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HEAT SINK WITH STACKED MULTI-LAYER POROUS MEDIA

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ABSTRACT

Previous studies have shown that stacked multi-layer mini-channels heat sinks with square or circular channels have advantages over traditional single layered channels in terms of both pressure drop and thermal resistance. In this work, porous media is used in the multi-layered stacked mini-channels instead of square or rectangular channels and the effect of the same on pressure drop and thermal performance is studied. Porosity scaling is done between the layers of porous media and is compared with unscaled stacked multilayer channel. Porosity scaling allows the porosity to vary from one layer to the next layer and could result in a lower pressure drop and better thermal performance.

Keywords: porous media, forced convection heat transfer, mixing effect

NOMENCLATURE

A_{pi}	Surface Area of Heated Surface (m^2)
C	Form Coefficient
C_2	Inertial Resistance Factor ($1/m$)
D_p	Pore Diameter (m)
R	Thermal Resistance (K/W)
R''	Unit Thermal Resistance ($K/W/cm^2$)
Re_p	Modified Reynolds Number
T_s	Average Heat Sink Surface Temperature (K)
$T_{w,in}$	Inlet Water Temperature (K)
V_f	Volume (m^3)
V_t	Total Volume (m^3)

Greek Symbols

α	1/viscous resistance ($1/m^2$)
ρ	Density (kg/m^3)

ϕ	Porosity
μ	Viscosity ($kg/m.s$)

1 Introduction

The need for advances in cooling of electronic equipment has become very significant due to the extensive development of semiconductor and microelectronic equipment technology. The electronics industry has moved toward higher circuit density and faster operation speed [1]. This calls for cooling systems which can dissipate high power from small surface area. This is the main characteristic of cooling of microelectronic equipment. Mini and micro channel heat sinks are being used currently for such systems. Heat sinks are usually characterized by small surface area and are made of a thin block of metal. Liquid coolant flows in small channels in the heat sink and absorbs the heat from the electronic equipment. The channels through which the coolant flows are usually very small and this gives rise to a thin thermal boundary layer [2]. This results in a large heat transfer coefficient. Mini and micro channels are widely used in electronic cooling with a liquid coolant. However, the modern semiconductor and micro-electronics technology requires very high performance heat sinks.

Metal foams which act as porous media have been widely studied [3–9] in many applications and have been used to enhance heat transfer in forced convection. Studies [10] showed that fitting a heat exchanger with porous media enhances the heat transfer rate. It was also suggested [11] that using porous media in heat sinks improved the thermal performance and heat flux of up to 6000 kW/cm^2 could be removed. Numerical investigations [12] of fluid flow and heat transfer characteristics of a pipe filled with porous media showed that the large contact surface area offered by porous media enhances the heat transfer performance.

Porous media has also been used widely in many contemporary areas such as combustion-heat exchanger system [13, 14], two-phase flow [15, 16], solar collectors [17–19], drying processes [20], geothermal energy [21] and biological systems [22, 23].

Previously, various approaches have been used to improve the performance of heat sinks. It was shown [24] that optimizing the ratio between the total surface area of the channel walls in contact with the fluid and the area of the circuit results in a very high performance heat sink with a thermal resistance lower than 0.1 (K/W) and was tested up to 790 (W/cm^2). The same study [24] also showed that the thermal resistance was independent of the power level. Relaxing restraints [25] such as laminar flow, fixed fin thickness to channel width ratio and aspect ratio could also improve the thermal performance of heat sinks. By relaxing the above mentioned restrictions, thermal resistance [25] was reduced by 0.056(K/W) when compared to those obtained in previous investigations. All these studies have used various approaches to optimize the thermal performance of heat sinks.

In this study, the effect of porous media in the thermal performance of heat sink is studied and minichannels with uniform porosity and varying porosity in the flow normal direction are compared. Water, which is the most commonly used working fluid in heat sinks, is used in this study. The application of porous media has been combined with the constructal law. Constructal law [26–28] allows the flow geometry or flow architecture to morph. A previous study [29] has reviewed the effect of scaling according to constructal law in minichannel heat sinks. The objective of this study is to investigate the effects of intentionally varying the porosity along the flow normal direction. CFD models have also been used [29] to show that scaled multi-layered mini-channels with square cross-section channel gives better thermal and hydraulic performance when compared to unscaled mini-channels.

