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Multi Layer Micro Pin-fins Heat Sinks for Better Performance

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ABSTRACT
This study numerically investigates the feasibility and advantages of using a multilayer pin-fin heat sink to increase the overall performance of the heat sink. For the purpose of determining overall performance of the pin-fin heat sink a figure of merit (FOM) term is introduced in this paper, which constituted of both the thermal resistance and the pumping power of the heat sink. Higher the FOM of a heat sink better is its overall performance. A computational fluid dynamics software CoventorWARE™ is used for the analysis of micro heat sink performance. A small portion of the entire heat sink is modeled in this study assuming repeatability towards both sides for the ease of analysis. The developed models consist of two sections, the substrate (silicon) and the fluid (water at 278K). A uniform heat flux is applied to the base of the heat sink. A single layer micro pin-fin heat sinks with same dimensions as of the multi layer heat sink was also modeled for the comparison purpose. Temperature distribution at five different locations from the inlet to the outlet section is also analyzed to study the temperature distribution over the heat sink. Circular pin-fins were used in both the multilayer and single layer micro heat sinks. Feasibility of using micro channels as the second layer was also investigated in this paper and it proved to have advantages over using pin-fin structures on both layers. A geometric optimization based on the substrate thickness of the second layer of the double layer heat sink showed that the substrate thickness of the second layer doesn't have any effect on the overall thermal resistance of the heat sink.

KEYWORDS
Micro channel heat sinks, Heat sinks, Pin fin, Stacked heat sinks, Multi layer heat sinks.

NOMENCLATURE

Cp	: specific heat (J/KgK)
H	: Height of the model (m)
k	: thermal conductivity (W/Km)
L	: length of the model (m)
PP	: pumping power (W)
ΔP	: pressure drop (kPa)
q"	: heat flux (W/m^2)
R	: Thermal resistance (K/W)
T	: Temperature (K)
W	: width of the model (m)

Subscripts

f	: fluid
s	: solid
non	: non dimensionalized quantity

INTRODUCTION
Heat sinks are devices that are used to dissipate heat from any heat generating component and thereby increasing its life time and performance. The applications of heat sinks can be classified into 1) improving the thermal control of electronic devices, assemblies, and other components by increasing the heat dissipation surface area of the device, and 2) increasing the reliability and functional performances of electronic, communication and power systems [1]. As the electronic industry is moving towards the miniaturization of electronic components, it needs more efficient methods to dissipate the heat generated from the electronic circuits. According to the Moore's law, the size of the transistors will go down exponentially with time and it has been proven right during the past few decades. With the invention of advanced power semiconductor technologies, the need for an aggressive cooling has been delayed several years. However, the heat flux from the semi conductor devices has increased significantly up to 500W/cm^2 [1]. The constrains in the space available for the heat dissipation and need for high heat transfer coefficient led to the concept of introducing multi layer heat sinks in this

decade. Multi layer microchannel heat sinks were the first to be introduced and published in this field [3-8]. A two layer micro channel heat sink have proved to increase the heat transfer efficiency of the micro heat sinks by 30% [3-8].

Macro pin-fin heat sinks were used as heat dissipation devices sincelast decade. The study and research on the micro scale pin-fin heat sinks started towards the beginning of this decade [9-11]. The pin-fin heat sinks are preferred over other micro heat sinks due its high heat transfer coefficient [9-11]. In pin-fin heat sinks the liquid flow will be continuously disturbed by the fin structures, thereby forming a continuously developing flow which increases the heat transfer efficiency of the pin-fin heat sink. Different geometries of pin-fin structures have been developed and studied over the past few years [9-11]. The thermal resistance and pressure drop across the device are equally important design constrains while designing a micro heat sink [11]. By introducing the multi layer heat sink the overall thermal resistance decreases drastically thereby reducing the pumping power required to attain liquid flow through the heat sink. In this paper the authors are trying to study the feasibility of using pin-fin heat sinks with multi layers as heat sink devices.

The benefits of using multi layer pin-fin heat sinks are studied using computer simulations generated using CoventorWARE™. The heat sink under consideration in this study has a length and width of 1 cm each and has two layers. Only a small section of the total micro heat-sink is modeled for the purpose of analysis because of the ease for analysis and due to the memory constrains of the computers. The model consists of two sections, the substrate and the fluid. The substrate used in the bottom layer had a total depth of 500 micrometers, width of 150 micrometers, and length of 1 cm. The pin fins are 150 micrometers tall and the hydraulic diameter of the pin-fin structures is kept constant at 100 micrometers. Circular pin-fin shapes are used in both layers of the heat sinks. Double layer heat sinks which contain micro channel as the second layer is also considered for investigation in this paper. A comparison between the models with pin-fin structures in the second layer and models with micro channels in the second layer is included in this paper. The effect of the substrate thickness of the second layer of the heat sink on the overall thermal resistance of the heat sink is studied using two different thicknesses of wafers, 500 μm and 350 μm. Silicon is used as the substrate, and a uniform heat flux of 666 kW/m^2 is applied at the bottom of the structure. Simulations are carried out for Reynolds numbers at inlet section ranging from 50 to 500. Water with initial temperature of 278.15K was used as the coolant in the heat-sink. Liquid flow direction in the first and the second layer of the heat sink are in opposite directions (counter flow). Simulations were conducted and the nondimensional overall thermal resistance of the heat sink and nondimensional pumping power were calculated from the results. A heat sink of same dimensions without pin-fin structures is developed for the purpose of nondimensionalizing thermal resistance and pumping power obtained from the different models. A figure of merit (FOM) was developed combining the nondimensional thermal resistance and nondimensional pumping power for all the models.

