel width changes, though. The results compared in Fig. 3(a) show that the performance is much better when the channel width is fixed than when the fin thickness is kept constant. It is seen that, as the channel number decreases, the thermal resistance is almost doubled from its lowest value. Considering the fact that the channel depth also needs to be adjusted to obtain this result, changing the channel width does not seem to be very effective. On the other hand, a good result can be achieved by adjusting only the fin thickness while the channel width is kept constant.

Finally, a series of calculations is carried out for various pumping powers. The results, presented on a log–log scale in Fig. 4, show that both the optimal dimensions and thermal resistance have a power-law de-

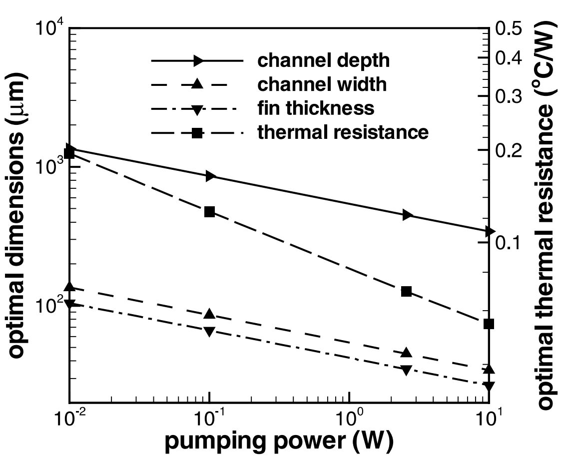

Fig. 4. Effect of the pumping power on the thermal performance of a microchannel heat sink.

pendence on the pumping power. Varying the size of the heat sink yields similar behavior as it alters *Re* only, which is the only relevant dimensionless parameter when *Pr* is fixed. Many engineers may find these simple but intriguing results useful in the practical heat-sink design.

5. Conclusions

A three-dimensional analysis procedure for the thermal performance of a microchannel heat sink has been developed and used to obtain the optimal fin-channel shape that minimizes the thermal resistance. Due to the thin thermal boundary layer in the developing region of the microchannel, the thermal entrance effect is substantial when the working fluid is water ($Pr \approx 7$). For a given pumping power, the channel depth, the channel width, and the fin thickness are varied under the constraint that the channel aspect ratio should be less than 10. Among various design variables, the channel width appears to be the most crucial quantity in dictating the performance of a microchannel heat sink. It is also found that the optimal channel shape remains relatively unchanged even when the channel-number variation is large and that both the optimal dimensions and thermal resistance have a power-law dependence on the pumping power.

Acknowledgements

This work was supported by KISTEP under Grant 2-578 through the National Research Lab. Program and, in part, by the Ministry of Education under the BK21 Project.

References

[1] D.B. Tuckerman, R.F.W. Pease, High-performance heat sinking for VLSI, IEEE Electron Dev. Lett. 2 (1981) 126–129.
[2] R.J. Phillips, Microchannel heat sinks, in: Advances in Thermal Modeling of Electronic Components and Systems, vol. 2, ASME Press, New York, 1990 (Chapter 3).
[3] R.W. Knight, J.S. Goodling, D.J. Hall, Optimal thermal design of forced convection heat sinks – Analytical, ASME J. Electron. Packag. 113 (1991) 313–321.
[4] J.C.Y. Koh, R. Colony, Heat Transfer of microstructure for integrated circuits, Int. Commun. Heat Mass Transfer 13 (1986) 89–98.
[5] S.J. Kim, D. Kim, Forced convection in microstructures for electronic equipment cooling, ASME J. Heat Transfer 121 (1999) 639–645.
[6] R.K. Shah, A.L. London, Laminar flow forced convection in ducts, Adv. Heat Transfer (Suppl. 1) (1978).
[7] G.N. Vanderplaats, in: Numerical optimization techniques for engineering design, McGraw-Hill, New York, 1984, pp. 71–87.