NUMERICAL ANALYSIS OF THERMO-HYDRAULIC PERFORMANCE IN PIN ENHANCED MICROCHANNEL HEAT SINK

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Abstract: Enhancement of thermal performance of microchannel heat sink (MCHS) by introducing pins in the channel is the main objective of this study. A single microchannel with and without pins is studied for different flow and thermal conditions. The pins are introduced from the top cover instead of the conventional method of pin-fin protruding from the base of heat sink as it drastically reduces the fabrication cost. Pins of different heights are numerically analyzed. Of the different configurations, enhanced MCHS with pin height of 1.2 mm give the best results. This is due to the fact that as clearance between base and pins increase, there is an accelerated flow in the clearance region producing more mixing of fluid and higher heat transfer performance. Effects of fluid bypassing the pins become dominant when the clearance region is further increased, resulting in lower heat transfer performance. The introduction of pin increases the pressure drop; it is highest for full height pin and decreases with pin height. The best configuration is then studied for different flow rates. The thermal performance increases with increase in flow rate, the temperature gradient decreases by 33% for Reynolds number of 618 but at an increased pressure drop of 100% than the conventional MCHS. A correlation showing the effect of pin height on Nusselt number is proposed.

***IndexTerms* - microchannel, forced convection, temperature gradient**

I. INTRODUCTION

Recent developments in micro-electro mechanical systems and ultra large scale integrated technologies are becoming increasingly dependent upon the ability to dissipate huge amounts of heat from very small areas. This has motivated researchers to focus on the improvement of thermal and hydrodynamic performance of microchannels. After the pioneering work by Tuckerman and Pease [1], a number of researchers have made attempts to improve the performance of conventional parallel MCHS. Most of the researchers who studied pin-fin microchannels studied an array of pin-fins creating a microchannel flow area. Carlos et al. [2] developed micro pin-fin with variable fin density to generate more uniform temperature at the IC chip interface. The novel design generated thermal resistance ranging from 0.14 K/W to 0.25 K/W with a pressure drop lower than 90 kPa. Carlos et al. [3] compared inline and offset micro pin-fin heat sinks with variable fin density and deduced that the offset pin-fin configuration is capable of achieving a much lower thermal resistance of 0.1 K/W. Ali Kosar and Peles[4] experimentally studied the thermal and hydraulic performance of shrouded staggered micro pin fins and compared using the correlations developed by earlier researchers. Abel et al. [5] experimentally studied pressure drop and heat transfer in a single phase micro pin fin heat sink. The measured pressure drop and temperature distribution were used to evaluate average friction factor and local averaged heat transfer coefficient and Nusselt number. They examined the previous friction factor correlations and only one was found to agree with the experimental data. Khan et al. [6] optimised the inverse trapezoidal cross section MCHS by using multi-objective algorithm. The design points were selected using Latin hypercube sampling, surrogate models were created at each design points and Pareto sensitivity specified. Effect of width, depth and angle of channel on thermal resistance and pressure drop was studied and they concluded that the width parameter was the most sensitive one.

Kuppusamy et al. [7]placed triangular shaped micro-mixers in the walls of microchannels and optimised the design by considering two different designs and two different flow directions. They obtained optimum flow rate, flow direction and micro mixer design that give thermal enhancement without additional pressure drop. They conclude that the micro-mixer design increases the thermal performance along with reduced pressure drop.

Gaikwad et al. [8] designed the fluid flow path based on pinnate type leaf venation for development of secondary flows in the microchannels. The numerical analysis is carried out of the enhanced MCHS subjected to heat flux from 65 to 200 W/cm^{2}and cooled by water flowing at different flow rates. The performance is compared with conventional MCHS. The overall enhancement in the performance of new design is 1.4 to 1.85.

Husain and Kim [9] performed shape optimisation of MCHS of trapezoidal cross section using response surface approximation. Optimisation of parameters relating microchannel width, depth and fin width for minimum thermal resistance was done by choosing the design points using fractional factorial sampling method and validating with analytical and experimental results. The performance was compared with reference work and found to have 12 % less thermal resistance.

Abdoli and Dulikravich[10] optimised micro heat exchangers having two and three floor configurations and having alternating flow directions by coupling two solvers viz. Quasi-1d thermo-fluid analysis and a fully 3-d steady heat conduction analysis. Response surface approximation and multi-objective genetic algorithm was employed for multiple objectives. They concluded that the multi-floor microchannel heat exchangers can manage heat flux of 800W/cm^2.