RELEVANT LITERATURES

Chong S.H. et al. tried to optimize both the single and double layer macro channel heat sink with rectangular cross section in 2002 [3]. A model based on thermal network was developed and the results obtained were compared with the results attained from a commercially available CFD package. The overall thermal resistance of the heat sink was studied and the study concluded that both the single layer and double layer heat sinks performed better in the laminar region than in the turbulent region.

Wei X., and Joshi Y., investigated the optimization of the stacked micro channel heat sink based on a simple thermal resistance network [4]. Objective was the reduction of thermal resistance and the design constrains were maximum pressure drop and maximum fluid flow rate. The variables used in the study were aspect ratio of the channel, fin thickness and ratio of fin thickness to the channel width. The study showed that the optimal number of layers for a particular pumping power was three. The study also suggested that shorter the micro channel, lower will be the thermal resistance. The same group extended their study in 2004and a numerical simulation based on the multi grid method was used to investigate the optimization of the heat sink [5]. The objective, constrains, and variables used in the study was the same as of the previous study. The result obtained from the new work was compared with the previous results. The major conclusion of the study was that the thermal resistance of the micro channel heat sink at high fluid flow rates depends on the material thermal conductivity. At low fluid flow rates the thermal conductivity of the material does not have much effect on the overall thermal resistance of the heat sink.

Jeevan K. et al. conducted an optimization study on the double layer micro channel heat sink numerically using the finite element method [6]. Genetic algorithm was used for the thermal optimization and the aspect ratio of both the channels and fins were the variables used in the optimization. The study didn't reveal anything new different from the previous works. The study concluded that by using double layer heat sink the thermal resistance of the heat sink can be reduced.

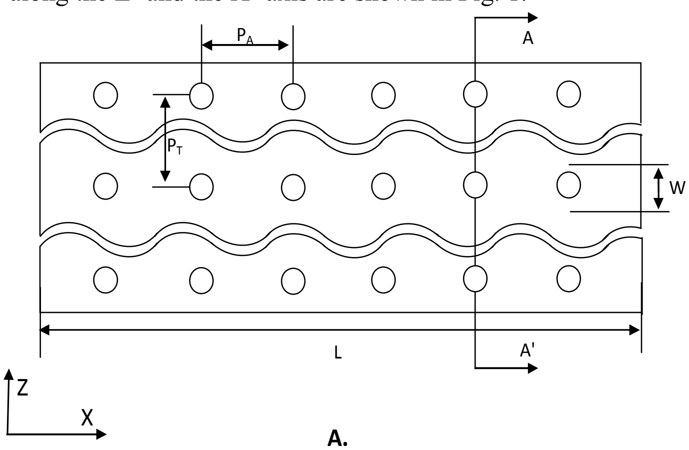
Skandakumaran P. et al. studied the performance of a multi layer micro channel heat sink both experimentally and numerically [7]. The heat sinks were manufactured in SiC material using extrusion freedom technique. The investigation showed that for a fixed fluid flow rate the pressure drop across the device decreased while using multi layer heat sinks but the thermal performance degraded. At low flow rates the experimental model showed deviation from the numerical model developed based on the thermal resistance network.

Hegde P., et al. investigated the feasibility of using two phase stacked micro channel heat sink for electronic cooling [8]. Finite element method was used for the investigation study and the study was conducted over a range of heat fluxes. The paper concluded that the thermal resistance of the double layer heat sink with two phase flow is 40% less than the single layer heat sink. Addition of a third layer will reduce the thermal resistance by another 20%.

