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Numerical Analysis And Parametric Study of Micro-channel Heat Exchanger For Single Phase Water Cooling

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Abstract— Today with the miniaturization of electronic components with increased speed of computational capabilities, thermal management of microelectronic components is a challenging problem. Overheating of these micro components and micro devices led to the use of mini and micro channels in the above mentioned technologies. The aim is to eliminate as fast as possible the maximum heat quantity from these systems in order to ensure an increased reliability and functional stability. Micro channel heat exchangers with direct liquid cooling are capable of removing high heat flux. Whereas, this high heat removal capability of this device is associated with increased pressure drop penalty. Hence channel size optimization becomes necessary in selecting correct channel geometry for desired application. Present paper deals with the numerical analysis and parametric study of microchannel. This parametric study will be useful for selection of channel geometry according to system requirements. The results shows that for the same cooling requirements (for same ambient to junction temperature difference 60oC, same chip size 10mmX10mm and same heat flux 300 W/cm2) you have a choice of selecting channel geometries for: 1) Channel with hydraulic diameter of 100 μm having high convective heat transfer coefficient of 40000 W/m2 K and maximum pressure drop of 35kPa. 2) Channel with hydraulic diameter of 180 μm having low convective heat transfer coefficient of 20000 W/m2 K and minimum pressure drop of 5KPa.

Keywords— Microchannels, electronic cooling, channel geometry, pressure drop

I INTRODUCTION

Now days, there is rapid growth of applications which requires high heat transfer rate and fluid flow in relatively small channels. Such examples include electronic cooling, MEMS devices for biological and chemical analysis and cooling of laser gun etc. Development of new applications requiring cooling of components by passing single phase liquid in microchannels has motivated researchers to study thermo-hydrodynamic performance of microchannels.

In the present work, numerical analysis is carried out to decide channel geometry required for thermal performance with minimum pressure drop condition. The constraints for channel geometry optimization include maximum junction temperature difference, chip size, heat flux and pressure drop.

II LITERATURE REVIEW

Microchannels were first proposed for electronic cooling applications by Tuckerman and Pease [1]. They have experimentally demonstrated direct circulation of water in microchannels fabricated in silicon chips for electronics cooling applications. They proved that microchannel heat sink was able to dissipate 790 W/cm^2 with a maximum substrate temperature to inlet water temperature difference of 71°C.

However, the pressure drop was quite large it is around 200 KPa with plain microchannels. Mark E, Steinke and Satish G Khandlikar [2], reviewed single phase heat transfer techniques for applications in micro channels, minichannels and microdevices. The major techniques include flow disruptions, flow pulsations, breakup of boundary layer, entrance region, vibration, electric fields, swirl flow, secondary flows and mixers. In this paper applicability of these techniques for single phase flows in mini channel and micro channel is evaluated. The micro channel and minichannel single phase heat transfer enhancement devices will extend the applicability of single phase cooling for critical applications, such as microprocessors cooling.

Satish G Khandlikar and Harshal R Upadhye [3], have done channel size optimization. As the heat flux increases beyond about 200W/cm^2, the heat transfer and pressure drop characteristics of the plain channels dictate the use of turbulent flow through the channels, which suffers from an excessive pressure drop penalty. Therefore, they have done theoretical analysis of a 10×10^{-3} m $\times$ 10×10^{-3} m silicon chip incorporating plain microchannels for heat transfer and pressure drop and presented results in parametric plots. These results show that the enhanced structures are capable of dissipating heat fluxes extending beyond 300W/cm^2 using water as the coolant in a split-flow arrangement with a core pressure drop of around 35 kPa.

Evan G Colgan and et al [4], describes a practical implementation of a single-phase Si microchannel cooler designed for cooling very high power chips such as microprocessors. Through the use of multiple heat exchanger zones and optimized cooler fin designs, a unit thermal resistance 10.5^{0}C-mm^2 /W from the cooler surface to the inlet water was demonstrated with a fluid pressure drop of 35 KPa.

Gaurav Agarwal, Manoj Kumar Moharana and Sameer Khandekar [5], have studied thermo-hydrodynamics of simultaneously developing single phase flow through a mini-channel array experimentally. They observed that developing flow provides very high heat transfer coefficients in entrance region and therefore of interest for mini micro scale heat flux removal application.

