# THEORETICAL CALCULATION AND SIMULATION OF SURFACE-MODIFIED SCALABLE SILICON HEAT SINK FOR ELECTRONICS COOLING

by

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*A surface-modified scalable heat sink that can be fabricated by applying silicon microfabrication technology has been proposed in this paper. Theoretical estimation of the heat sink thermal resistance is based on the heat sink with overall size of 1 cm × 1 cm × 1 cm, and four kinds of structure with various total number of grooves on the surface of fins have been investigated. Finite element analysis has been conducted by using COMSOL Multiphysics where fluid dynamics and heat transfer are taken into account. As a result, the lowest heat sinks thermal resistance of 6.84 °C per Watt is achieved for the structure with a larger fin area (13.1 cm$^2$) and a higher inlet air flow rate (4 m/s), suggesting an optimum fin area depending on the air flow rate.*

Keywords*: silicon heat sinks, thermal resistance, surface modification*

## Introduction

Thermal management of electronic devices and integrated circuits (IC) is significant for maintaining high performance within a limited footprint of the systems defined by manufacture technologies. Despite the increasing number of transistors per chip and the gate length scaling down to 5 nm in 2021 [1, 2], the operational speed of CPU has been capped at 4.2 GHz since early 2000's due to the limitations of thermal design power of the chip. The power dissipation of a high-performance chip is expected to rising up to 1000 [Wcm$^{-2}$] in future [3]. Thermal resistance is a key parameter to determine the cooling performance of a finned heat sink [4], and this work is focused on all-silicon efficient heat spreaders where silicon microfabrication technologies can be employed.

Since an idea of silicon micro-channel heat sink was introduced firstly by Tuckerman *et al*. [5], a lot of efforts have been made to improve the thermal resistance of the heat sink structures. As a key parameter for heat sink cooling performance, thermal resistance has been reduced mainly through increasing the surface area of heat sinks in past researches [6-8]. For example, Peles *et al*. [9] increased the surface area of a silicon heat sink across pin fin structure, and by deriving a simplified expression for the total thermal resistance calculation,

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the forced convection heat transfer performance has been investigated with obtained the Nusselt number and coolant pressure drop. Colgan *et al.* [10] reported a single-phase off chip silicon micro-channel heat sink for high power chips. A unit thermal resistance of 10.5 °C per Watt per $mm^2$ was demonstrated with the pressure drop of 5 psi, which are comparable to Tuckerman *et al.* [5] results. These works suggest that modifications of surface structure pattern can greatly affects the convection heat transfer, but the coolant is more prefer to water or other refrigerants. Danish *et al.* [11] simulated the performance of a microchannel pin fin hybrid heat sink with single-phase water flow. The computation results showed that the hybrid heat sink could maintain a 30.6% lower temperature rise at the hotspot than the non-hybrid heat sink. The heat transfer and thermal resistance of metaled micro-channel heat sinks by liquid cooling method have been widely explored in the past researches [12-16]. However, for investigation of the air cooled finned and pin-finned silicon heat sinks, to some extent, are still demanding. Silicon material is more advanced in fabrication elaboration structures, and the fast development of silicon micro and nanofabrication technology [17, 18] can provide further flexibility to determine the optimal geometry of the heat sinks' fin area.

In this paper, the impacts of various surface patterns on heat transfer of silicon heat sinks are studied by employing a finite element modelling software, COMSOL Multiphysics. Analytical formulas are proposed for thermal resistance, $R_{hs}$, calculation, and the effects of surface modification on the cooling performance of the fin-type silicon heat sink are investigated.

## Theoretical calculation

### *Basic equations*

The basic equations employed in this estimation are:

$$R_{\text{hs}} = \frac{1}{h\left(A_b + N_{\text{fin}}\eta_{\text{fin}}A_{\text{fin}}\right)} \tag{1}$$

where $h$ is the convection heat transfer coefficient, $A_b$ – the heat sink base area, $A_{\text{fin}}$ – the surface area of the heat sink fins, $N_{\text{fin}}$ – the number of fins, and $\eta_{\text{fin}}$ – the fin efficiency, which is expressed as:

$$\eta_{\text{fin}} = \frac{\tan h\left(mH_{\text{fin}}\right)}{mH_{\text{fin}}} \tag{2}$$

where $H_{\text{fin}}$ is the height of fins and $m$ is represented as:

$$m = \sqrt{\frac{2h}{k_{\text{fin}}t_{\text{fin}}}} \tag{3}$$

where $k_{\text{fin}}$ is heat conductivity of the fins, $t_{\text{fin}}$ – the thickness of the fins, and for fully developed laminar flow, the convection heat transfer coefficient can be expressed as:

$$h = \text{Nu}\frac{k_{\text{fluid}}}{b} \tag{4}$$

where $k_{\text{fluid}}$ is the heat conductivity of the fluid, $b$ – the distance between the adjacent fins, and Nusselt number is:

$$\mathrm{Nu}=\left[\frac{1}{\left(\frac{\mathrm{Re}\,\mathrm{Pr}}{2}\right)^{3}}+\frac{1}{\left(0.664\sqrt{\mathrm{Re}}\,\mathrm{Pr}^{0.33}\sqrt{1+\frac{3.65}{\sqrt{\mathrm{Re}}}}\right)^{3}}\right]^{-0.33} \tag{5}$$

where Re is Reynolds number and Pr – the Prandtl number, when heat transfer was happened with laminated air flow, it equals to 0.707.

### *Estimation model and results*

An idea of the surface-modified heat sink model is schematically shown in fig. 1(a). The calculations are focused on obtaining the heat sink thermal resistance with changing the number of grooves on the fins. Parts of the results have been discussed in [19], higher flow rate with 4 m/s have been investigated here. Four heat sinks' structure with various total number of grooves, $N_g$, on the surface of the fin have been investigated, which are 0, 24, 36, and 60 grooves, indicate case A, B, C, and D, respectively. For the sake of simplification, the average fin thickness, $t_{\mathrm{ave}}$, and average distance between fins, $b_{\mathrm{ave}}$, schematically shown in fig. 1(a) are used for calculation of the surface-modified structures. Key parameters varied are summarized in tab. 1.

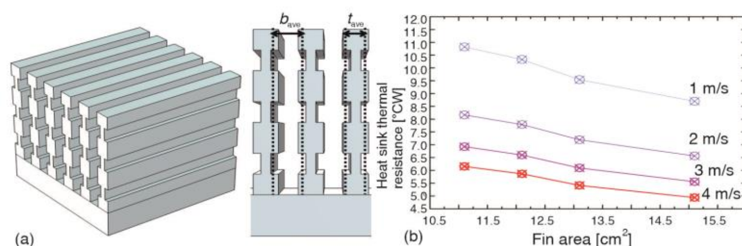


**Figure 1. Schematic diagram of the surface-modified heat sink model (a) and calculated thermal resistance in the air-flow for fin heat sinks with a different number of grooves on the fins (b)**

**Table 1. Key parameters used for calculation based on the analytical formula**

| Case | $N_g$ | $A_{\mathrm{fin}}$ [cm$^2$] | $t_{\mathrm{ave}}$ [mm] | $b_{\mathrm{ave}}$ [mm] |
|---|---|---|---|---|
| A | 0 | 11.10 | 1.00 | 0.80 |
| B | 24 | 12.10 | 0.70 | 1.10 |
| C | 36 | 13.10 | 0.78 | 1.00 |
| D | 60 | 15.10 | 0.74 | 1.06 |

The calculation result is shown in fig. 1(b) and it clearly suggests that, the more grooves, the lower thermal resistance can be achieved. With fixed heat sink structure, lower thermal resistance can be obtained with higher inlet velocity applied. In other words, increasing the surface area, $A_{\mathrm{fin}}$, indicates cooling performance is improved with active surface modification.

## Simulation module and boundary conditions

### *Simulation module*

Figure 2 shows one of the modules with 36 grooves on the fins built in the simulation software for clearly demonstration, and the dimensions applied for all of the simulation

modules are listed in tab. 2. The air is flowing horizontally through the wind tunnel from inlet to outlet, and the direction is along the groove structure.

