

A Comparative Analysis of Studies on Heat Transfer in Small-scale Channels

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Abstract: The high rates of heat transfer obtained by using micro or small channels makes them an attractive alternative to conventional methods of heat dissipation, especially in microelectronics cooling system. In this paper, three-dimensional fluid flow and heat transfer phenomena for water in rectangular, elliptic and triangular small-scale channels is investigated. In all computational region this paper improved the Boussinesq approximation and the physical properties of the coolant are changing with its temperature. The pressure drop and the temperature increasing of the coolant in three different form channels are obtained. The numerical simulation shows that the small rectangular channels' heat transfer and flow resistance characteristics are optimal.

Keywords: Micro electronics cooling, small scale channels, computational fluid dynamics, Conjugate heat transfer

1. INTRODUCTION

With the development of new high performance chips, the ability to remove heat from the high heat flux region around chips becomes an important factor in the design of reliable computing systems. The conventional air-cooling techniques are unlikely able to meet the cooling needs. During the last decade, several novel cooling schemes have been proposed and extensively studied both experimentally and theoretically. The first alternative is liquid cooling, and this cooling system will consist of a compact heat sink embedded with microchannels.

Recent advances in microfabrication techniques have led to revival of the concept of the ultra-compact microchannel heat exchangers for electronic cooling which was originally introduced by Tuckerman and Pease^[1,2]. In principle, such microchannels can be machined at the back of a substrate of electronic chips and, therefore, can reduce the internal (contact) thermal resistance of the heat sink practically to zero. The cooling rates in such microchannel heat exchangers should increase significantly due to a decrease in the convective resistance to heat transport caused by a drastic reduction in the thickness of thermal boundary layers. The over all excellent potential capacity of such a heat sink for heat dissipation is based on the large heat transfer surface-to-volume ratio of the microchannel heat exchangers.

The flow and heat transfer in microchannels have become a subject of growing research attention since the early 1990s. Several pioneering experimental studies^[3-5] found significant deviation of the flow parameters in the microchannels compared to those in the conventional 'large' channels. In particular, the authors reported a decrease in the friction factor for the gas flow in the microchannels relative to its value

predicted using standard correlations for laminar flow. The review of these works has been recently presented by Peng and Wang^[6]. In contrast, Webb and Zhang^[7] claim that the accepted single-phase flow correlations adequately predict their experimental data for round and rectangular tubes with hydraulic diameter of the order of 1 mm. Furthermore, as Nakayama^[8] have pointed out, the geometric precision and tolerance control in the experimental assembly is often beyond the manual skills of the experimenter, and this leads to the considerable experimental uncertainty and inconsistencies in the reported measurements. Unfortunately, we were able to identify only few papers that provide the experimental data for friction and heat transfer in the microchannels under conditions which are directly relevant to those found in the integrated circuits of modern electronic devices, i.e., when the substrate (heat spreader) is highly conducting and massive in comparison to channel dimensions and, therefore, the conjugate conduction-convection effects are important. A new type of heat sink combining microchannels and microheat pipes is therefore possible, and its heat transfer characteristics need to be studied. Few attempts have been made to analyze the microchannel flows and heat transfer, theoretically. In particular, Weisbergetal.^[9] numerically analysed a conjugate heat transfer problem in the microchannel heat exchanger. Yu et al.^[10] performed a scale analysis of the flow and heat transfer characteristics in microtubes. Several new theoretical models were also proposed recently in order to describe the peculiar features of the microscale flow and heat transfer. Lietai.^[11] utilized the kinetic theory of gases to derive the effective fluid transport properties close to the wall surface and, then, to obtain analytical expressions for the friction factor and the Nusselt number. Malaetai.^[12] and Yangetai.^[13] included the surface electrokinetic effects in the analysis of the

flow and heat transfer in microchannels. However, due to the lack of a clear fundamental understanding of the transport mechanisms on microscale, most of these theories appear to be rather speculative and, therefore, have not gained a wide acceptance. Chenetal. [14] studied the possibility of using the Navier-Stokes equations for compressible gas in continuum flow for mulation together with the slip and non-slip boundary conditions in order to describe gas flow in microchannels. Recently, Fedorov and Viskanta [15] extended the efforts to analyse the transient dynamics of conjugate heat and mass transfer in microchannels in the presence of the surface chemical interactions.

