# NUMERICAL STUDY OF MICROSCALE HEAT SINKS USING DIFFERENT SHAPES & FLUIDS

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**Abstract:** In this paper the COMSOL Multiphysics module was used in integration with Fluid Flow & Conjugate Heat transfer Module. Investigations have been conducted to better understand & establish the fluid flow & heat transfer characteristics in different micro channel heat sinks. These micro channel heat sinks combines various attributes & highly sophisticated Process using micro scale heat transfer. The high heat transfer to volume ratio is another significant feature which makes them favourable over other solutions in same heat transfer removal range. The present study deals with different shapes & materials of the micro channels(rectangular, trapezoidal etc) & different properties of the Liquid Coolants using certain custom liquids. Liquid cooling promises to be a more compact arrangement and its use has been reported recently for cooling the central processing unit of a large computing system. Different parameters like heat flow rate, heat transfer coefficient, outlet temperature are studied with different flow rates and an optimal best range of flow rate for all the fluids separately by considering maximum heat flow rate, minimum base temperature and less pumping power has been considered.

**Key Words:** Microchannel; Single-phase; Friction factor; Liquid

## 1 Introduction

With the improvement of computational speed, thermal management becomes a serious concern in computer system. CPU chips are squeezing into tighter and tighter spaces with no more room for heat to escape. Total power-dissipation levels now reside about 110 W and peakpower densities are reaching 400–500 W/mm$^2$ and are still steadily climbing. As a result, higher performance and greater reliability are extremely tough to attain. But since the standard conduction and forced-air convection techniques no longer be able to provide adequate cooling for sophisticated electronic systems, new solutions are being looked into liquid cooling, thermoelectric cooling, heat pipes, and vapor chambers. Compared with air-cooling, Liquid-cooling is a much more efficient way for drawing heat away from the processor and outside the system. With forced flowing of the convective liquid, an improvement over air-cooling was found by manyfolds. However, the use of liquid as cooling fluid has some limitations. The low thermal conductivity of water may lower its effectiveness as a heat transfer fluid. Also, circulation of water can be driven only by mechanically moving pumps that may be unreliable, occupy large spaces, and contribute to vibration or noise.

Since the pioneering work of Tuckerman and Pease [1] in 1981,many studies have been conducted on micro-channel heat sinks as summarized by Phillips [2] and more recently, by Morini [3]. The need for cooling in high power dissipation (100 W/cm$^2$) systems in several scientific and commercial applications such as microelectronics require something beyond the conventional cooling solutions. A number of studies have investigated the thermal design optimization of micro-channel heat sinks to determine the geometric dimensions that give optimum performance.

The entire microchannel heat sink should be used as the computation domain instead of a single unit cell. Lots of researches have been performed to better understand the cooling mechanism and improve the cooling capability of mini/micro channel, which are mainly classified as theoretical analysis [4], numerical simulation [6-[7], structure optimization, fabrication technology, practical application. In addition to the various interfacial effects discussed above, the cross-sectional shape of the channel can have a great influence on the fluid flow and heat transfer inside noncircular microchannels was experimentally confirmed by Wu and Cheng [8].

## 2 Nomenclature

| Dimensional Parameter | |
|---|---|
| $w_c$ | Width of microchannel |
| $h_c$ | Height of microchannel |
| a | Aspect ratio |
| $h_d$ | Hydraulic diameter |
| L | Length of microchannel |
| $W_c$ | Width of flow channel |
| $H_c$ | Height of flow channel |
| p | Length of parallel side 1 in trapezoidal channel and mixed geometry |
| q | Length of parallel side 1 in trapezoidal channel and mixed geometry |
| θ | Angle of inclination of geometry with vertical |
| a | Aspect ratio |
| $A_c$ | Area of flow cross-section |
| $A_b$ | Area base of microchannel |
| $A_w$ | Area of flow channel surface |
| Other Parameters | |
| k(W/mK) | Thermal conductivity |
| $\rho(kg/m^3)$ | Density |
| μ(Pa-s) | Dynamic viscosity |
| $C_p$(J/molK) | Molar Heat capacity |
| $U(m/s^2)$ | Magnitude of velocity |
| γ(Cp/Cv) | Heat capacity ratio |
| Re | Reynolds number |
| F | Friction factor |