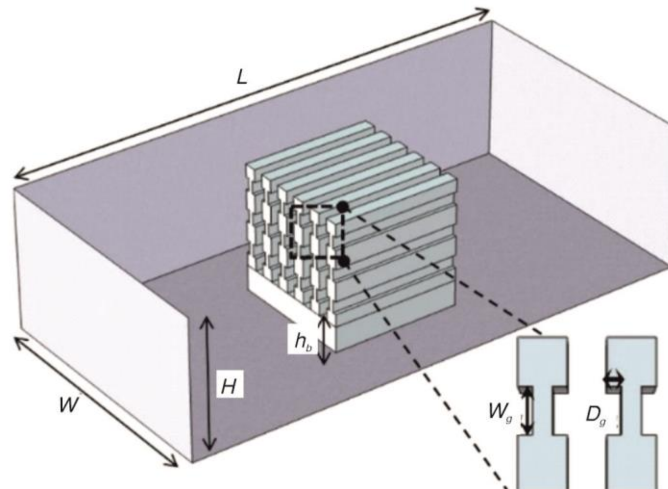


**Figure 2. Simulation module built in COMSOL Multiphysics, with the parameters of outside wind tunnel and on-fin grooves**

**Table 2. Key parameters used for simulation**

| Case | $N_g$ | $h_b$ [mm] | $W$ [cm] | $H$ [cm] | $L$ [cm] | $W_g$ [mm] | $D_g$ [mm] |
|---|---|---|---|---|---|---|---|
| A | 0 | 2.0 | 2.0 | 1.1 | 4.0 | 0 | 0 |
| B | 24 | 2.0 | 2.0 | 1.1 | 4.0 | 2.0 | 0.3 |
| C | 36 | 2.0 | 2.0 | 1.1 | 4.0 | 1.0 | 0.3 |
| D | 60 | 2.0 | 2.0 | 1.1 | 4.0 | 0.7 | 0.3 |

### *Boundary conditions*

The parameters set up to some of the boundaries are listed in tab. 3.

**Table 3. The parameters setting in the boundary conditions**

| Name | Value |
|---|---|
| Initial Values | $T = 293.15$ K = 20 °C |
| Heat Flux | $q = 10$ [Wcm$^{-2}$] |
| Wall | $u = 0$ |
| Inlet | $v_{in} = 3$ m/s |
| Outlet | $P = 0$ |

The boundary conditions used under the *Laminar Flow* module are: initial values, wall, inlet, and outlet.

The *outlet* boundary is applied to the outlet of the wind channel, which is used to make sure no fluid from the outlet can flow back to the channel.

## Simulation results and discussion

Figure 3 shows the simulation results of the 3-D temperature distribution under air-flow and the detailed 2-D distribution as a cross-section for the Case A with flat fins, fig. 3(a), Case B with 24 grooves, fig. 3(b), Case C with 36 grooves, fig. 3(c), and Case D with 60

grooves, fig. 3(d). The heater power $Q$ is 10 W, the inlet flow rate $v_{in}$ is 3 m/s, and the inlet fluid temperature $T_{in}$ is 20 °C.

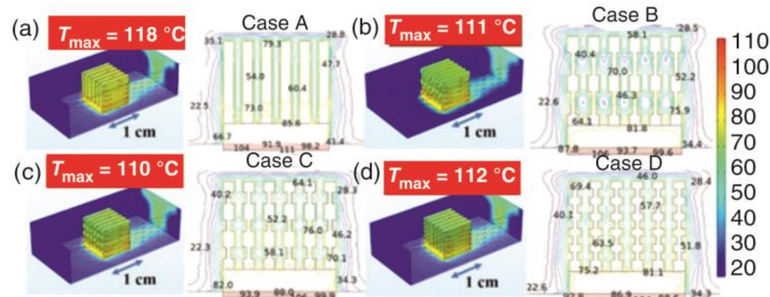


**Figure 3. Simulation results of surface temperature distribution with 3 m/s inlet air-flow velocity for the heat sink with flat fins (a), with 24 grooves (b), with 36 grooves (c), and with 60 grooves (d)**

The total thermal resistance of the heat sink, $R_{tot}$, from the simulation results is estimated as: $(T_{max}-T_{in})/Q$. Figure 4 shows how, $R_{tot}$, is changed with increasing the surface area $A_{fin}$ for various inlet flow rates. While the cooling performance improved by introducing grooved-fin structures, $R_{tot}$ decreases first and then increases under the flow rate of 1-4 m/s. Note that the lowest $R_{tot}$ is observed for the structure with the larger $A_{fin}$ under the higher flow rate in this range. The results strongly suggest that there is an optimum $A_{fin}$ depending on the flow velocity. For the flow rate of 4 m/s, minimum $R_{tot}$ of 6.84 °C per Watt is obtained at $A_{fin}$ of 13.1 cm$^2$. As the number of grooves increases, the surface area of the heat sink is increased by 1 cm$^2$ from Case A to Case D. In that case, the total convective surface area increases but it also introduces larger pressure drop at the same time. Besides, with more volume of the fin reduced, the heat conduction of the fins dominated and this contributes to the cooling performance decreased oppositely.