2 Heat Sink Porosity Scaling

Porosity is defined as ratio of the volume occupied by the fluid to the total volume of the material [30].

$$\phi = \frac{V_f}{V_t} \quad (1)$$

The important parameters that are used as an input in the simulation of flow through porous media are porosity, form coefficient C , viscous resistance $1/\alpha$, and inertial resistance C_2 . These parameters can be obtained from the Brinkman-Hazen-Dupit-Darcy equation [30]. The form coefficient is calculated using the relation,

$$C = \frac{C_f}{\sqrt{K}} \quad (2)$$

TABLE 1. Porosity parameters for unscaled mini-channels

Porosity	$\frac{1}{\alpha}(1/m^2)$	$C_2(1/m)$
14%	14.54×10^5	73.5

TABLE 2. Porosity parameters for the scaled 5 layer mini-channel

Layer	Porosity	$\frac{1}{\alpha}(1/m^2)$	$C_2(1/m)$
1	14%	14.54×10^5	73.5
2	18%	18.17×10^5	114.9
3	22%	22.71×10^5	179.5
4	28%	28.3×10^5	280.5
5	35%	35.49×10^5	438.3

The constant C_f often takes the value 0.55 ($1/m$) and K is permeability whose value is fixed at 10^{-7} . The viscous resistance ($1/m^2$) is computed using the relation

$$\frac{1}{\alpha} = \frac{\phi}{K} \quad (3)$$

The inertial resistance factor ($1/m$) is computed using the formula

$$C_2 = 2 \times C \times \phi^2 \quad (4)$$

Porosity scaling allows the porosity to vary in the minichannel. Porosity scaling is done in the model with stacked five layer minichannels, where the porosity is varied only in the direction normal to the flow direction. The porosity scaling factor is given by the equation:

$$\frac{\phi_{k+1}}{\phi_k} = \gamma \quad (5)$$

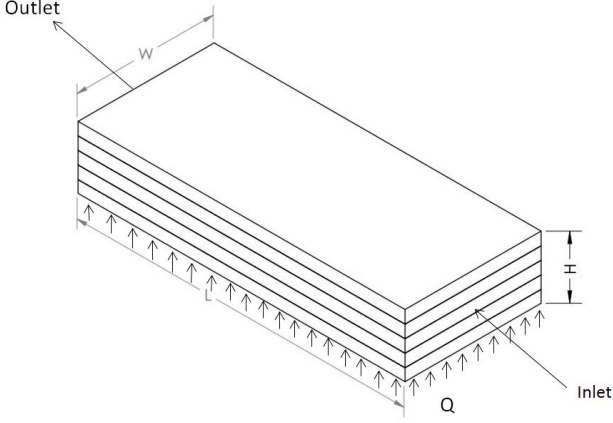
In this equation, ϕ_k is the porosity of the k^{th} layer, ϕ_{k+1} is the porosity of the $(k+1)^{th}$ layer and γ is the porosity scaling factor.

When a porosity scaling factor of 1.25 is used, the porosity of each layer increases away from the bottom wall, i.e., the lowest layer has the least porosity and porosity increases in the subsequent layers. The values of the porosity parameters, i.e., the viscous resistance, inertial resistance and porosity have been calculated and tabulated after scaling the porosity. The values are tabulated and are given in the Table 1 and Table 2.

These parameters are used as porous media input in the CFD code FLUENT. The dimensions of the mini-channels considered are given in Table 3. A schematic diagram of

TABLE 3. Dimensions of the mini-channels

No. of layers	Height(mm)
1	1.52
2	2.54
3	3.56
4	4.57
5	5.59

**FIGURE 1.** Schematic Diagram of the Stacked Five Layer Heat Sink

the five layer heat sink is given in Figure 1. All the samples considered have the same length and width of $30.5 \times 12.7 \text{ mm}^2$

3 Numerical Simulation

Reynolds number can be redefined for porous media using the pore diameter as the characteristic length.

$$Re_p = \frac{\rho D_p \mu}{\mu} \quad (6)$$

It is shown that non-Darcy flow becomes significant for Reynolds number greater than 40 [31]. In this study, the minimum Reynolds number, Re_p , is 56.32 and the maximum is 281.65. Flow through porous media is modeled in Fluent by adding an extra source term to the standard momentum equations. For a homogeneous porous medium, the extra source term is given by the following equation:

$$S_i = -(u_i \frac{\mu}{\alpha} + \frac{1}{2} C_2 \rho |u| u_i) \quad (7)$$

The mini-channel model with both stacked multi-layers and unstacked layers have been modeled using ANSYS Design Modeler. Mesh generation of all the 3D models is then

done using the pre-processor ANSYS meshing tool. The entire model is discretized using hexahedral mesh elements. The minimum orthogonal quality of the mesh is 1.0 and the maximum aspect ratio is 1.81.