| | |
|---|---|
| Nu | Nusselt number |
| $q''(W/m^2)$ | Heat flux |
| $h(W/m^2K)$ | Heat transfer coefficient |
| Q(W) | Total heat load |
| Tw(K) | Temperature of the channel wall |
| Tf(K) | Average temperature of the fluid |
| $P(N/m^2)$ | Pressure |
| $\Delta T_{avg}(K)$ | Average temperature difference between wall and fluid |
| $F(N/m^3)$ | Body force |

## 3 Numerical Model

. The particular focus of this study is the shape arrangement effects on the fluid flow and heat transfer inside the heat sinks. All of the geometric dimensions of these heat sinks are the same except the inlet/outlet locations. The Mathematical model consist of heat sink including the inlet/outlet ports, inlet/outlet plenums, and microchannels. The microchannels are the efficient means of heat transfer. Since there are lot of constraints in the geometry, the performance of the micorchannels varies a lot with different geometrical shapes. The microchannels can be circular, rectangular or any other custome easy to fabricate shape. But considering the fabrication constrainst, only two shapes are considred, i.e rectangular & trapezoidal. Study of different flow section namely, Rectangular, Trapezoidal been done using comsol multiphysics simulation considering all other aspects constant. Apart from this microchannel has been tested for two different fluids i.e. water and ethylene glycol. Since fluids have significant impact on the heat transfer & in combination with the geometrical properties of the channels, the fluid properties can be optimized.

Geometric dimensions of these heat sinks are calculated in such a way that for different aspect ratio the hydraulic diameter is same. Different design points are considered so that data can be collected over a wide range. Because of the difference in inlet velocity, the resultant flow fields and temperature distributions inside these heat sinks are also different under a given heat flux applied at the bottom of the heat sink. Using the thermal resistance, overall heat transfer coefficient and pressure drop coefficient to quantify the heat sink performance, it is also found these heat sinks have better performance among the heat sinks studied.

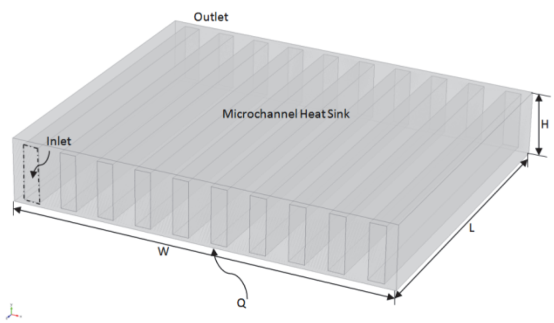
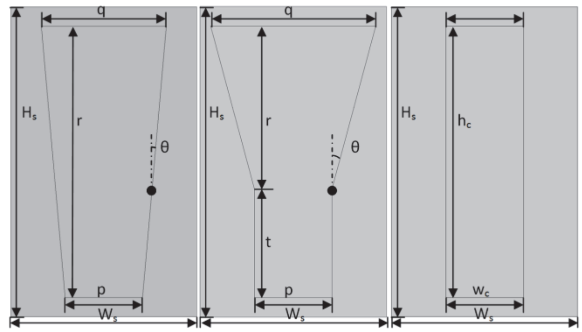



Figure 1 Microchannel heat sink W=12mm, L=10mm & 2mm

Microchannel study is carried under using following assumptions.

- Steady state
- Incompressible fluid.
- Laminar flow.
- Constant properties of both fluids and solid.