Reddy et al. [11] performed multi-objective constrained optimization to find optimum geometry of pin fins, inlet water pressure for cooling of electronic components having a uniform heat flux on 4mm by 3 mm footprint and a centrally located hotspot. Minimizing the maximum substrate temperature below 85°C and minimizing the pumping power were the multiple objectives of the convection process. Response surface analysis with genetic algorithm having Pareto sensitivity is used for the optimisation. After optimisation the structural analysis was also performed to confirm the structural integrity of the micro pin fin array.

Yadav et al. [12] have used pin fins as the method of enhancement in the microchannel performance. Three different configurations based on position of fins were studied. They deduced that the overall enhancement factor is greater than one for all three configurations. Optimisation of the enhanced microchannels was carried out by using univariate search technique. Number of pin fins, pitch, diameter and height of pin fins are optimized. Heat transfer enhancement of 160% is obtained for optimised design. Jia et al. [13] created a fan shaped fin inserted in the microchannel for performance enhancement. They also created different configurations based on fin positions and then optimised the various parameters of pin fin and microchannel. They found the optimum parameters as relative fin diameter of 0.375, relative fins space of 1, and relative fin height of 0.04 which for Reynolds number of 637 reached an enhancement factor of 1.55.

Wang et al. [14] experimentally studied the effect of pillar/pillars in upstream microchannel heat sink on thermal performance. Six different designs of which three were of a single pillar having circular, triangular or diamond shape, one design having two circular pillars, one design containing circular pillar of larger diameter and last design of conventional microchannel were studied. The design was fabricated on one side of silicon wafer using deep reactive ion etching (DRIE) technique. On the other side of wafer, same technique was used to create contact pad holes, inlet and outlet. The heater was created by sputtering of Titanium (120 nm thick) and Aluminium (1µm thick) on Pyrex wafer and both Pyrex wafer and silicon substrate were anodically bonded to form the design. The heater also acts as resistance temperature detector and is used to measure the average bottom surface temperature. The microchannel was subjected to a heat flux of 8W/cm^2 while the coolant flow of air was corresponding to Reynolds number of 90 to 5600. Particle image velocimetry technique was used to study the velocity field and calculate turbulent kinetic energy. They concluded that the presence of pillar instead of conventional microchannel improved the heat transfer performance by a factor of two. Of the different shapes of same cross sectional area, triangular pillars gave the best results. Velocity measurements indicated that the large scale vortex interactions helped in fluid mixing and increased heat transfer.

Thus most of the research work is in the pin-fin array. Very less work is done in the field of flow disturbance in microchannel using pin-fins. According to forced convection mechanism theory, for a constant heat flux boundary conditions, the temperature difference between the coolant and channel wall remains constant along the flow length in the fully developed condition. Fully developed flow is one of the main reasons for this condition. So to improve the thermal performance, a mechanism to disturb the fully developed flow is needed. In this paper a new configuration of flow disturbing micro pins in a microchannel is numerically analysed. In this configuration, pins are introduced into the microchannel with the help of the top acrylic cover. The size, shape and location of pins are the variables against which the performance is studied.

II. ENHANCED MCHS DESIGN

The numerical analysis was carried out using the commercial computational fluid dynamics software ANSYS FLUENT 18.2. The footprint area for both conventional MCHS and P-MCHS is 25mm x 25mm. The number of channels is 25. To reduce the computational time, only a single channel is considered for simulation (Fig.2.1). The pins of circular cross section are uniformly spaced along the entire length of the microchannel. The dimensions of the conventional MCHS and pin enhanced microchannel heat sink (PE-MCHS) are given in table 1.

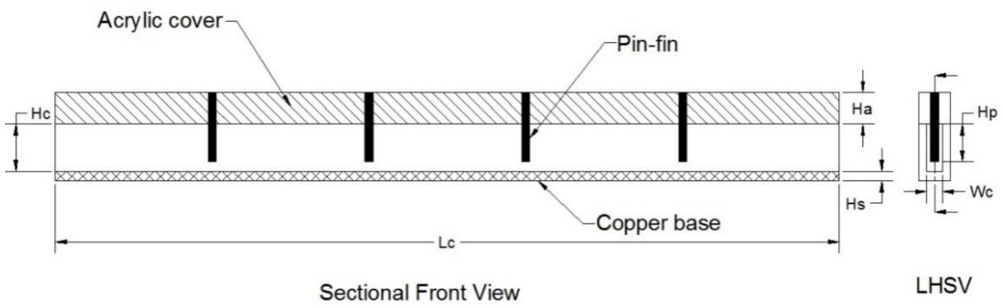

Figure 2.1. Pin enhanced configuration in a microchannel

Table 2.1 Dimensions of pin enhanced MCHS (All dimensions in mm)