It is clear that successful design of the microchannel based heat sinks requires fundamental understanding of the transport processes occurring on the microscale. In this paper, we employ the numerical simulations to gain insight into combined convection-conduction heat transfer in the microchannel. To this end, the system of three-dimensional Navier-Stokes equations for conservation of mass, momentum, and energy based on the continuum flow assumption is used as a mathematical model of the process. First, the validity of the developed mathematical model is critically assessed by comparing the theoretical predictions with available measurements of the friction factor coefficient and thermal resistance in the microchannel heatsink. Then, the local temperatures are calculated with the aim to identify the heat flow pattern established in the microchannel under the strong influence of three-dimensional conjugate effects. Finally, the average heat transfer characteristics are also obtained, and important practical recommendations are provided regarding the over all heat sink performance and possible locations of the structural failure due to excessive thermal gradients.

2. ANALYSIS

2.1 Physical model, model equations, and boundary conditions

Consider a three-dimensional asymmetric rectangular channel shown schematically in Fig. 1. The channel geometry shown in table1. A theoretical model for flow and conjugate heat transfer in the

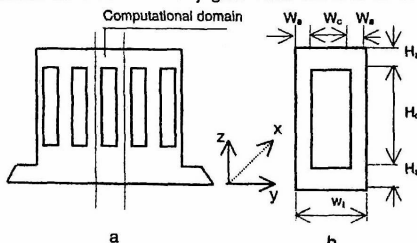


Fig. 1. Schematic of microchannels

model for flow and conjugate heat transfer in the small channel is developed under the following assumptions:

- 1) the hydraulic diameter of small channels under development is 0.5-1.0mm. This yields a typical Knudsen number for water as a coolant to be between 3.5×10^{-6} and 3.5×10^{-5} , which lies in the continuous flow region ($Kn < 10^{-3}$) [16]. Hence, the conservation equations based on the continuum model (Navier-Stokes equations of motion) can still be used to describe the transport processes
- 2) the transport processes are considered to be steady-state and three-dimensional
- 3) the flow is incompressible
- 4) thermal radiation is neglected
- 5) the development of the flow and temperature fields at the inlet of the channel is carefully resolved since it is the region where the largest temperature gradients and, therefore, thermal stresses are expected to occur
- 6) the thermo physical properties are temperature dependent.

Table 1: Parameters of rectangular small channels (mm)

H _c	W _c	H _t	W _t	L _t	H _s	W _s	d _s
0.9	0.45	1.8	0.9	30	0.9	0.225	0.6

Under the stated assumptions, the conservation equations of mass, momentum, and energy for the fluid and the solid channel walls can be written as follows (see Fig.1 for coordinate system):

Conservation of mass (continuity)

$$\frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} = 0 \quad (1)$$

Conservation of momentum

$$\frac{\partial(\rho u u)}{\partial x} + \frac{\partial(\rho u v)}{\partial y} + \frac{\partial(\rho u w)}{\partial z} = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x} \left(\mu \frac{\partial u}{\partial x} \right) + \frac{\partial}{\partial y} \left(\mu \frac{\partial u}{\partial y} \right) + \frac{\partial}{\partial z} \left(\mu \frac{\partial u}{\partial z} \right) \quad (2)$$

$$\frac{\partial(\rho v u)}{\partial x} + \frac{\partial(\rho v v)}{\partial y} + \frac{\partial(\rho v w)}{\partial z} = -\frac{\partial p}{\partial y} + \frac{\partial}{\partial y} \left(\mu \frac{\partial v}{\partial x} \right) + \frac{\partial}{\partial y} \left(\mu \frac{\partial v}{\partial y} \right) + \frac{\partial}{\partial z} \left(\mu \frac{\partial v}{\partial z} \right) + \rho g \quad (3)$$

$$\frac{\partial(\rho w u)}{\partial x} + \frac{\partial(\rho w v)}{\partial y} + \frac{\partial(\rho w w)}{\partial z} = -\frac{\partial p}{\partial z} + \frac{\partial}{\partial z} \left(\mu \frac{\partial w}{\partial x} \right) + \frac{\partial}{\partial y} \left(\mu \frac{\partial w}{\partial y} \right) + \frac{\partial}{\partial z} \left(\mu \frac{\partial w}{\partial z} \right) \quad (4)$$

Conservation of energy

$$u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} = a \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) + S \quad (5)$$

A uniform heat flux is imposed as a boundary condition at the heated bottom-wall, and the other outside walls of the channel are insulated. The velocity and temperature of the fluid entering the channel are specified. A no-slip boundary conditions to couple the energy equations for the fluid and solid phases.

