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NUMERICAL ANALYSIS OF TRAPEZOIDAL SHAPE DOUBLE LAYER MICROCHANNEL HEAT SINK

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Abstract- With increasing demand for higher computational speed and emerging micro-systems, thermal management poses serious challenge for efficient cooling. Among these liquid cooling using microchannels has gained significant attention and has been extended to its double layer configuration which eliminates the drawback of significant temperature variations in single layer system. The double layer configuration has been primarily analyzed for rectangular ducts. In this study the performance of trapezoidal shape double layer microchannel heat sink is investigated and compared to rectangular double layer heat sink of same flow area. Four different possible configurations are analyzed and comparative study among respective counter and parallel configuration is performed followed by comparison among each configuration. The performance is evaluated on the basis of maximum temperature attained at the heated surface as well as minimum temperature variations. Finally the best performing configuration is compared with double layer rectangular heat sink. Analysis shows that among various trapezoidal configurations, the one with larger side face to face is most suitable. Further comparative study with rectangular system shows that performance of trapezoidal double layer heat sink is superior in both aspects, i.e. minimum thermal resistance as well as minimum temperature variations.

Keywords*- Double Layer; Heat Sink; Microchannel; Rectangular duct; Trapezoidal duct.*

I. INTRODUCTION

The recent developments in field of microelectronics and ever increasing demand for higher computational speed has escalated power density levels in modern electronic systems. With market trends pushing for miniaturization, International Technology Roadmap for Semiconductors (ITRS) has predicted that peak power consumption in high performance desktop applications will rise to 198 W by 2015[1] and expected dissipation of heat flux in next generation microprocessors and microelectronic components is over 1000W/cm^2 [2].Consequently, there is an immediate need for efficient cooling system to cope with rise in temperature levels by virtue of dissipation of large amount of heat dissipation within a small space. With operating parameters of several active and passive components being temperature dependent, this further necessitates efficient thermal management for consistent operation and reliability of the circuits. The conventional cooling systems such as air cooling, heat pipes, thermoelectric cooling etc. either seem to reach their practical limit or are incompatible with new microelectronic components. To overcome this, researchers have proposed several solutions which have been summarized in [3], according to their heat removal capacity. Of all the proposed solutions, for power dissipation levels of the order 1 MW/m^2 or more, cooling techniques using liquids hold a promising future which has been a propelling factor for significant work over the past decade pertaining to liquid cooling using microchannel. This is attributed to several advantages such as their direct integration on the substrate (electronic chip) which can reduce thermal contact (internal) resistance almost to zero. Moreover, reduced hydraulic diameters allow for significantly high values of heat transfer coefficients, of the order 10^3 W/m^2 along with providing increased heat transfer area to volume ratio.

After the introduction of concept of microchannel cooling by Tuckerman and Pease [4] (who demonstrated that very small thermal resistance (9x10^{-6} K/(W/m^2)) is possible with power density of 790 W/cm^{2}with microchannels 50 µm wide and 300 µm deep) significant numerical as well as experimental contributions have come up in this field as well as review articles pertaining to geometrical configurations, validation of macro scale relations at micro-scale etc. [5-9]**.** Despite several advantages mentioned above, large temperature gradients in flow direction in small region are one of the major drawbacks in microchannels. This is mainly due to difference in heat expansion coefficients which may result in residual stresses and hence affect reliability as well as operating characteristics. Vafai and Zhu [10] proposed the concept of double layer heat sink and showed that for small temperature variations in the chip, pressure drop required in case of such design is considerably smaller than single layered structure. It was further observed that the stream-wise temperature rise on the base surface substantially reduced compared to that of single layered heat sink. Following this, significant research has been carried out to further explore the advantages of such kind of system. Recently, Hung et al. [11] numerically analysed heat transfer characteristics double-layered microchannel heat sink in which they investigated effects of substrate materials, coolants, and geometric parameters such as channel number, channel width ratio, channel aspect ratio, substrate thickness, and pumping power on the temperature distribution, pressure drop, and thermal resistance. It was observed that with substrate materials having a higher thermal conductivity ratio, coolant with high thermal conductivity and low dynamic viscosity significantly enhanced the performance of double layer system. The results also showed superior performance of ouble-layered microchannel heat sink over single ayered by an average of 6.3%. It is always desired to have optimal operating parameters in any system. In lieu of this, optimization of microchannel (single and double layered system) has also been studied by various authors [12, 13]. Investigations of stacked microchannels, in which several layers of heat sinks are placed one over another, can also be found in literature along with their optimization studies [14, 15].