Effects of viscous dissipation are negligible Analysis has been done using three-dimensional steady and laminar flow. The computational domain is taken as the entire heat sink including inlet/outlet ports. The 3D-Model has been designed assuming the conjugate heat transfer condition for microchannel heat sink using Comsol multiphysics. A uniform heat flux of $100W/cm^2$ is applied at the bottom surface of the microchannel heat sink. Heat transfer in unit cell is a conjugate problem which combines heat conduction through the solid and dissipated away by convection of the cooling fluid in the microchannel.




2(a) 2(b) 2(c)

Figure 2: Microchannel sections, Trapezoidal, Trapezoidal & Rectangular Mix Section, Rectangular (L=10mm, W=12mm & H=2mm)

The Microchannel heat sink model consists of Substrate of length L having different flow sections as represented in 2(**a**) **2(b) 2(c)**

**Figure 2**. The bottom surface of heat sink at x=0 is uniformly heated with a constant heat flux and top surface at x=H, is well insulated, also the adiabatic conditions are applied at the other boundaries. The flow is assumed to be laminar and both hydro-dynamically and thermally fully developed.

The different configurations of the geometry for rectangular channel are shown in table 1.

Table 1 Parameters of Rectangular channel

| S/No | $w_c$(mm) | (a) | $h_c$ (mm) | $h_d$(mm) |
|---|---|---|---|---|
| 1. | 0.5 | 2 | 1 | 0.66 |
| 2. | 0.5 | 2.5 | 1.25 | 0.71 |
| 3. | 0.5 | 3 | 1.5 | 0.75 |
| 4. | 0.5 | 3.5 | 1.75 | 0.77 |

The different configurations of the Trapezoidal section are shown in table 2.

Table 2 Parameters of Trapezoidal channel

| S/No | h(mm) | θ(deg) | p(mm) | q(mm) | d(mm) |
|---|---|---|---|---|---|
| 1. | 1 | 5 | 0.41 | 0.58 | 0.67 |
| 2. | 1.25 | 5 | 0.39 | 0.61 | 0.70 |
| 3. | 1.5 | 5 | 0.38 | 0.61 | 0.71 |
| 4. | 1.75 | 5 | 0.38 | 0.60 | 0.71 |

The different parameters in the combiner section are shown in Table 3.

Table 3 Parameters of Rectangular and trapezoidal Mix

| S/No | t(mm) | r(mm) | θ(deg) | p(mm) | q(mm) | d(mm) |
|---|---|---|---|---|---|---|
| 1 | 0.7 | 0.3 | 15 | 0.50 | 0.67 | 0.67 |
| 2 | 0.7 | 0.55 | 15 | 0.47 | 0.77 | 0.71 |
| 3 | 0.7 | 0.8 | 15 | 0.4 | 0.85 | 0.72 |
| 4 | 0.7 | 1.05 | 15 | 0.37 | 0.93 | 0.72 |

### 3.1 Heat transfer

The Heat Transfer in Solids model uses the heat equations as the mathematical model for heat transfer in solids:

$$\rho C_p u.\nabla T = \nabla.(k\nabla T) + Q \qquad (1)$$

For a steady-state problem the temperature does not change with time and the first term disappears. It has these material properties: density ρ, heat capacity Cp, thermal conductivity k (a scalar or a tensor if the thermal conductivity is anisotropic), and Q, which is the heat source (or sink) one or more heat sources can be added separately.

Table 4 Properties of substrate material.

| Property | Copper | Aluminum |
|---|---|---|
| k(W/m-K) | 401 | 205 |
| $\rho(kg/m^3)$ | 8960 | 2700 |
| $C_p$(J/molK) | 24.440 | 24.200 |

### 3.2 Thermal Insulation

Thermal Insulation means that there is no heat flux across the boundary: Here Thermal insulation is provided on the top, right, left, inlet and outlet sides of the channel.

$$- n.(k\nabla T) = 0 \qquad (2)$$

This condition specifies where the domain is well insulated. Intuitively this equation says that the temperature gradient across the boundary must be zero. For this to be true, the temperature on one side of the boundary must equal the temperature on the other side. Because there is no temperature difference across the boundary, heat cannot transfer across it.