2.2 Properties of water

As mentioned earlier, some properties of water allowed varying with temperature. The functional relations between density, specific heat, thermal conductivity and viscosity with temperature are

$$\rho = a_0 + a_1 T + a_2 T^2 + a_3 T^3 \quad (6)$$

$$C_p = b_0 + b_1 T + b_2 T^2 + b_3 T^3 \quad (7)$$

$$\lambda = c_0 + c_1 T + c_2 T^2 + c_3 T^3 \quad (8)$$

$$\nu = d_0 + d_1 T + d_2 T^2 + d_3 T^3 + d_4 T^4 + d_5 T^5 \quad (9)$$

In Eqs. (6)-(9), thermal conductivity λ is in W/mK, viscosity ν is in m²/s and temperature T is in K respectively. The various constants are listed in Table 2.

Table 2: Constants in properties vs. temperature correlations

x	a _x	b _x	c _x	d _x
0	291.75	7965.51	-2.33	1.454
1	6.21	-30.88	0.021	-0.02087
2	-0.017	0.081	-4.89E-5	1.20155E-4
3	1.38E-5	-6.81E-5	3.76E-8	-3.46E-7
4				4.99E-10
5				-2.88E-13

2.3 Method of solution

in this paper, the characteristics of heat transfer and fluid flow are analysed by CFD. The FVM is used to solve the continuity, momentum and energy equations. The power-law scheme is used to model

the combined convection-diffusion effect in the transport equations. The SIMPLER method is used to resolve the pressure-velocity coupling. In the present study, a solution is deemed converged when the energy imbalance in the energy equation is less than 10^{-8} .

3. RESULTS AND DISCUSSION

3.1 Model validation

The numerical algorithm and computer program were carefully evaluated by comparing model predictions with available experimental data for friction and conjugate heat transfer in microchannels [17-21]. Generally, the heat exchanger has many identical

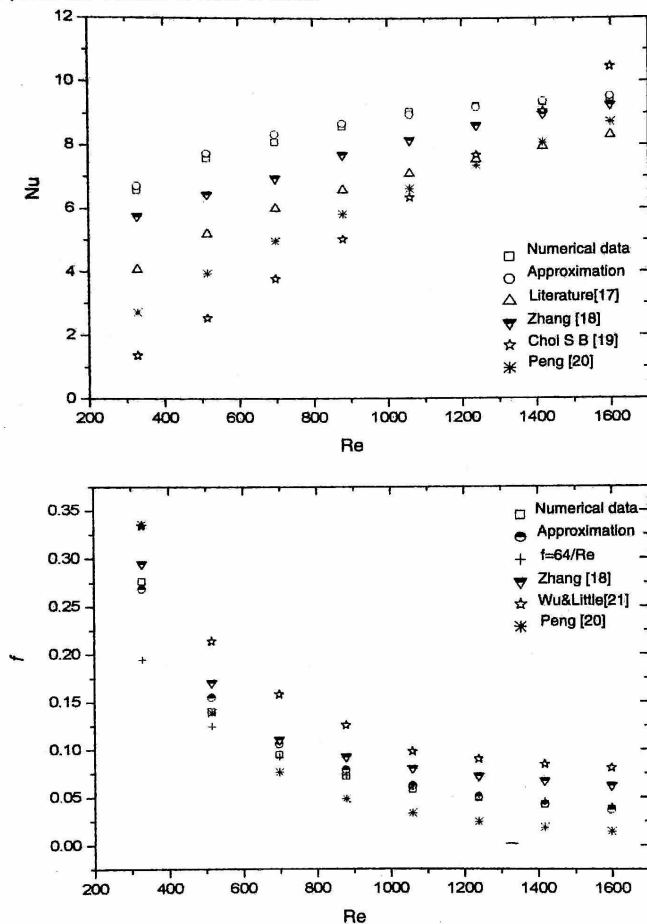


Fig. 2. Comparison of the numerical results and experimental data friction and the convective heat transfer Nusselt numbers for water flow in the copper small channels

small channels, so that the unit cell can be represented by the asymmetric rectangular channel as shown in Fig. 1. The channel's geometrical dimensions of the unit cell are summarized in Table 1 (see Fig.1 for notation). In the numerical study, water was used as a working fluid with the inlet temperature (T_0)=15°C. A uniform heat flux (q) of 100W/cm² was imposed at the outer bottom-wall of the rectangular channel (see Fig. 1.), while all other external surfaces were insulated.