The walls have a no slip boundary condition. Appropriate boundary conditions are applied to the models which are then solved. The boundary conditions are as follows:

1. Inlet: The boundary condition of the inlet of the mini-channels is set to velocity inlet. Five different values of flow rates, 100 ml/min , 200 ml/min , 300 ml/min , 400 ml/min and 500 ml/min are considered.
2. Wall: The four sides of the mini-channels are set to Wall boundary with a no-slip boundary condition. On the bottom wall, a constant heat flux of 80 W is applied.
3. Outlet: Atmospheric pressure is prescribed at the outlet boundary condition.
4. Fluid: In the fluid zone boundary condition, the computed porous media parameters, porosity, viscous resistance and inertial resistance are given as inputs. The fluid considered in this study is water.

The governing equations of the 3D, steady state flow in the mini-channels are solved using the commercial CFD code ANSYS FLUENT. Convergence criteria for residue mass, momentum and energy are set at 10^{-6} .

4 Results and Discussion

From Figure 2, it can be seen that the mini-channel with porosity scaled layers shows the best performance in terms of overall pressure drop across the channel. Mini-channel with a single layer unscaled porosity layer shows the highest pressure drop. The highest pressure drop decrease is obtained when comparing the single layer unscaled mini-channel and the five-layer scaled mini-channel. In the five-layer scaled mini-channel, pressure drop is decreased by 81% at the lowest flow rate and 78% at the highest flow rate when compared with single-layer unscaled mini-channel. When comparing the five-layer unscaled and scaled mini-channels, the pressure drop is decreased by 28% at the lowest flow rate and 24% at the highest flow rate.

Figure 3, shows the effect of varying porosity scaling factors on the pressure drop. Different porosity scaling factors are considered which increases and decreases the porosity in the flow normal direction. From Equation 5, a porosity scaling factor greater than 1, increases the porosity in the flow normal direction from the bottom wall and a porosity scaling factor less than 1, decreases the porosity in the flow normal direction from the bottom wall.

A porosity scaling factor equal to 1 implies that the porosity is uniform and constant. When the porosity scaling factor is less than 1, i.e. porosity decreases in the flow normal direction from bottom wall, the pressure drop increases considerably. Porosity scaling factor is greater than 1, the pressure drop reduces.

When the pressure contours are studied, the reason behind the lower pressure drop in scaled mini-channels is ap-

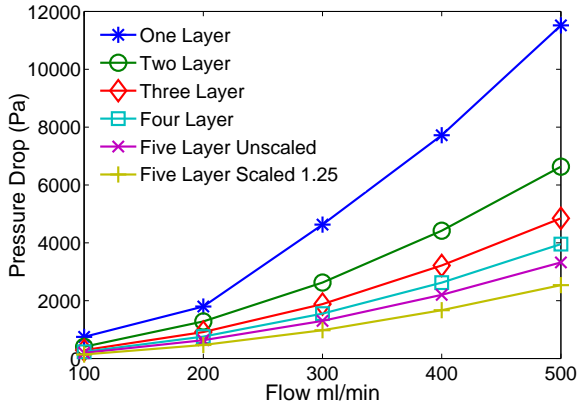


FIGURE 2. Pressure drop comparison of scaled and unscaled mini-channels

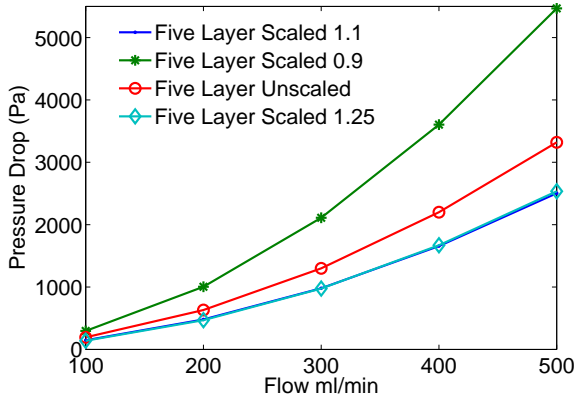


FIGURE 3. Pressure drop comparison of mini-channels with different porosity scaling factors

parent (Figures 4, 5, and 6). In the unscaled models, the porosity is constant throughout the flow domain. However, in the scaled model, the porosity increases in the flow normal direction by a factor of 1.25. Porosity is the ratio of the fluid volume to the total volume. Hence, when the porosity increases for a fixed total volume, the fluid volume increases. This creates a lower pressure drop.