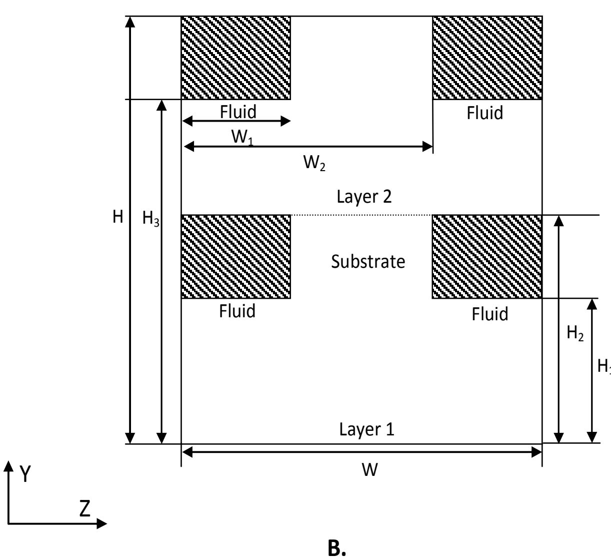
All the studies discussed above deals with the multi layer micro channel heat sink. In this paper authors are trying to investigate the feasibility of using multi layer pin-fin heat sinks for electronic cooling. The paper also investigate the feasibility of using micro channel as the second layer of the multi layer heat sink, and the performance of the both the models are compared with the single layer pin-fin heat sink with same dimensions.

THEORY

A top view of the pin-fin heat sink micro pin fin heat sink along the Z- and the X- axis are shown in Fig. 1.



A.



B.

Figure 1: Top view and cross sectional view of the double layer heat sink A) along x axis B) along z axis

The total length of the micro pin fin heat sink along the X-axis is L, the width of the heat sink is W, and the total height of the heat sink is H.

While solving the numerical equations for this model, certain assumptions were made.

1. The microfluidic device under consideration is operating under steady state.
2. The fluid does not undergo phase change while flowing through the microfluidic device.
3. Non slip boundary condition is assigned to the walls in contact with the fluid in the model.
4. The fluid is assumed to be incompressible.
5. The thermo physical properties are assumed constant.
6. Phenomenon such as viscous heat dissipation and flow maldistribution are considered negligible.

The temperature profile and the flow pattern in the repeating unit of the micro pin-fin heat sink is obtained by solving four governing equations numerically using CoventorWARE™. These governing equations consist of continuity equation, three momentum equations and two energy equations. The above mentioned governing equations in vector form can be represented as follows:

$$\nabla \cdot \vec{V} = 0 \tag{1}$$

$$\rho(\vec{V} \cdot \nabla \vec{V}) = -\nabla P + \mu \nabla^2 \vec{V} \tag{2}$$

$$\rho \vec{V} C_P \nabla \cdot T_F = k_F \nabla^2 T_F \tag{3}$$

$$k_S \nabla^2 T_S = 0 \tag{4}$$

Equation 1 is the continuity equation while Eq. 2 represents the momentum equation of the repeating unit. Equation 3 is the energy equation for the liquid and Eq. 4 represent that of the silicon substrate. Certain boundary conditions are to be defined in order to solve the governing equations of the current model and these are discussed below. The liquid velocity at the inlet section of the micro pin-fin heat sink in the X-direction (V_x) is an input parameter for this model. It is calculated from the liquid flow rate (Eq. 5) and the cross sectional area of the repeating unit by assuming uniform flow rate at the inlet section of the repeating unit. The velocities in the other two directions at the inlet section are taken to be zero. The pressure at the outlet of the device is assumed to be zero. For the actual liquid flow through the heat sink the pressure at the outlet of the heat sink need not always be zero, but here in this study as the pressure drop across the heat sink is the only point of interest the assumption is valid and is given by Eq. 6. One of the assumptions made towards the beginning of this study is that the liquid velocity at the walls of the channel is zero and is represented using Eq. 7.

$$\dot{V}_{in} = \dot{v} \tag{5}$$

$$P_{out} = 0 \tag{6}$$

$$\vec{V}_{wall} = 0 \tag{7}$$

Equations 8 and 9 represent the boundary conditions used for solving the energy equation of the substrate. In order to simulate the actual heat transfer scenario in a heat sink, a uniform heat flux is applied at the bottom of the model and is represented using Eq. 8. In case of an actual heat sink made on the silicon wafer, the top surface will be bonded to a glass plate. Thus, heat transfer (heat loss) through the top of the pin-fin structure is negligible in comparison with heat transfer through other surfaces. Accordingly, for the easiness of simulation the top of every pin-fin is considered to be insulated as represented by Eq. 9 ($\partial\Omega_T$ represents the top surface of the pin-fins in the second layer).

$$-k_S \frac{\partial T_S}{\partial y}\bigg|_{x,y=0,z} = q'' \tag{8}$$

$$\frac{\partial T_S}{\partial y}\bigg|_{\partial\Omega_T} = 0 \tag{9}$$

As the model considered in the study is only a small section of a micro pin-fin heat sink both sides of the substrate are assigned symmetry boundary conditions. This assumption is made based on the fact that the repeating unit modeled in the study can be replicated towards both sides forming the full heat sink. In addition, the inlet and outlet sections of the substrate are also considered to be insulated from heat loss to the surroundings which can either be the ambient or the bulk substrate (Eq. 10).