It is interesting to note that all the studies carried so far have considered rectangular cross-section duct in upper and lower channel. However, for same cross-sectional area, hydraulic diameter of a trapezoidal duct is smaller as compared to that of rectangular duct. Since the performance of microchannels is largely governed by hydraulic diameters (both in terms of heat transfer coefficient as well a pump power), use of trapezoidal shape double layer microchannel is proposed in this study and its performance is analysed in various aspects. Three dimensional conjugate heat transfer study is carried out to analyse various configurations of such design and the results are compared with that of rectangular duct of same cross sectional area and equal volume of substrate material. The performance in each case is evaluated on the basis of minimum thermal resistance (or maximum base temperature attained) as well as minimum thermal variations.3

II. ANALYSIS

In this section, various aspects of analysis like computational domain, governing equations, boundary conditions etc. are explained briefly. Various dimensions used in the study are also resented.

A. Computational domain

Fig.1 shows various possible configurations and their respective computational domains of trapezoidal shape double layer microchannels analysed in this study. These have been labelled as V1, V2, V3 and V4. Rectangular configuration having same cross sectional area as that of trapezoidal duct is also shown in Fig. 1. The dimensions of solid region are kept such that total volume of the substrate remains the same which is necessary for appropriate comparison. $W_{S,TR}$ and $W_{S,R}$ represent thickness of substrate in case of trapezoidal and rectangular duct respectively.

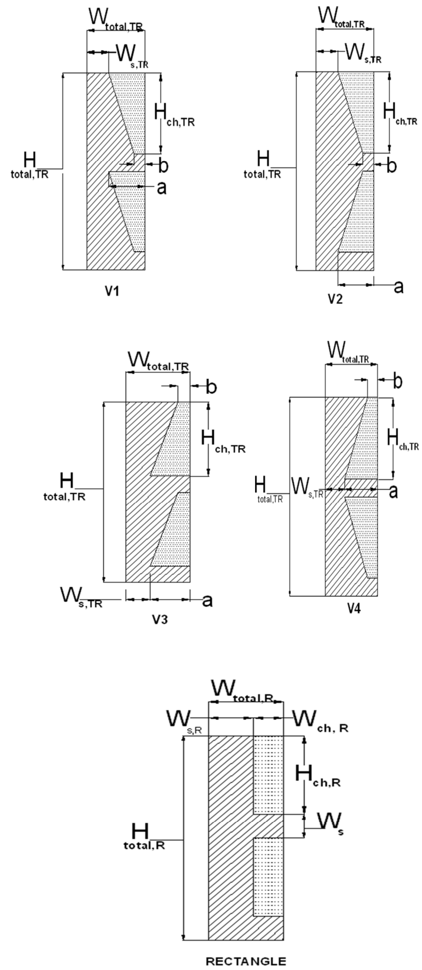

Figure. 1 Geometric Configurations

Similarly, $W_{total,R}$, $W_{total,TR}$, $H_{total,R}$ and $H_{total,TR}$ denote total width and height of computational domain in case of rectangular and trapezoidal configuration. The length of microchannel is 8000 μm while other dimensions used have been selected based on dimensions available in literature [9, 16]. Various dimensions used in the analysis are described TABLE I.

TABLE I VARIOUS DIMENSIONS USED IN ANALYSIS (MICRONS)

$W_{s,TR}$	$W_{s,R}$	W_s	$W_{total,TR}$	$W_{total,R}$
30	47.5	30	80	80
$H_{ch,R}$	$H_{ch,TR}$	$H_{total,R}$	$H_{total,TR}$	a
100	100	260	260	50
b	$D_{h,TR}$	$D_{h,R}$	$W_{ch,R}$	
15	76	78.8	32.5	

The analysis is based on the following assumptions:

- Steady state flow.
- Incompressible fluid.
- Laminar flow.
- Constant properties of both fluids and solid.
- Effects of viscous dissipation are negligible.

B. *Governing Equations*

Based on the above assumptions the governing equations of mass, momentum and energy as applied to the fluid region were:

Continuity: $\nabla.\vec{V_f} = 0$ (1)

Momentum: $\rho_f \vec{V_f}.\nabla\vec{V_f} = -\nabla P_f + \mu_f \nabla^2 \vec{V_f}$ (2)

Energy: $\vec{V_f}.\nabla T_f = \alpha_f \nabla^2 T_f$ (3)

where the variables $\vec{V}$, μ, ρ and α represent fluid velocity, viscosity , density and thermal diffusivity respectively. *'P'* and *'T'* denote pressure and temperature while the subscript *'f'* denotes fluid. The following energy equation was applied to solid region.