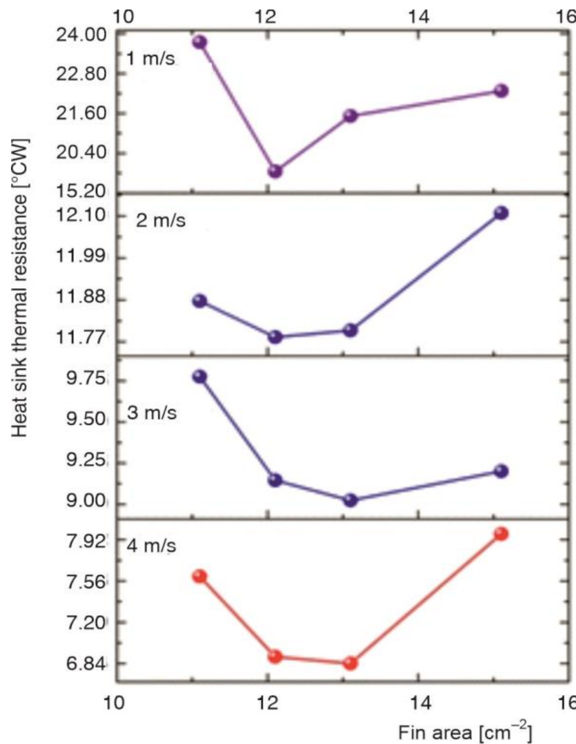


**Figure 4. Total thermal resistance extracted from simulation results as a function of the surface area for various flow conditions; note that every single point in the figure represents Case A to Case D, from left to right side**

## Conclusion

In summary, we have proposed several novel surface-modified scalable silicon heat sinks that can be fabricated by applying silicon microfabrication technology. Through both theoretical calculation and simulation estimation methods, we have confirmed that there is an optimum value of the total surface

area to minimize the total thermal resistance, which also depending on the velocity of the coolant. The results indicate that with specifying inlet velocity of the flow (2 m/s and 4 m/s in this case), the increasing of the surface area can generate a better cooling performance. However, sustainable growth of the surface area will counterproductive, the cooling performance is worse than the heat sink without any fin grooves. The discrepancy between the theoretical calculations and simulation results might depend on two reasons as follows.

- Lack of consideration of conduction heat transfer.
- The duct dimensions of the outside wind tunnel built in the simulation module are larger than the cross-sectional dimensions of the heat sink in this work, where the flow bypass phenomenon cannot be avoided.

Furthermore, substantial agreement has been gained between theoretical estimation and simulation results from the value of thermal resistances. The gradient of distinction between the theoretical calculation and the simulation is reduced as the inlet velocity increased gradually. These results contribute to identify an optimum range for the fabrication of the silicon heat sinks.

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## Nomenclature

$A_b$ – heat sink base area [m$^2$]
$A_{\text{fin}}$ – surface area of the fins [m$^2$]
$b$ – distance between adjacent fins [m]
$b_{\text{ave}}$ – average distance between fins [m]
$D_g$ – width of grooves [m]
$h$ – heat transfer coefficient [Wk$^{-1}$m$^{-2}$]
$h_b$ – base height [m]
$H$ – height [m]
$k$ – heat conductivity [W/km]
$L$ – length [m]
$N_{\text{fin}}$ – number of fins
$N_g$ – number of grooves
Nu – Nusselt number
Pr – Prandtl number
$P$ – pressure drop [Pa]
$Q$ – heater power [W]
$q$ – heat flux [Wcm$^{-2}$]
$R_{hs}$ – heat sink thermal resistance [°C/W]
Re – Reynolds number (= $UD/n$)
$t_{fin}$ – fin thickness [m]
$T$ – temperature [°C]
$u, v$ – flow velocity [ms$^{-1}$]
$W$ – width [m]
$W_g$ – height of grooves [m]

*Greek symbol*

$\eta_{\text{fin}}$ fin efficiency

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