$$\frac{\partial T_S}{\partial x}\bigg|_{x=L,y,z} = \frac{\partial T_S}{\partial x}\bigg|_{x=0,y,z} = 0 \tag{10}$$

$$\begin{aligned} &\frac{\partial T_S}{\partial z}\bigg|_{x,0\le y\le H1,z=0} = \frac{\partial T_S}{\partial z}\bigg|_{x,H2\le y\le H3,z=0} \\ &= \frac{\partial T_S}{\partial z}\bigg|_{x,0\le y\le H1,z=W} = \frac{\partial T_S}{\partial z}\bigg|_{x,H2\le y\le H3,z=W} = 0 \end{aligned} \tag{11}$$

Equations 12 to 16 represent the boundary conditions used for solving the energy equation of the liquid. The liquid at the inlet section is kept at a constant temperature of 278.15 K and the outlet section of the liquid is considered to be adiabatic, i.e. heat transfer through the outlet section of the liquid is zero. Equations 12 and 13 represent these two conditions.

$$T_F\big|_{x=0,y,z} = T_{in} \tag{12}$$

$$\frac{\partial T_F}{\partial x}\bigg|_{x=L,y,z} = 0 \qquad 13)$$

As discussed earlier in the case of the substrate, both sides of the liquid are considered to be symmetric. Thus, the heat transfer and fluid flow through these walls is set at zero (Eq. 14 and Eq.15). The heat loss from the top of the liquid to the ambient is considered to be zero, same as in the case of pin-fin top surface explained earlier in this study. This condition is obtained using the boundary condition given by Eq. 16. The heat transfer from the substrate to the liquid through the interface between the two is defined using Eq. 17. Here $\partial\Omega$ represents the interface between the solid and liquid.

$$\begin{aligned} &\frac{\partial T_F}{\partial z}\bigg|_{x,H1\le y\le H2,z=0} = \frac{\partial T_F}{\partial z}\bigg|_{x,H3\le y\le H,z=0} \\ &= \frac{\partial T_F}{\partial z}\bigg|_{x,H1\le y\le H2,z=W} = \frac{\partial T_F}{\partial z}\bigg|_{x,H3\le y\le H,z=W} = 0 \end{aligned} \tag{14}$$

$$\begin{aligned} &\frac{\partial P}{\partial z}\bigg|_{x,H1\le y\le H2,z=0} = \frac{\partial P}{\partial z}\bigg|_{x,H3\le y\le H,z=0} \\ &= \frac{\partial P}{\partial z}\bigg|_{x,H1\le y\le H2,z=W} = \frac{\partial P}{\partial z}\bigg|_{x,H3\le y\le H,z=W} = 0 \end{aligned} \tag{15}$$

$$\frac{\partial T_F}{\partial y}\bigg|_{x,y=H,0\le z\le W1} = \frac{\partial T_F}{\partial y}\bigg|_{x,y=H,W2\le z\le W} = 0 \tag{16}$$

$$k_F \frac{\partial T_F}{\partial \vec{n}}\bigg|_{\partial\Omega} = k_S \frac{\partial T_S}{\partial \vec{n}}\bigg|_{\partial\Omega} \tag{17}$$

All the above governing equations subjected to the above boundary conditions are solved using CoventorWare™. It employs a finite volume method to solve all the governing equations using the appropriate boundary conditions. The second order upwind scheme is used for discretizing the convective terms and the central different scheme is used for the diffusive terms of these equations. In the case of the models with micro channels as the second layer the boundary condition at z=0 and W and y=H will change accordingly.

In this study, both the thermal resistance as well as the pumping power is used for achieving the optimal structure design. The thermal resistance is calculated as the ratio of the difference between the maximum temperature of the substrate and the inlet temperature of the liquid to the heat applied at the bottom of the substrate (Eq. 18). Pumping power across the channel needed for the analysis is calculated as the product of pressure drop and flow rate of the liquid (Eq. 19).

$$R_{th} = \frac{T_{S,out} - T_{F,in}}{q} \tag{18}$$

$$PP = \Delta P \times \dot{V} \tag{19}$$

The pumping power and the thermal resistance attained at each flow rate and pressure drop of different structures are nondimensionalized using the thermal resistance and pumping power obtained from a model of the same overall dimensions but without any pin-fin structures in it as shown in Eq. 20 and 21.

$$R_{th,non} = \frac{R_{th,stu}}{R_{th,without\_stu}} \tag{20}$$

$$PP_{non} = \frac{PP_{stu}}{PP_{without\_stu}} \tag{21}$$

Since the optimization of the pin-fin shape is a multi objective problem, a weighted average method is used for calculating the FOM in this analysis [2]. The FOM is calculated as follows.

$$FOM = \frac{1}{(n_1 \times R_{th,non}) \times (n_2 \times PP_{non})} \tag{22}$$

Where $n_1 + n_2 = 1$

By varying the values of n_1 and n_2 the relative contribution of each parameter in FOM can be determined according to the objective of the design. In normal situations, the value of n_1 and n_2 is taken as 0.5, thus giving equal significance to each function used for determining FOM. As the pumping power and the thermal resistance are inversely proportional to FOM; the higher the FOM, the better the overall performance of the heat sink. The maximum, minimum, and average temperature at the outlet of both the substrate and the liquid are measured along with the pressure drop and flow rate across the channel for further analysis.