Energy (for heat transfer):

$$\nabla^2 T_s = 0 \quad (4)$$

'T_s' represents the temperature of solid region with subscript *'s'* representing solid region. Hydraulic diameter was calculated as: $D_h = 4A/S$ (5)

Here *A* and *S* are the area and perimeter of the single channel respectively. The hydraulic diameters are also tabulated in Table 1 and these correspond to upper/lower channel.

C. *Boundary Conditions*

Since the governing equations being solved are partial differential equations their solution is largely affected by boundary conditions. The following boundary conditions are applied to the computational domain in the present study. The adiabatic conditions were applied at the following faces:

- Top surface as the heat sink cover is usually made of poorly conducting material.
- The entrance and exit walls of the solid region considering heat transfer due to fluid as dominant factor.
- Outer wall of solid region owing to symmetry condition.

Uniform heat flux, *q''* (=10^6 W/m^2) is applied at the base, at *y* = 0 µm while uniform velocity and temperature conditions (300 K) were imposed at the inlet of both the systems. For counter flow, plane *z=0* represents inlet for lower channel and outflow for upper channel while for parallel flow it denotes inlet for both the channels. Continuity of temperature and heat flux as well as no slip condition was assumed at solid-liquid interface while symmetry conditions were imposed on the plane *x*=0 in all the cases. The solid region was assumed to be made of silicon while water is used as cooling medium.

D. *Solution method and grid independence*

The continuity, momentum, and energy equations were solved using general purpose finite volume based code, FLUENT. The standard scheme for pressure discretization, SIMPLE algorithm for pressure velocity coupling and the first order upwind scheme for momentum and energy equations were used. For grid independence, three grid sizes were tested separately for each geometrical configuration. In lieu of computational resources and time, further refinement of grid was stopped when variation in results upon further decrease in grid size was below 1%.

III. RESULTS AND DISCUSSION

The performance of various configurations of trapezoidal shape double layer heat sink is evaluated. The results are compared for parallel and counter arrangement of each followed by comparison among each configuration. The range of flow rate considered for comparison varies from 7.8×10^{-9} m^3/s to 9.74×10^{-8} m^3/s which corresponds to Reynolds number 45 to 567. The flow rate mentioned here corresponds to total flow rate (i.e. sum of flow rate in upper and lower channel). For sake of simplicity, abbreviations such as 'PF' for parallel flow, 'CF' for counter flow, 'UC' for upper channel 'LC' for lower channel have been used extensively.

E. *Comparison among respective parallel and counter arrangement*

In this section, comparative study is performed between PF and CF of various configurations and the performance is evaluated on the basis of maximum temperature as well as minimum temperature variations attained at heated surface. Whereas former is one of the basic criteria and desired to be as low as possible, the latter is necessary as it forms the basic reason for double layer arrangement.

- *Maximum temperature versus flow rate*

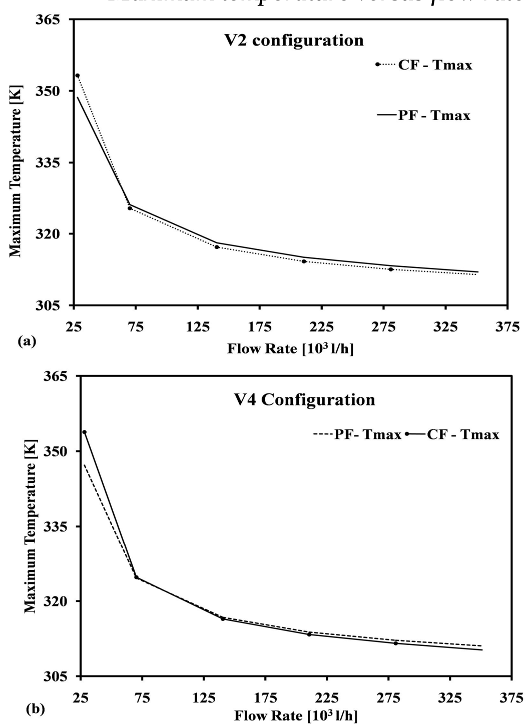

Figure 2. . Maximum Temperature reached at the base for CF and PF for (a) V2 (b) V4