### 3.3 Walls

**u**=0, Boundary condition: No Slip

### 3.4 Fluid

Continuity, momentum and energy equation.

$$\nabla.(\rho u) = 0 \qquad (3)$$

$$\rho(u.\nabla)u = .[-pI + \mu(\nabla u + (\nabla u)^T) - \frac{2}{3}\mu(\nabla.u)I] + F \quad (4)$$

$$\rho C_p u.\nabla T = \nabla.(k\nabla T) + Q \qquad (5)$$

Table 5 Properties of coolant fluid.

| Property | Water |
|---|---|
| K(W/m-K) | 0.60405 |
| $P(Kg/m^3)$ | 1000 |
| $C_p$(kJ/KgK) | 4.18 |
| Γ(Cp/Cv) | 1 |
| μ(Pa-s) | .00147 |

### 3.5 Initial value

u=0,P=0

T=293.15K

### 3.6 Inlet

$$P=P_0, [\mu(\nabla u + (\nabla u)^T) - \frac{2}{3}\mu(\nabla.u)I].n = 0 \qquad (6)$$

$P_0$=10kPa, 50KPa, Boundary condition: Pressure, no viscous stress, $T_0$=293.15K

### 3.7 Outlet

$$P=P_0, [\mu(\nabla u + (\nabla u)^T) - \frac{2}{3}\mu(\nabla.u)I].n = 0 \qquad (7)$$

$P_0$=0,Boundary condition: Pressure, no viscous stress

### 3.8 Heat flux

$$n.(k\nabla T) = q_0, q_0 = 100 Wcm^{-2} = q'' \quad (8)$$

Temperature $T_0$=293.15K

Reynolds number is defined as follows

$$Re = \frac{\rho u l}{\mu} \quad (9)$$

Where ρ is the fluid density, u is the average velocity of the fluid in the channel, which is equal to the inlet velocity, l is the hydraulic diameter, μ is the dynamic viscosity.

The average Nusselt number of the micro channel is calculated by using the formula

$$Nu = \frac{hl}{k} \quad (10)$$

$$h = \frac{Q}{A_w (T_w - T_f)} \quad (11)$$

$$q = \frac{Q}{A_b} \quad (12)$$

The pumping power and friction factor are calculated by the following

$$\text{Pumping power} = V\Delta P = u A_c \Delta P \quad (13)$$

$$f = \frac{\Delta P}{2\rho u^2} \cdot \frac{l}{L} \quad (14)$$

Where V is the volumetric flow rate in a heat sink, ΔP is the pressure drop across the channel, and L is the length of the channel, $A_c$ is the cross-sectional area for a channel.

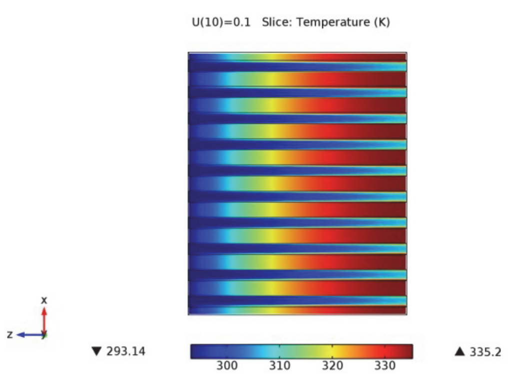



Figure 3 Micro channel showing the variation of temperature in different channel of substrate

Due to conjugate heat transfer, the velocity boundary layer & thermal boundary layer formation takes place & the heat transfer throughout the channel depends upon the layer formation.