The overall thermal resistance is defined by the relation,

$$R = \frac{T_s - T_{w,in}}{q} \quad (8)$$

The unit thermal resistance, R'' is defined as,

$$R'' = \frac{T_s - T_{w,in}}{q/A_{pi}} \quad (9)$$

From Figures 7 and 8, it can be seen that the thermal resistance of scaled heat sinks is lower than unscaled heat sinks. The benefit of stacking layers ceases beyond the third layer. However, when a five layer mini-channel heat sink is

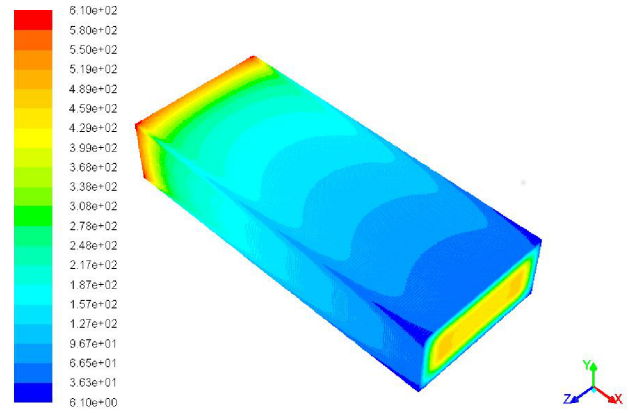


FIGURE 4. Total Pressure contour of unscaled five-layer mini-channel with flow rate of 100ml/min

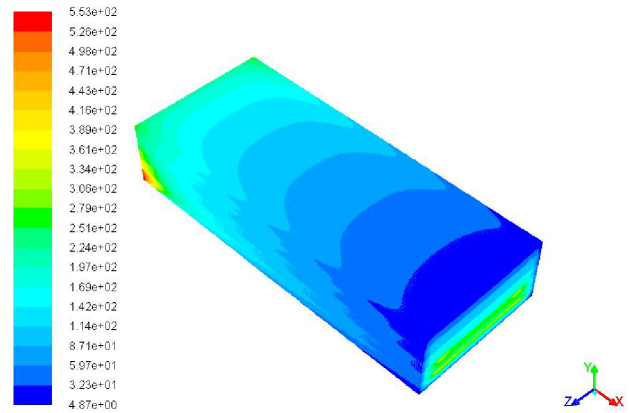


FIGURE 5. Total Pressure contour of scaled ($\gamma=1.25$) five-layer mini-channel with flow rate of 100 ml/min

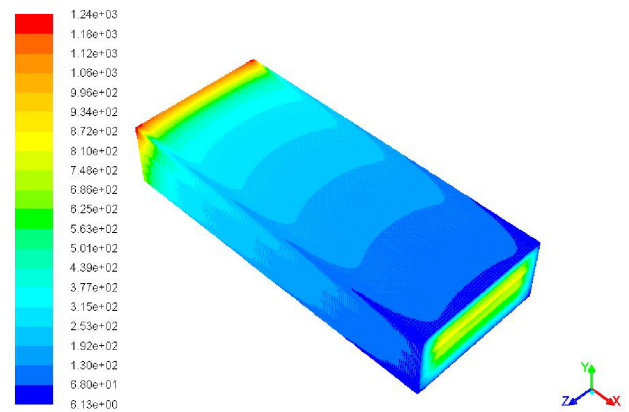


FIGURE 6. Total Pressure contour of scaled ($\gamma=0.9$) five-layer mini-channel with flow rate of 100 ml/min

subjected to a porosity scaling by a factor of 1.25, the thermal resistance is lower than any of the unscaled heat sinks. At the lowest flow rate, the thermal resistance of the scaled heat sink is 3% lower than that of the unscaled heat sink and at the highest flow rate, the thermal resistance of the scaled heat sink is 9% lower than that of the unscaled heat sink. The maximum volumetric heat transfer coefficient of

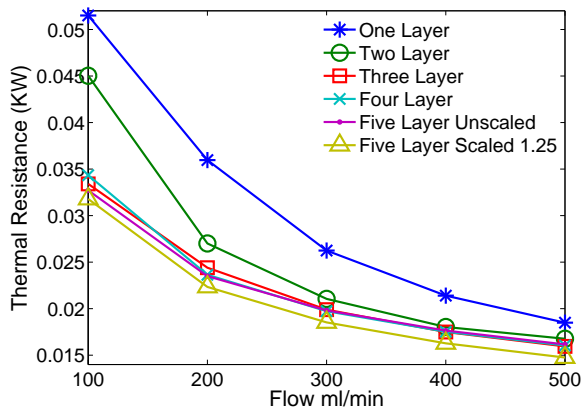


FIGURE 7. Comparison of Thermal Resistance of one to five layers with five layer heat sink with porosity scaling factor 1.25