Sr. No.	Conventional Channel dimension			Enhanced MCHS Pin dimension		
	L_c	W_c	H_c	pin height H_P	H_a	H_s
1	25	0.5	1.5	1.5 to 1.1	1	0.3

III. SIMULATION MODEL

Three-dimensional numerical analysis was done for the simulation model shown in figure 3. The computational domain consists of single channel with side walls while the pins are suspended from top cover as shown in figure 2. The model consists of two solid regions the first is microchannel and the pins and the second is top cover. The space in between these solids is the fluid region. The pins and side walls and base of microchannel are of copper while top cover is of acrylic.The pin height H_P is always less than channel height H_c(figure 3).The numerical analysis is carried out for conventional MCHS with no pins and enhanced MCHS with different fin heights as shown in table 1.

The numerical analysis is done by solving the governing equations for conservation of mass, momentum and energy.

The governing equations are as follows:

Continuity equation:

$$\nabla \cdot (\rho \vec{v}) = 0 \tag{1}$$

Momentum equation:

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

Energy equation for liquid:

$$\nabla \cdot (\rho \vec{v} \, (CpT) = \nabla \cdot (k \nabla T) \tag{3}$$

Energy equation for solid:

$$\nabla \cdot (k \nabla T) = 0 \tag{4}$$

The computational domain is meshed using edge mesh with size of 25µm (figure 4 & 5). The total number of elements created are 3436472. The channel and pin surfaces are treated as no-slip boundary conditions. Constant heat flux boundary condition of 65 W/cm^{2}is applied to the bottom heater. A fully developed velocity profile corresponding to the various flow rates having corresponding Reynolds number of 350 to 620 with inlet temperature of 300 K is applied to inlet and pressure outlet condition at outlet.

The three dimensional, double precision, solver is used with SIMPLE scheme for pressure-velocity coupling. For the spatial discretization scheme, second order is used for the pressure equation while second order upwind scheme is used for both momentum and energy equations. In solution controls, the under-relaxation factors used are: 0.5 for pressure and for momentum, and 1.0 for density, momentum and energy. In the monitors, the residual convergence criterion of 10-6 is set for all equations. Copper with constant thermal conductivity of 387.6 W/mK is chosen as the solid material for pins and microchannel. Water with temperature dependent properties is assigned to fluid.