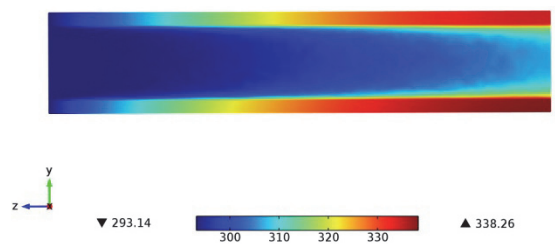



Figure 4 Thermal boundary layer formation inside the channel.

In the figure 4, the thermal boundary layer formation is shown. The layer is developing & thus the maximum heat transfer takes place throughout.

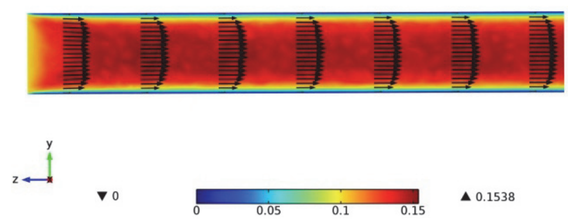



Figure 5 Velocity boundary layer formation & velocity profile inside the channel

In the figure 5, the velocity boundary layer formation is shown. The velocity boundary layer formation takes place at the beginning & thus it is fully developed.

## 4 Results and discussions

The present study has been done with a known no of channels & the performance of the different shapes of the micro channel is analyzed and compared in this section. The results are plotted against the Reynolds's Number. The variation of heat transfer coefficient is also shown against the flow rate. The performance of all shapes has been evaluated. The results are compared at the same hydraulic diameter in all shapes. Different aspect Ratios are also considered.

### 4.1 Temperature of heat sink

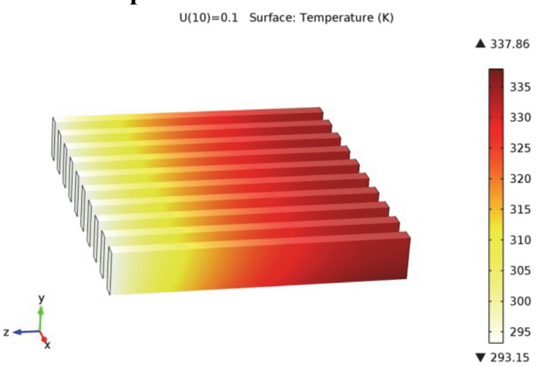



Figure 6: Temperature on the walls of the micro channel

The temperature variation for different shaped channel structure are shown with respect to the different flow rate. By considering aspect ratio of 2, the maximum temp takes place in trapezoidal section as shown in figure 7.

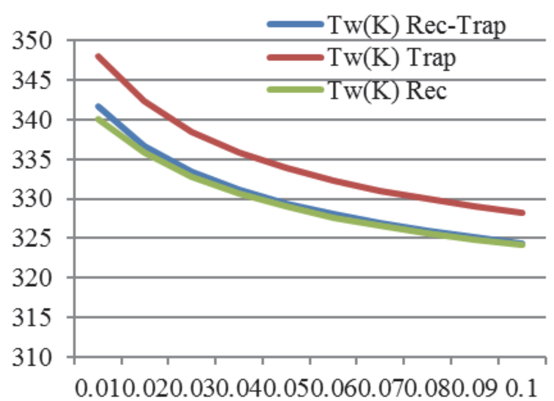



Figure 7: Max Temp vs Flow rate at AR/2

The temperature variation for aspect ratio 2.5 is shown in Figure 8.

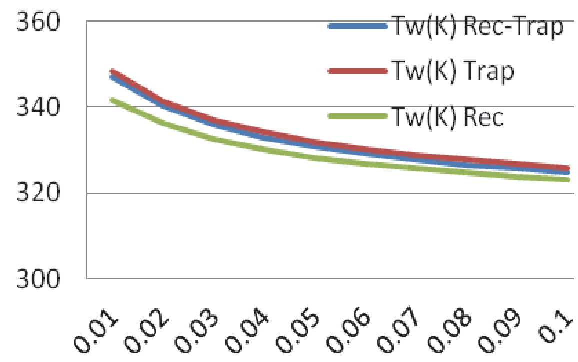



Figure 8: Max Temp vs Flow rate at AR/2.5

It was also noticed that the variation of the temperature for all other shapes & aspect ratio are almost same as shown in figure 9.