Fig. 2 compares the numerical results and the experimental data. Clearly, a good agreement for between the numerical and experimental has obtained. In calculation of the friction coefficient the numerical results and the experimental data are very close. The Nusselt number appears to be slightly under predicted. Particularly at the small Reynolds numbers. The experimental data and our numerical results show an approximately increase trend with an increase in Re.

3.2 temperature distribution

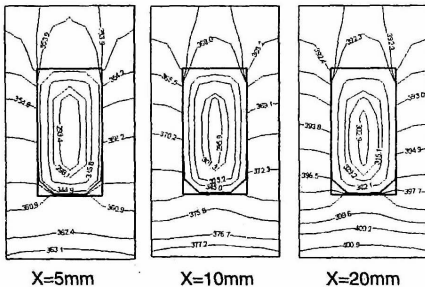


Fig.3. Temperature distributions at three different cross-sections along the channel

The temperature distributions at three y-z cross-sections along the channel are shown in Fig. 3. It can be seen that the average temperature of the solid is lowest at x=5mm, that is to say, the intensity of heat transfer (heat fluxes) between solid and fluid is in its peak in the inlet portion of the channel. It also can be seen that the heat flux, imposed at the bottom wall only, is immediately spread out by conduction within the copper substrate and eventually is transferred to the fluid through all four inner walls of the channel.

3.3 comparative on forms of small channels

Employing microchannels is a good way to cool electronics for its large surface-to-volume ratio. However, what is the best form? In this paper, rectangular, triangular and elliptic small channels are studied to compare their resistance and heat transfer characters.

Fig.4 shows the schematic drawing numerical configuration. Heat, supplied to the copper substrate through bottom wall, is removed by flowing water through internal small channels. The channel's detailed geometrical dimensions of the unit cell are summarized in Table 3 (see Fig.4 for notation). The

boundary conditions are same just as the rectangular channels mentioned earlier.

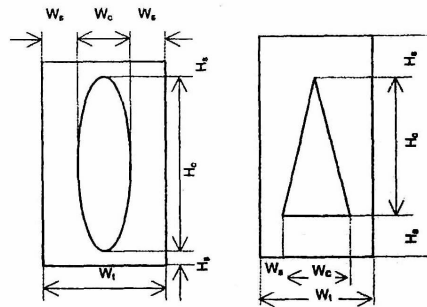


Fig.4. Schematic of elliptic and triangular microchannels

Table 3: Parameters of three different form micro-channels (mm)

	Rectangular	Elliptic	Triangular
H_c	1.5	1.6	2.4
W_c	0.75	0.8	1.2
H_t	3.0	3.2	4.8
W_t	1.5	1.6	2.4
L_t	50	50	50
H_s	0.75	0.8	1.2
W_s	0.375	0.4	0.6
d_e	1.0	1.038	0.937

To eliminate other factor influence on the heat transfer and the resistance, the equivalent diameters of the three different form small channels are almost identical. A uniform heat flux are set to compare conveniently, and the other boundary conditions are also identical.

Fig.5 shows the pressure drop of fluid for removing equal heat per mass coolant in laminar and turbulent flow respectively. It can be seen from the schematics that the pressure drop in triangular small channel is maximal, while the pressure drop in rectangular channel is minimal. That is to say, in the all region, when dissipating equal heat the resistance in rectangular is minimal and the flow channel need the power is least.

Fig.6 shows the fluid temperature increase for removing equal heat per mass coolant in laminar and turbulent flow respectively. It can be seen that the fluid temperature increases significantly with removing more heat flux per mass water. Especially the water in triangular small channel increases apparently. In the all region, the rectangular channel's fluid temperature increase is minimal.