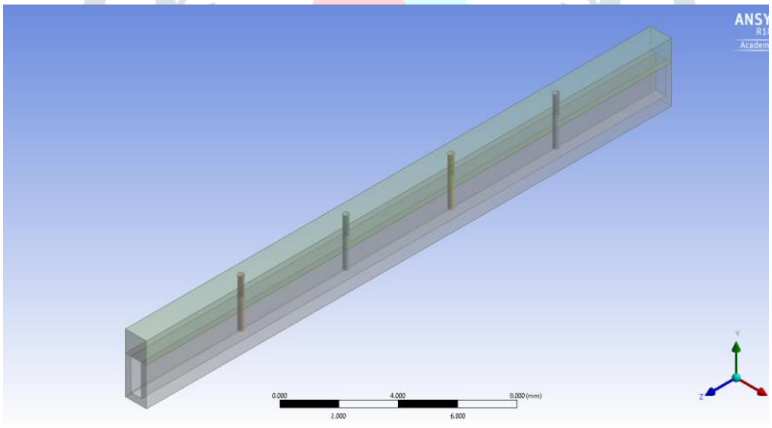

Figure 3.1 Simulation model of pin enhanced MCHS

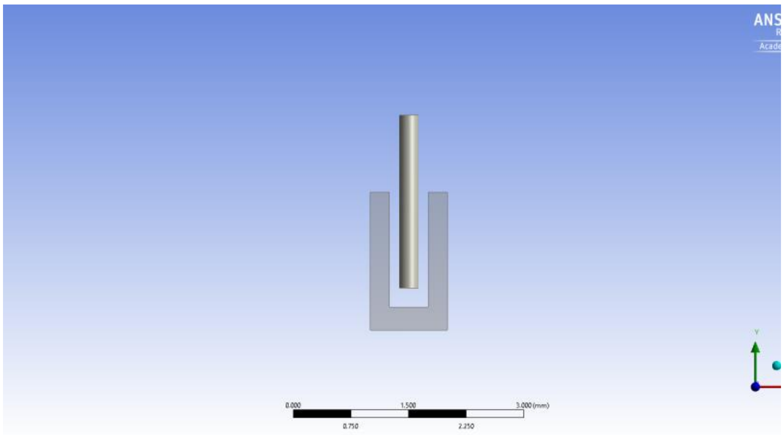

Figure 3.2 Simulation model showing the gap between bottom surface of microchannel and pin inserted from top

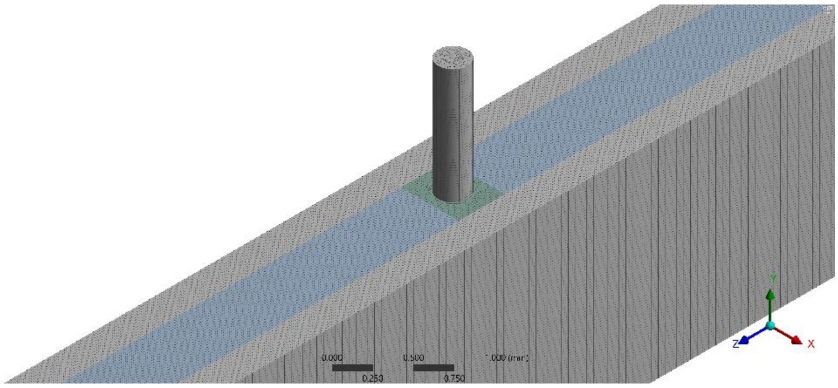

Figure 3.3 Meshing of microchannel, fluid region and pin of pin enhanced MCHS

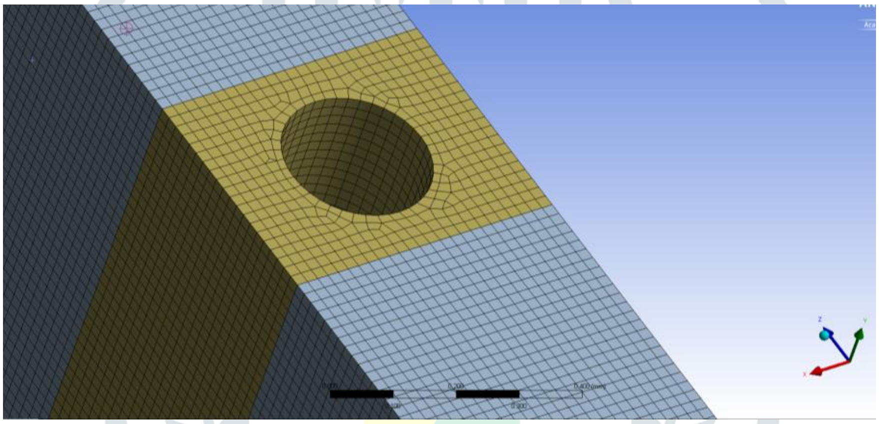

Figure 3.4 Meshing of fluid region of pin enhanced MCHS

IV. DATA REDUCTION

The Reynolds number is given by the expression:

$$Re = \frac{\rho U_m D_h}{\mu} \quad (5)$$

where ρ, U_m,μ are fluid density, mean velocity, and dynamic viscosity respectively. D_h is the hydraulic diameter of the channel and is given by

$$D_h = \frac{2\,H_c W_c}{(H_c + W_c)} \quad (6)$$

The average heat transfer coefficient is given by

$$h_{ave} = \frac{q_w A_{bottom}}{A_{con}(T_s - T_f)} \quad (7)$$

Where q_w is the heat flux applied to the bottom of MCHS, A_{bottom} is the bottom surface area, A_{con} is the convection heat transfer area, T_s, T_f are the average surface temperature and fluid mean temperature respectively.

The average Nusselt number is obtained by

$$Nu_{ave} = \frac{h_{ave} D_h}{k_f} \quad (8)$$

Non-dimensional length factor

$$Z^* = \frac{z}{L_c} \quad (9)$$

where z is the distance from inlet along the length of channel.