MESH DEPENDENCY AND OPTIMIZATION

In order to obtain an accurate solution of the governing equations solved numerically using CoventorWare™ an appropriate meshing scheme has to be used with every model. For the study presented in this paper extruded meshing schemes is used to obtain the accurate solution of the governing equations. In the extruded meshing scheme the top X-Y plane of the model is meshed initially using the specified mesh setting and then the generated surface mesh is extruded in the Z-direction. Element size in the extruding direction is also a user defined quantity in CoventorWare™. Two different strategies are used to reach the optimized mesh setting for each of the models considered in this study. In the first strategy, the mesh sizes are varied till the maximum difference in liquid flow rate between the inlet and outlet section of the repeating unit is less than 0.1%. This strategy is used since mass flow rate has to be conserved across every face normal to flow for a heat sink. The second strategy used to obtain the optimized mesh setting is by checking the grid dependency of the meshed models. In this strategy the mesh dimensions are refined continuously and the output parameters like maximum substrate temperature, flow rate and pressure drop of the device are compared with the results obtained from the previously refined mesh model. If the maximum relative change in these parameters in comparison with the change from the previously refined mesh is within an acceptable range (less than 0.5%) the mesh is considered to be optimized. The maximum and minimum mesh size used for the Manhattan bricks meshing scheme along the X,Y, and Z direction are (25µm×12 µm ×10 µm) and (15µm×8 µm ×6 µm). The maximum and minimum element size for the extruded meshing scheme along the XY plane is 25µm and 15µm and along Z direction is 12µm and 8µm.

RESULTS AND DISCUSSIONS

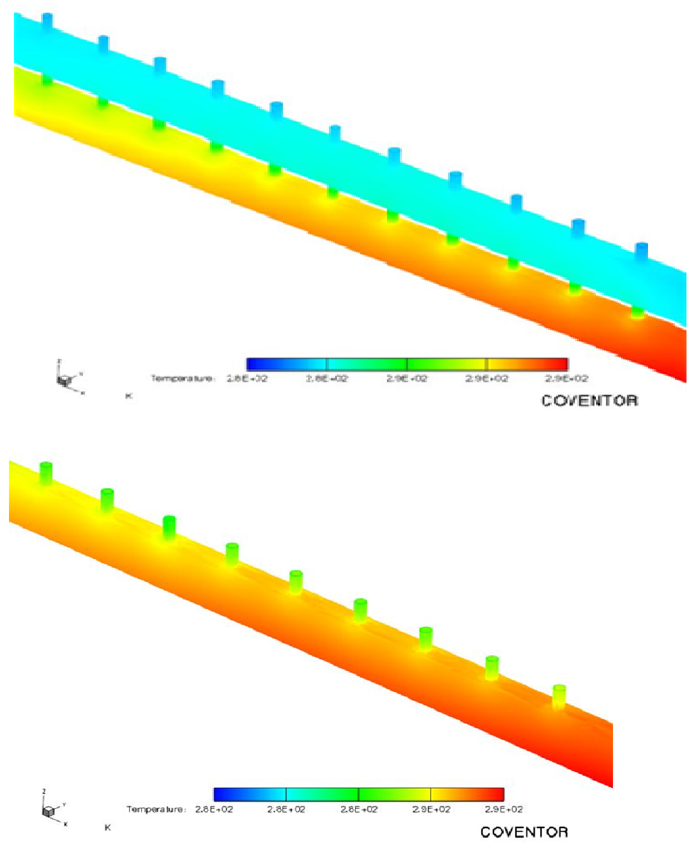


Figure 2: Contour plot showing the temperature profile in double layer and single layer heat sink.

The main objective of using multi layer pin-fin heat sink is to reduce the temperature difference between the inlet section and outlet section of the heat sink thereby reducing the overall thermal resistance of the heat sink. In this study a single layer micro pin-fin heat sink of same dimensions as of the multi layer heat sink is also developed for the comparison purpose. The contour plot of the temperature profile for both the single and the multi layer heat sink is shown in figure 2.

The temperature variation along the surface of both the single layer and the double layer is obtained and is compared as shown in figure 3. Temperature at five different locations along the length of the heat sink is acquired for this purpose. The locations at which the temperatures are collected are given in table 1. As discussed in the earlier sections, the Reynolds number of the fluid flow rate at the inlet section of the heat sink is kept constant for the comparison between different models. The temperature distribution along the single and the double layer heat sink shown in figure 3 is obtained at a Reynolds number of 250.