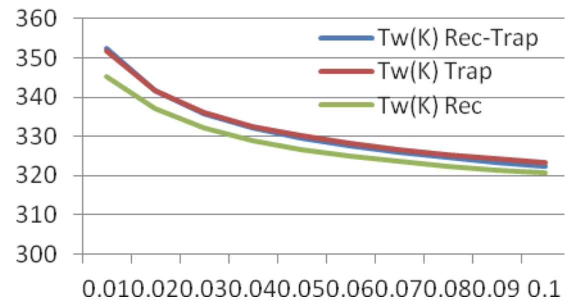



Figure 9: Max Temp vs Flow rate at AR/3

### 4.2 Heat Transfer Coefficient

The heat transfer coefficient varies a lot with the flow rate & the different shapes. Also with change in aspect ratio there is a considerable variation in heat transfer coefficient. The heat transfer coefficient is shown in figure 8.

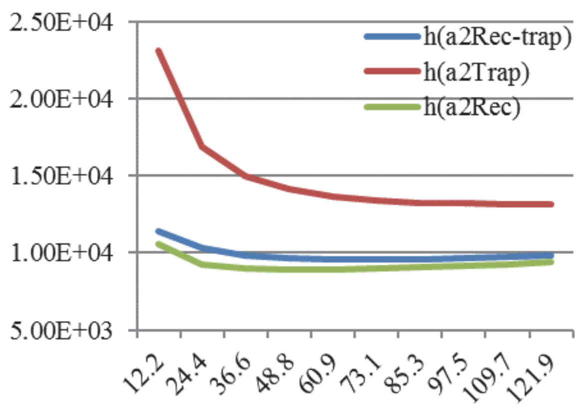



Figure 10 : Heat Transfer Coefficient vs Reynold's No. AR/2

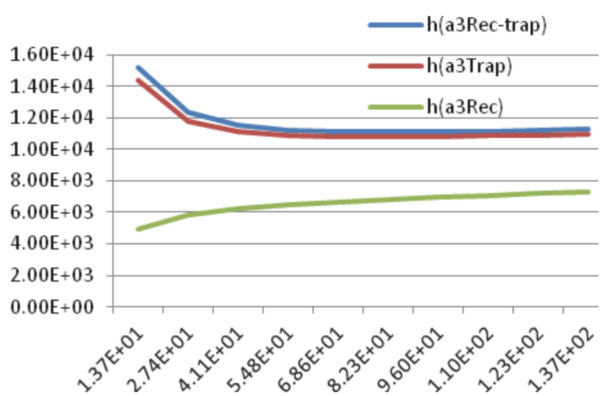



Figure 11 : Heat Transfer Coefficient vs Reynold's No. AR/3

## 5 Conclusions:

1. As the aspect ratio is increasing, the heat capacity of the fluid is also increasing but with increase in the Aspect Ratio there is a fabrication challenge. A trade off is to be maintained between the both.
2. In comparison to both rectangular & trapezoidal section, the rectangular section shows high performance.
3. The heat transfer coefficient of trapezoidal section is higher. But at the same time with increase in aspect ratio there is a lot of variation.
4. It can be deduced that for effective cooling by micro-channels, cross-section and flow rate plays an important role and has to be analyzed carefully.

**Acknowledgement:**

The authors are thankful to Director CSIO for allowing them to carry out their work. The research work is supported by Department of Science and Technology, India for providing grant (DST/TSG/FF/2011/187-C) & for supporting the research work.

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