V. RESULTS AND DISCUSSION

The simulation was initially carried out to find the effect of pin height on the thermal performance.Two important parameters for checking the thermal performance are the maximum bottom heater temperature, and temperature gradient at the bottom heater which indicates the possibility of formation of hotspots and formation of thermal stresses due to lack of uniformity in temperature distribution in the electronic components being cooled. Figure 6 shows the variation of temperature along bottom heater length for different configurations. All temperature profiles show non-linear behaviour. All enhanced MCHS show lower temperature variations than the conventional MCHS. The enhanced MCHS with pin height of 1.2mm shows the lowest temperature variation. The introduction of pins from the top helps in breaking of the continuous flow of fluid. In continuous flow the core region of the fluid remains unaffected and is not useful in absorbing the heat from the channel walls. The pins not only disturb the flow in horizontal (ZX) plane, but also in vertical (YZ) plane (figure 8 & 9). Due to lesser height of suspended pins than channel height,an orifice like structure is formed between pin bottom end and channel bottom surface which restricts the flow and increases the velocity. The core region of the fluid thus is disturbed causing more fluid to absorb the heat from microchannel walls and increasing the heat removing capacity of the design. Temperature gradient for different configuration is given in table 2. The enhanced MCHS with $H_p$of 1.5mm show the lowest temperature gradient followed by pin of height of 1.2mm. The temperature gradient in the bottom heater reduces by 33% from 1.2 K/mm in conventional MCHS to 0.8 K/mm for H_p of 1.5 mm. It reduces by 27% for H_p of 1.2mm. This is due to the fact that as clearance between base and pins increase upto 1.2mm, there is an accelerated flow in the clearance region producing more mixing of fluid (see figure 7) and higher heat transfer performance. Effects of fluid by-passing the pins become dominant when the clearance region is further increased (>1.2mm), resulting in lower heat transfer performance. The introduction of pin increases the pressure drop; it is highest for full height pin and decreases with decreased pin height (see Table 3). This is due to the fact that as pin height reduces there is reduction in the flow disturbance. Simulation for enhanced MCHS with $H_p$of 1.2mm subjected to different flow rates is carried out and its effect on the bottom heater temperature is shown in figure 9. As the flow rate increases the temperature along the bottom heater decreases.