Location	1	2	3	4	5
Single Layer (mm)	**2.75**	**4.75**	**6.75**	**8.75**	**9.75**
Double layer (mm)	**2.75**	**4.75**	**6.75**	**8.75**	**9.75**

Table 1: Locations along the heat sink at which the temperature readings are taken (total heat sin is 1 cm long)

It can be observed in figure 3 that as the distance along the heat sink from the inlet section increases the temperature also increases. Maximum temperature occurs at the outlet section of the pin-fin heat sink. Figure 3 shows the reduction in the measured temperature at five locations when the multilayer heat sink is used. The difference in the temperature is more predominant in the inlet section of the heat sinks. For single layer heat sink the slope of the temperature rise at the inlet section of the heat sink is more compared to the slope of the temperature variation of the double layer heat sink.

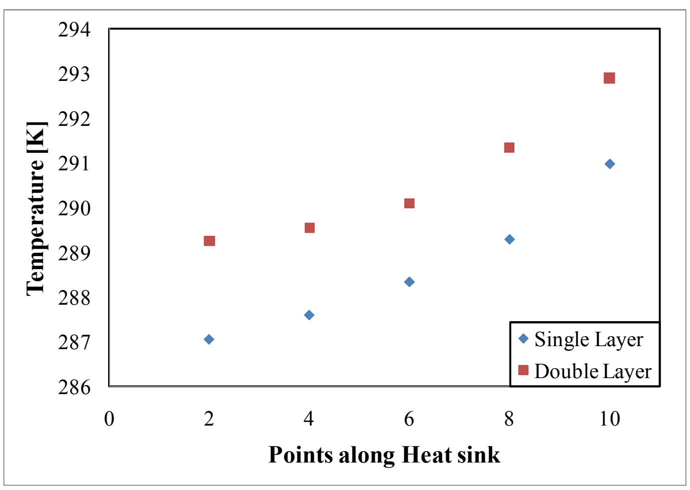


Figure 3: Temperature reading at five different locations along the length of both single and double layer heat sinks.

The feasibility of using a micro channel as the second layer of the heat sink is also investigated in this paper. The comparison of the thermal resistance of three different models is presented in figure 4.

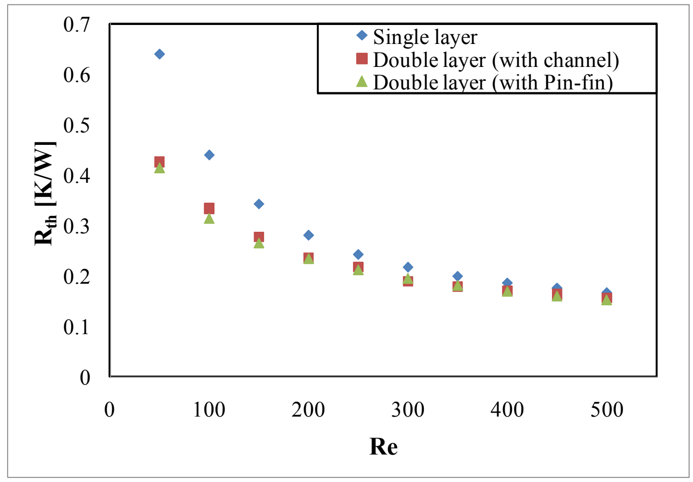


Figure 4: Comparison of thermal resistance for SLPF, DLPF, and DLMC.

The thermal resistance of a single layer micro pin-fin heat sink (SLPF) is compared to double layer heat sinks. The first double layer heat sink have micro pin-fin (DLPF) structures at both layers and the second double layer heat sink has a micro channel (DLMC) of 100 micrometer width as the second layer. The results shown in figure 4 show that the implementation of the double layer heat sink will reduce the overall thermal resistance of the micro pin-fin heat sink by a considerable amount. An interesting observation obtained from the plot is that instead of using a pin-fin heat sink as the second layer a micro channel can be used and can still get the same thermal performance.

At very low values of Reynolds numbers there is a minor shift in the value of the thermal resistance between the model with pin-fin on second layer and model with micro channel as the second layer. But as the value of Reynolds number increases the thermal resistance of both heat sinks retains very close values. Table 2 shows the percentage decrease of the thermal resistance of the pin-fin heat sink while using double layer form the results obtained from a single layer pin-fin heat sink.

Re	50	100	200	300	400	500
R_{th} SLPF	**0.641**	**0.440**	**0.281**	**0.217**	**0.186**	**0.167**
R_{th} DLPF	**0.414**	**0.314**	**0.234**	**0.194**	**0.168**	**0.152**
% Change	**76.33**	**65.55**	**45.99**	**30.19**	**18.44**	**9.01**

Table 2: Percentage change in thermal resistance between SLPF and DLPF heat sinks for various values of Reynolds numbers.