Fig. 2 (a-b) shows maximum temperature attained for V2 and V4 configurations respectively. It can be deduced that CF shows superior performance in both the configurations though for V4 configuration the difference is less as compared to other. It is also observed that at low flow rate (or *Re* number) the performance of PF is superior. This is attributed to low thermal resistance in PF at low Reynolds number whereas at higher flow rates thermal resistance is lower in CF due to efficient heat redistribution. This prevents significant rise in temperature of water in upper and lower channel towards their respective exits and hence superior performance of CF. Hence the results in case of trapezoidal configuration are similar to those of rectangular double layer heat sink at low as well as high *Re* number range as explained in the studies of Levac et al.[17]. Similar results were obtained in other two configurations and hence have been omitted for sake of brevity.

- *Temperature Variations versus flow rate*

Table II shows ratio of temperature variation, ΔT (difference between T_{max} and T_{min} at the heated surface), of PF and CF versus flow rate for all the configurations. It is defined as: $\Delta T_{PF,CF} = \frac{\Delta T_{PF}}{\Delta T_{CF}}$ (6)

It can be observed that in all the cases $\Delta T_{PF,CF}$ is greater than 1 which shows that temperature variation in PF is greater than in CF thereby favouring counter arrangement in all the cases.

F. Comparison among various configurations

In this section the performance of various trapezoidal configurations are compared based on similar criteria as in previous section. The main motive behind this is to find the best suitable configuration in case of trapezoidal double layer configuration. Since the performance of counter arrangement of each configuration is found to be superior, the results pertaining to same are compared. It is worth mentioning that in this comparison, major consideration is given to minimum temperature variation as this prevents thermal stresses to accumulate over a period which may otherwise lead to failure of the component. This aspect is also important for higher reliability of the system.

TABLE II. RATIO OF ΔT PF AND CF FOR ALL TRAPEZOIDAL CONFIGURATIONS

Flow rate x10^3 (l/hr)	V1 ($\Delta T_{PF,CF}$) [K]	V2 ($\Delta T_{PF,CF}$) [K]	V3 ($\Delta T_{PF,CF}$) [K]	V4 ($\Delta T_{PF,CF}$) [K]
28.8	1.383	1.349	1.202	1.420
70.2	1.517	1.478	1.333	1.538
140.4	1.476	1.447	1.385	1.492
210.6	1.439	1.418	1.401	1.479
280.8	1.413	1.392	1.423	1.477
351	1.394	1.352	1.460	1.506

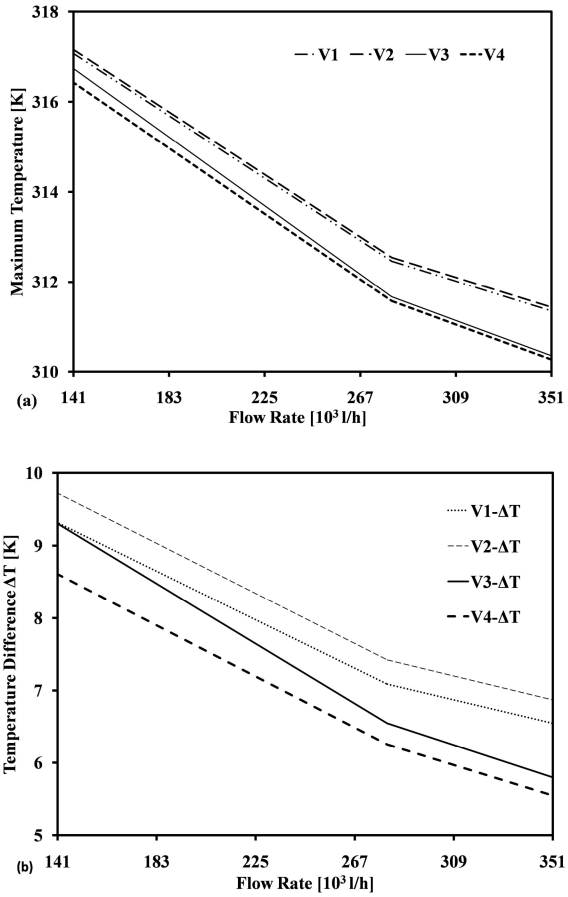

Figure 3. Comparison among all the 4 configurations on the basis of (a) maximum temperature (b) minimum temperature variations