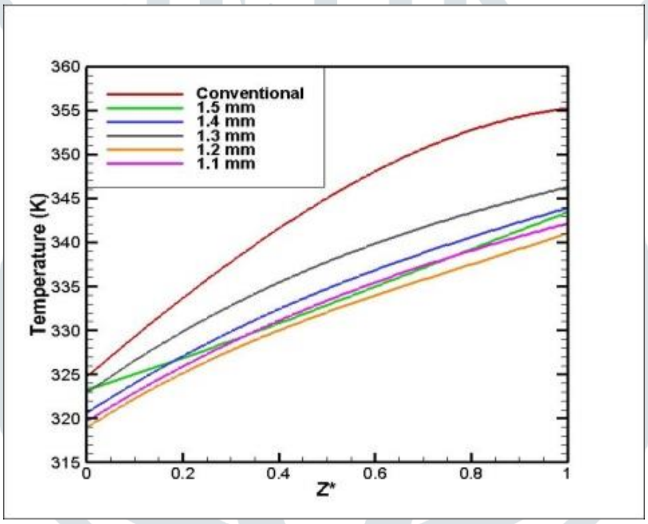

Figure 5.1 Bottom heater temperature for differentMCHS configurations

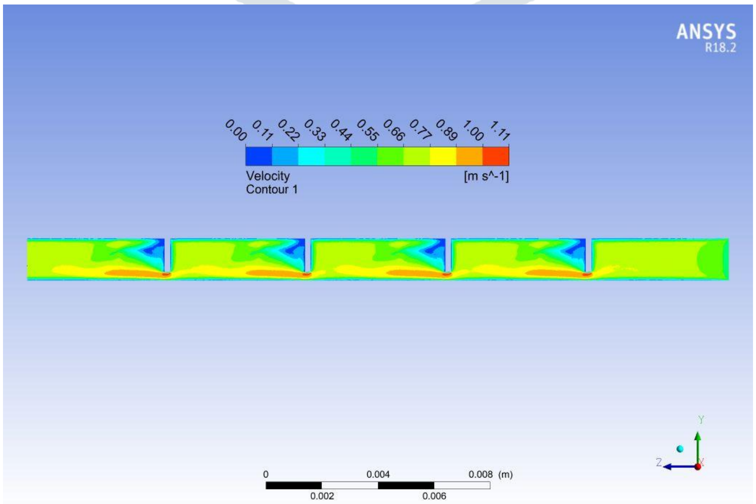

Figure 5.2 Velocity contour of fluid along the vertical plane

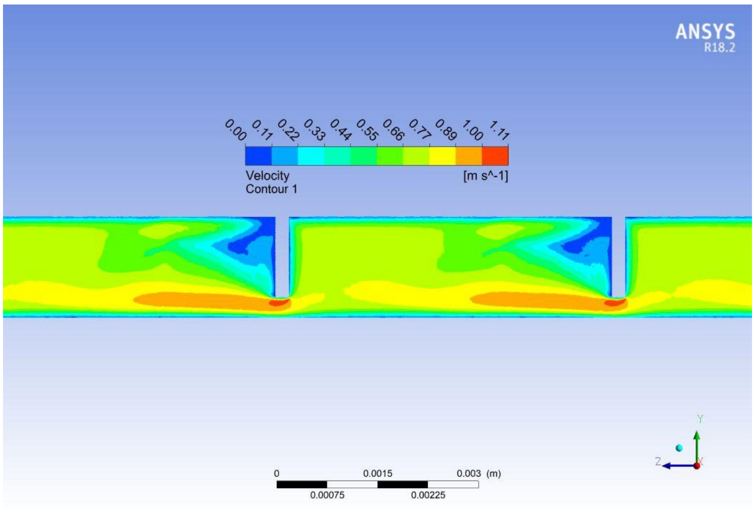

Figure 5.3 Close view of velocity contour of fluid along the vertical plane

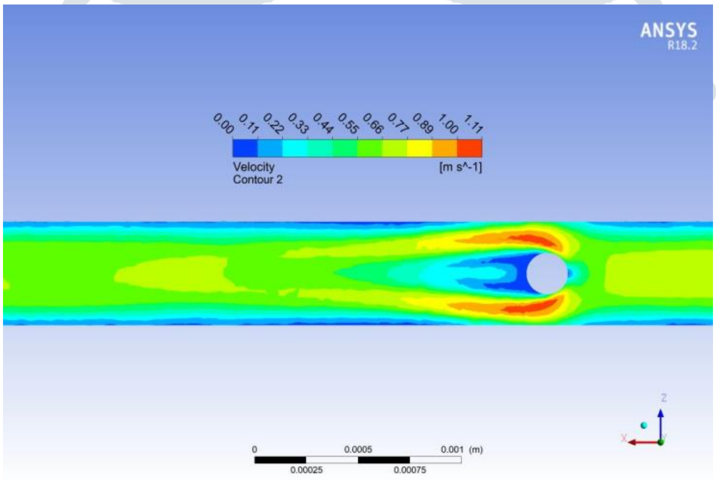

Figure 5.4 Velocity contour of fluid showingfluidmixing(top view)

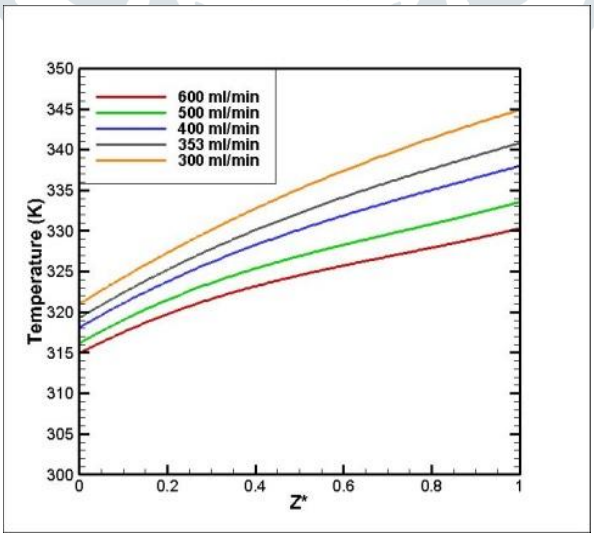

Figure 5.5 Bottom heater temperature for Hp= 1.2 mm for different flow rates

Table 5.1 Temperature gradient for various configurations

H_p (mm)	Conventional MCHS (without pin-fin)	Enhanced MCHS				
		1.5	1.4	1.3	1.2	1.1
Temp. Gradient $\left(\frac{\partial T}{\partial x}\right)$ K/mm	1.20	0.80	0.92	0.91	0.88	0.89