From table 2, it can be observed that at high values of Reynolds numbers the percentage reduction in the thermal resistance is less than 10%, but at low values of Reynolds numbers the reduction in the thermal resistance while using DLPF heat sink is very dominant. Since the Reynolds number of fluid flow in the inlet section of both the SLPF and DLPF are kept constant they produces same pressure drop across them. This leads to the domination of thermal resistance while calculating the FOM term.

The comparison plot of FOM between the SLPF heat sink and DLPF heat sink is given in figure 5. As the thermal resistance of the pin-fin heats sinks is the dominating factor in determining the FOM, the plot of FOM will also follow the thermal resistance plot. The FOM for the DLPF heat sink is very high at low values of Reynolds number and as Reynolds number increases the FOM for both heat sinks come closer. Since the thermal resistance of the DLMC heat sink is almost same as that of the DLPF heat sink, its FOM term will follow the same pattern of that of the DLPF heat sink, and thereby not included in figure 5.

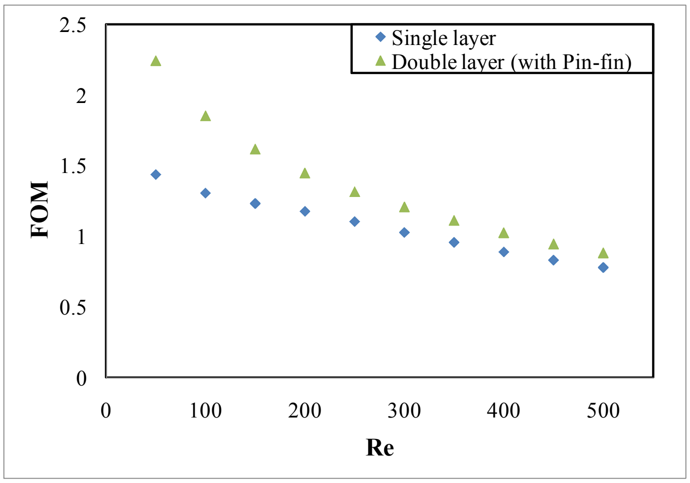


Figure 5: Comparison of FOM for SLPF and DLPF.

Even though there is not much difference in the thermal resistance between DLPF and DLMC heat sinks, the major advantage of using the micro channel as the second layer of the multi layer heat sink is the reduction in the pumping power needed to pump the fluid through the channel. Figure 6 shows the comparison plot of the pressure drop between the second layers of DLPF and DLMC.

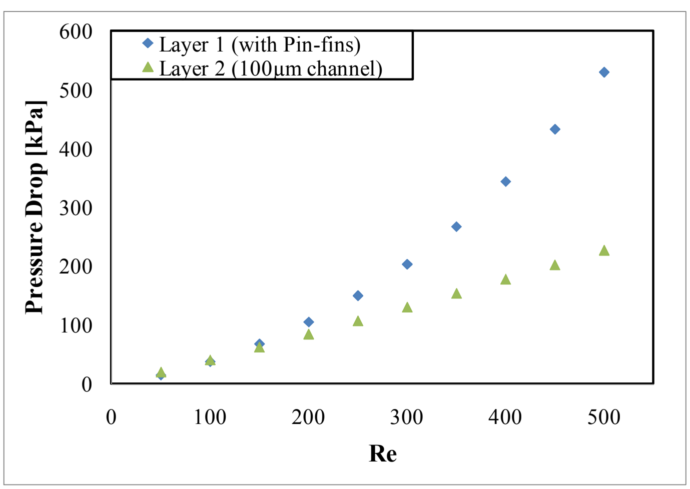


Figure 6: Comparison plot between the pressure drop in the second layer of DLPF and DLMC heat sinks.

It can be observed from the pressure drop plot that as the Reynolds number increases the pressure drop needed to attain the fluid flow across both the heat sinks increases. However, the pressure drop slop for DLPF heat sink is much higher than the DLMC heat sink. Since the lower layer of both the heat sinks has the same dimensions the pressure drop across them will remain the same. But as the second layer for both of them have different geometries, the DLMC has the advantage of having lower pumping power needed to attain the fluid flow. It can be noted from figure 6 that at very low values of Reynolds number both the heat sinks have the same pressure drop. So it can be concluded that at low values of Reynolds number the DLPF shows better overall performance and at high values of Reynolds number DLMC shows the best performance.

The effect of the substrate thickness of the second layer of both DLPF and DLMC is also investigated in this study. Substrate of two different thicknesses, 500µm and 300µm was used as the second layer for both the heat sinks. The effect of the substrate thickness on thermal resistance for both heat sinks is given in figures 7 and 8. Figure 7 gives the comparison of the thermal resistance for two different substrate thicknesses for second layer of DLMC. The plot shows that for lower values of Reynolds numbers the thermal resistance of the DLMC heat sink with smaller substrate thickness is lower than the one with thicker substrate. But at higher values of Reynolds number, the fluid flow rate though the heat sink is very high and hence this effect is nullified.. So the heat generated is swept away very fast and the second layer doesn't have much significance in determining the thermal resistance of the heat sink.