Fig.7 shows the solid temperature increase for removing equal heat per mass coolant. It can be seen from the left drawing that the solid temperature increase in triangular channel is evident. In the right one it can be seen that the increasing in rectangular and elliptic channel is very close. The temperature

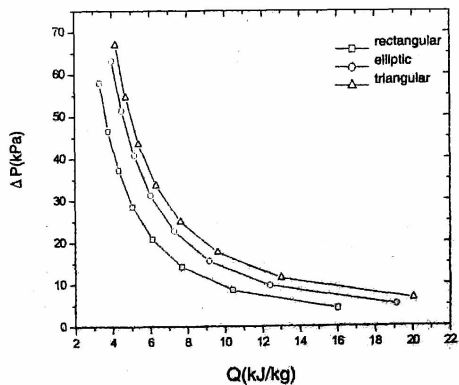
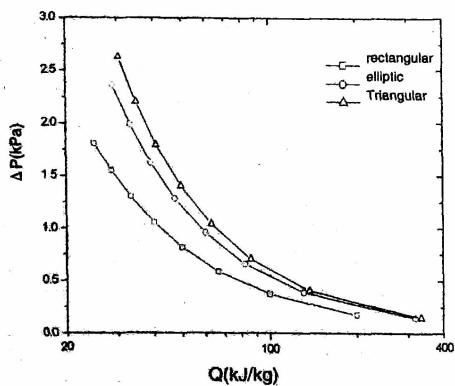


Fig.5 Pressure drop for removing equal heat per mass coolant

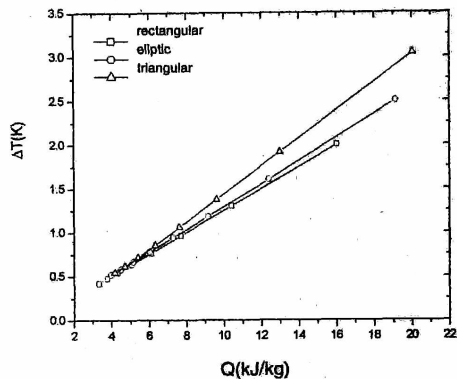
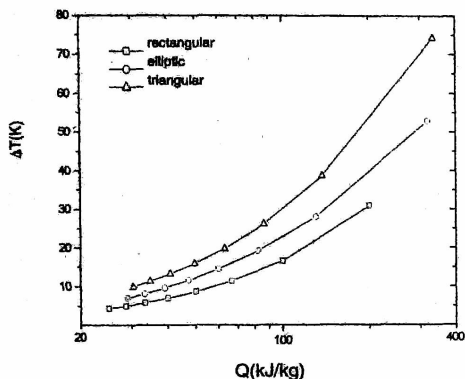


Fig.6 Fluid temperature increase for removing equal heat per mass coolant

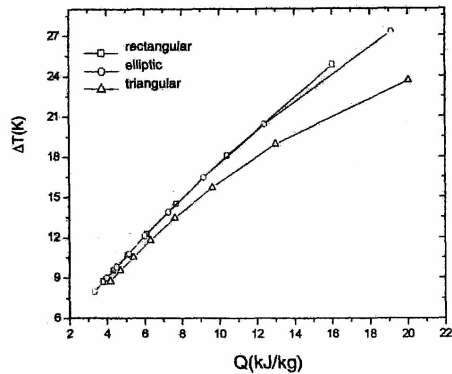
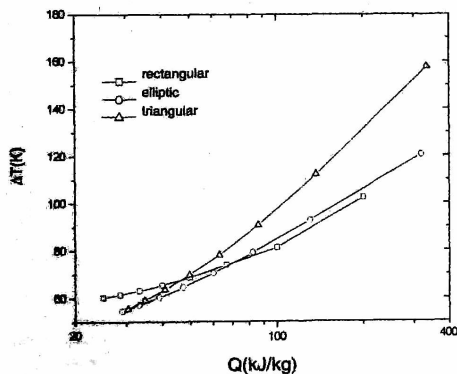


Fig.7 Solid temperature increase for removing equal heat per mass coolant

increase in triangular channel is small comparatively; it may be caused by its configuration.

The numerical simulation shows that the small rectangular channels' heat transfer and flow resistance characteristics are optimal. One may utilize the small rectangular channels in cooling micro electronic system.

4. CONCLUSION

The fluid flow and heat transfer inside small channels heat sink has been numerical studied. Our numerical method is valid by comparing with several literates' investigation. Especially the friction coefficient is very close to these literatures; while the heat transfer Nusselt number is slightly disagree with part literature in small Reynolds number.

Three different form small channels are numerical analysed. When removing equal heat the pressure drop in rectangular is less and the fluid temperature increase is also minimal. The heat transfer and resistance characters of the rectangular small channel are optimal in the three form channels.

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