Table 5.2. Pressure drop for various configurations

H_P (mm)	Conventional (without pin-fin)	Enhanced MCHS				
		1.5	1.4	1.3	1.2	1.1
ΔP (kPa)	0.51	1.25	1.26	1.14	1.04	0.97

The heat transfer correlation for the pin-fin as given in classical heat transfer relates the Nusselt number with Reynolds number and Prandtl number and is given by

$$\mathrm{Nu} = \mathrm{cRe^m Pr^n} \qquad (10)$$

The exponent m and n are mostly constant and show the relation of Reynolds number and Prandtl number with Nusselt number. In many studies n is assumed as 1/3. Mei et al. [] proposed a new correlation which accounts for the pin-fin tip clearance. The proposed correlation is:

$$\mathrm{Nu} = \mathrm{c}\left(\frac{\mathrm{h_c}}{\mathrm{h_f}}\right)^{\mathrm{j}} \mathrm{Re^m Pr^{0.333}} \qquad (11)$$

Based on these correlations, a new heat transfer correlation (Eq.) showing the effect of clearance between pin and base (Hc/Hp), Reynolds number and Prandtl number with Nusselt number is developed. A non-linear multiple regression analysis is used to determine the various coefficients and the correlation is given by

$$\mathrm{Nu} = 0.977\left(\frac{\mathrm{H_c}}{\mathrm{H_p}}\right)^{-0.06} \mathrm{Re^{0.85} Pr^{-0.75}} \qquad (12)$$

The variation between actual values obtained by numerical analysis and by correlation (equation 12) is obtained by mean absolute error (MAE)

$$\mathrm{MAE_{Nu}} = \frac{1}{n}\sum_{k=1}^{n} \frac{\mathrm{abs(Nu_{sim} - Nu_{pred})}}{\mathrm{Nu_{sim}}} \times 100\% \qquad (13)$$

where 'n' is the number of data points.

Maximum value of MAE is 18% indicating the closeness of the correlation.

VI. CONCLUSIONS

A novel pin configuration is studied to improve the performance of conventional MCHS.

- The enhanced MCHS with H_p of 1.2 mm gives the optimum results. For the same flow rate, it has the lowest temperature along bottom heater (Average temperature difference of 12 K lower than conventional MCHS). The temperature gradient in the bottom heater reduces by 33% from 1.2 K/mm in conventional MCHS to 0.8 K/mm for H_p of 1.5 mm. The pressure drop for this configuration is more by 100% than conventional MCHS.
- The pressure drop is highest for H_p=1.5 mm, decreases with reduction in pin height for enhanced MCHS and is lowest for conventional MCHS.
- The fabrication cost of pins protruding from MCHS base is huge compared to the pins suspended from top cover. The improvement in thermal performance is thus at very little additional cost.
- A new correlation showing the effect of pin clearance on Nusselt number is developed.

Nomenclature			
A	Area	μ	viscosity, (Pa s)
C_p	Specific heat, (kJ/kg K)		
D_h	Hydraulic diameter, (mm)	**Subscripts**	
D_p	Diameter of pin (mm)	i	inlet
Hc	Height of channel (mm)	m	mean
h	heat transfer coefficient, (W/sq. m K)	s	surface
K	Thermal conductivity (W/m K)	max	maximum
Lc	Total channel length, (mm)	c	channel
m	mass flow rate, (kg/s)	w	wall
Nu	Nusselt number	app	apparent
P	Pressure, (Pa)	bottom	base
q_w	Heat flux (W/sq. cm)	conv	convective
Re	Reynolds number	ave	average
T	temperature, (K, °C)	f	fluid
U	Velocity, (m/s)	sim	simulation

W	Width (mm)	pred	predicted
Greek		**Acronyms**	
Δp	Pressure drop	MCHS	Microchannel heat sink
$\Delta T/\Delta Z$	Temperature gradient (K/mm)		
ΔT	Temperature difference		
ρ	density, (kg/cubic m)		

VII. ACKNOWLEDGMENT

The authors would like to thank the Government College of Engineering Karad, India and AICTE Delhi, India for the computing facility and experimental setup used for this work.

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