As can be seen from Fig. 3(a), maximum temperature attained is lower in case of V3 and V4 configuration in comparison to other two. In addition, it is also observed from Fig. 3(b) that in terms of minimum temperature variations both these configuration perform considerably better. This may be explained due to the fact that in both these configurations, larger side of trapezium in upper channel faces downwards. This results in better heat carrying capacity as compared to other configurations. Further it is to be noted that V4 configuration is best suited in all aspects among all the configurations analysed. In this configuration longer edges of trapezium are face to face. The superior performance may be by virtue of minimum thermal resistance offered to heat flow from lower to upper channel. This can be justified by minimum substrate thickness near their interface as well as maximum area of fluid available for heat transfer.

G. Comparison between Rectangular and Trapezoidal configurations

Till now the best suitable configuration for trapezoidal shape double layer heat sink is found. However, since in literature only rectangular configurations have been analysed it becomes necessary to compare their performance so as to justify such a configuration. This issue has been dealt with in this section. Since cross-sectional area of both is same, the performance is compared on the basis of pump power. This is further more practical approach of comparison as in any system, pump power governs the net available flow rate and hence performance of microchannels. Owing to superior performance of CF rectangular configuration using water, as can be found in several studies, only this configuration is analysed. Figure 4(a) shows thermal resistance of rectangular and trapezium configuration while temperature variations are shown in Fig. 4(b). We define thermal resistance as

$$R_{th} = \frac{T_{max} - T_{inlet}}{q''} \quad (7)$$

where T_{max} is maximum temperature at heated surface, T_{inlet} is the inlet temperature of coolant and q'' is heat applied at heated surface. The results as observed are quite interesting and show that at larger values of pump power the performance of trapezoidal shape heat sink is superior both in terms of minimum thermal resistance and minimum temperature variations. This may be attributed to lower hydraulic diameter in case of trapezium. It is to be noted that even though lower hydraulic results in larger pressure drop and hence pump power, but superior performance in this case may be explained on the basis of larger area of developing thermal boundary layer with minimum conduction resistance between them. This region is the base of upper channel and top of lower channel. Further larger flow area means more mass or coolant to carry heat thereby giving superior performance.

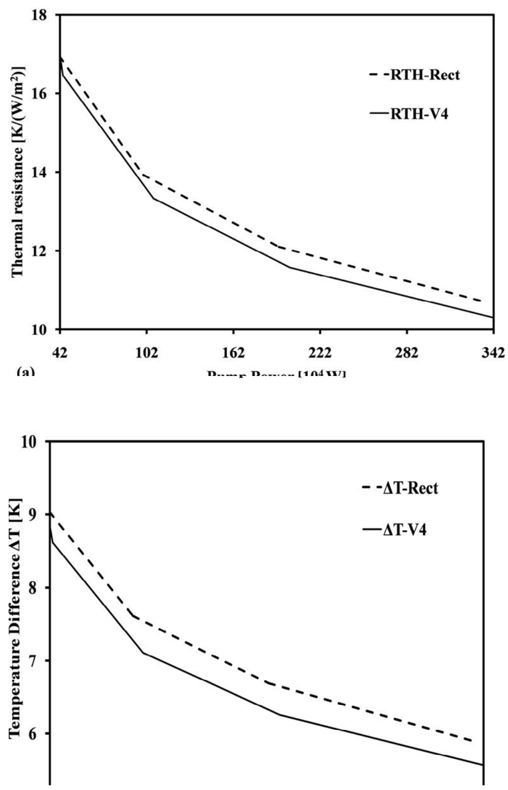

Figure 4. (a) Thermal Resistance versus flow rate (b) minimal temperature variation versus flow rate for V4 (DLCF) and DLCF with rectangular duct

IV. CONCLUSION

Three dimensional conjugate heat transfer analysis is carried out numerically to compare the performance various configurations of trapezoidal shape double layer heat sink. It is observed that like rectangular ducts, performance of rectangular duct is superior in terms of minimum temperature variations. Of various configurations analyzed the one with larger side of trapezium face to face shows best performance amongst all. This configurations is compared with rectangular double layer heat sink and it is observed that for the range of flow rate and dimension analyzed the performance of trapezoidal double layer is superior in all aspects i.e. minimum thermal resistance as well as minimum temperature variations. The study can hence be extended to optimize shape of trapezoidal ducts for optimal performance.

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