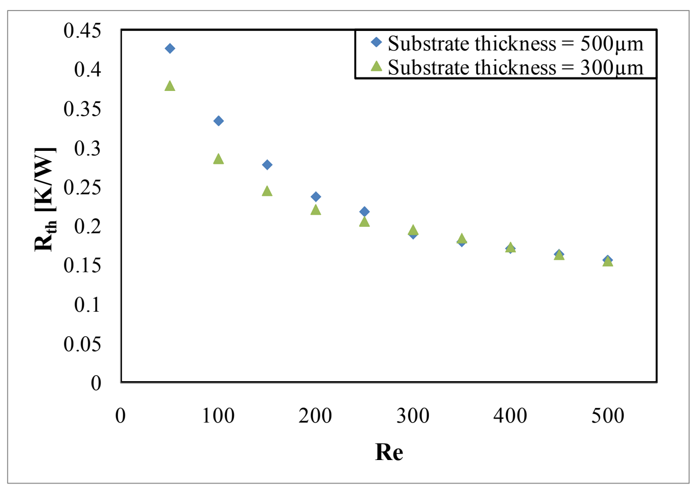


Figure 7: Comparison plot of thermal resistance for two different second layer substrate thicknesses of DLMC heat sinks.

An interesting observation that was made while investigating the effect of second layer substrate thickness for the DLPF heat sink is that, even at low vales of Reynolds number the thermal resistance is not affected much by the substrate thickness. Figure 8 shows the comparison plot of thermal resistance of DLPF with two different substrate thicknesses. There is a slight decrease in the thermal resistance value of the heat sink with thinner substrate but not predominant as the previous case. This phenomenon can be explained using the fluid flow rate through both the devices at low Reynolds number. For DLMC the liquid flow rate at low Reynolds number is very low compared to the fluid flow rate inside the DLPF at the same values of Reynolds numbers. The very low value of fluid flow rate through the second channel is responsible for variation in the thermal resistance for DLMC heat sinks with two different substrate thicknesses.

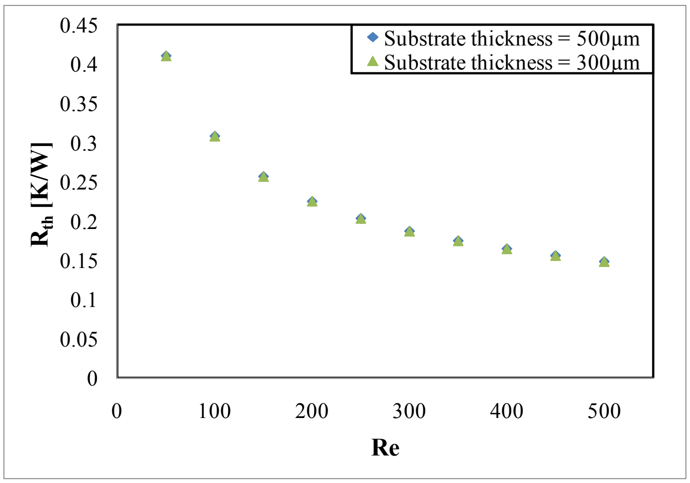


Figure 8: Comparison plot of thermal resistance for two different second layer substrate thicknesses of DLPF heat sinks.

CONCLUSION

The advantage of using double layer micro pin-fin heat sink is demonstrated in this paper with the help of computer simulations generated using CoventorWare™. All the models generated for the investigation purpose had a width and depth of 1 cm each. Silicon was used as the substrate material and water at 278.15K was used as the cooling agent. The effect of using a micro channel as the second layer of the double layer heat sink was also studied in this paper. The effect of substrate thickness of the second layer of double layer heat sink was also investigated during this study. The major conclusions of the study are given below.

1. The thermal resistance of the heat sink is reduced up to 75% while using double layer heat sink compared to single layer heat sink at low values of Reynolds number.
2. At high values of Reynolds number (500) the reduction in the thermal resistance while using the double layer heat sink was limited to less than 10%.
3. Using micro channel as the second layer of the double layer heat sink gave the same reduction in thermal resistance as while using pin-fin in the second layer.
4. Using micro channels as the second layer has the advantage of reduction in the pumping power needed to attain fluid flow though the heat sink
5. Varying the thickness of the second layer substrate of the heat sink doesn't have much effect on the thermal resistance of DLPF heat sink.
6. Varying the thickness of the second layer substrate of the heat sink showed a slight variation in thermal resistance at low values of Reynolds number for DLMC heat sink.

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