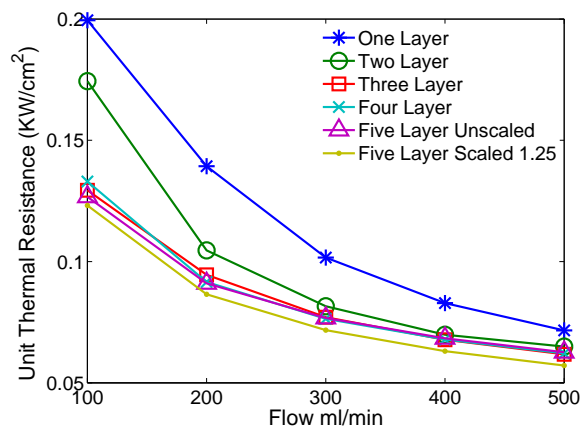


FIGURE 8. Comparison of Unit Thermal Resistance of one to five layers with five layer heat sink with porosity scaling factor 1.25

the unscaled five layer heat sink is $28.5 \text{ MW}/\text{m}^3\text{K}$ and the maximum volumetric heat transfer coefficient of the scaled heat sink is $31 \text{ MW}/\text{m}^3\text{K}$.

From Figure 9, it can be seen that volumetric heat transfer coefficient increases much more quickly with flow rate than the unscaled heat sink. This shows that scaled heat sink has better thermal dispersion, as flow rate increases, than an unscaled heat sink [6]. Considering the reduction in pressure drop also, the scaled five layer heat sink exhibits a two-fold advantage over the unscaled five layer heat sink.

When the scaled five layer heat sink is compared with a stacked five layer non-porous heat sink [29], it can be seen that the pressure drop is greater for all flow rates. However, the thermal resistance in the case of the porous media heat sink is lower.

5 Conclusion

Numerical analysis have been conducted for mini-channels with one, two, three, four and five layers and poros-

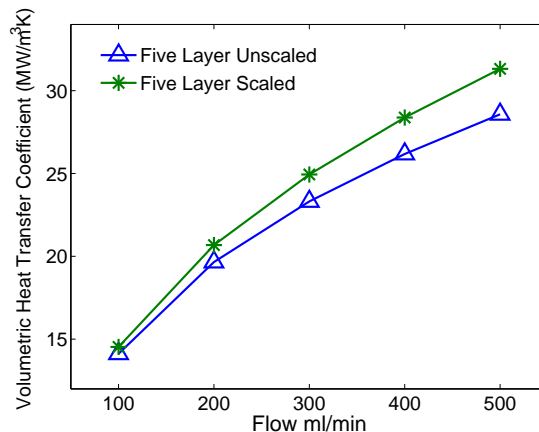


FIGURE 9. Comparison of Volumetric Heat Transfer coefficients of scaled and unscaled heat sinks

ity scaled five layer. A porosity scaling factor of 1.25 was used. Five different flow rates of water, 100, 200, 300, 400 and 500 ml/min were considered. The overall pressure drop and thermal resistance were computed for each case and studied. The following conclusions can be derived from this work: The overall pressure drop can be decreased by increasing the number of stacked layers. Pressure drop in the five layer porosity scaled heat sink is 81% and 78% lower than the single layer heat sink at the lowest and highest flow rate. When compared to the unscaled five layer heat sink, the pressure drop in the five layer porosity scaled heat sink is 28% and 24% lower at the lowest and highest flow rate. A similar trend is observed when thermal resistances are compared. Thermal resistance of the five layer porosity scaled heat sink is 3% and 9% lower than that of the unscaled heat sink at the lowest and highest flow rate. Scaled five layer heat sink has higher volumetric heat transfer coefficient compared to the unscaled five layer heat sink due to better thermal dispersion. A five layer porosity scaled ($\gamma=1.25$) heat sink has a two-fold advantage, in terms of pressure drop and thermal resistance, over the unscaled five layer heat sink. Stacking up layers decreases the thermal resistance up to the third layer. Decreasing the porosity in the flow normal direction from the bottom wall in the five-layered minichannel, increases the overall pressure drop compared to constant porosity mini-channels. In this study, the porosity was varied only in the flow normal direction. This study can be further extended to study the effect of intentionally varying the porosity along the axis of flow.

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