J D Patil and Dr. B S Gawali [8], have done numerical analysis of microchannel heat exchanger that is used to select appropriate channel geometry by using least thermal resistance value. Results obtained from analysis are used to plot convective heat transfer coefficient, thermal resistance and pressure drop.

III TERMS AND TERMINOLOGY USED FOR ANALYSIS

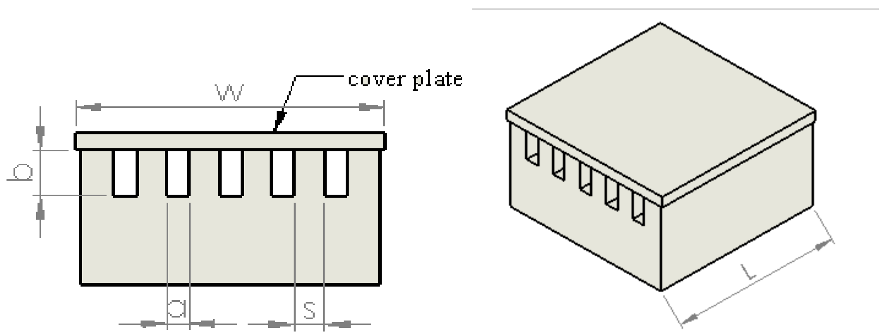

Fig. 1 ***Microchannel geometry***

Channel aspect ratio α_c is defined as the ratio of the channel width to the channel depth.

$$\alpha_c = \frac{a}{b}$$

Fin aspect ratio α_f is defined as the ratio of the fin thickness to the fin height.

$$\alpha_f = \frac{s}{b}$$

Fin spacing ratio β is defined as the ratio of fin aspect ratio to the channel aspect ratio.

$$\beta = \frac{\alpha_f}{\alpha_c}$$

Channel width is depend on chip width, number of channels, and the fin spacing ratio by the relation,

$$a = \frac{W}{n + (n+1)\beta}$$

Effective channel wall heat transfer surface area, considering fin (thickness of channel wall) efficiency effect, is given by

$$A_w = (2\eta_f b + a)Ln$$

Fin efficiency

$$\eta_f = \frac{\tanh(mb)}{mb}$$

Where $m=\sqrt{\frac{hP}{k_f A_c}}$

Nusselt number

$$Nu=\frac{hd}{k}$$

Hydraulic diameter

$$d=\frac{4ab}{2(a+b)}$$

Conduction thermal resistance of microchannel (base thickness of channel is taken as 1.5 times height of channel)

$$R_{cond}=\frac{1.5\,b}{k_f WL}$$

Convective thermal resistance

$$R_{conv}=\frac{1}{hL(2(a+b)n)\eta_f}$$

Resistance due to heating of the fluid as it absorbs energy passing through the heat exchanger.

$$R_{heat} = \frac{1}{C_p m_t}$$

Total thermal resistance

$$R_{Total} = R_{cond} + R_{conv} + R_{heat}$$

IV FLUID FLOW PARAMETERS

For the hydrodynamically developing flow, the dimension less axial distance x^+ is defined as

$$x^+ = \frac{x/d}{Re}$$

Axial pressure drop is expressed in terms of the incremental pressure drop as

$$\Delta p = \frac{2\ f\,Re\ \mu u_m x}{d^2} + K\ x\ \frac{\rho u_m^2}{2}$$

Where f is Fanning friction factor and K(x) is some times referred as the incremental pressure defect. It increases monotonically from a value of zero at x=0 to a constant value in the hyrodynamically developed region at $x > L_{hy}$ This constant value is referred as Hagenbach's factor.

$$L_{hy} = 0.05\,Re\,d$$

$$K\ x = 0.6796 + 1.2197\alpha_c + 3.3089\alpha_c^2 - 9.5921\alpha_c^3 + 8.9089\alpha_c^4 - 2.9959\alpha_c^5$$

and

$$f\,Re = 24\ \ 1 - 1.3553\alpha_c + 1.9467\alpha_c^2 - 1.7012\alpha_c^3 + 0.9564\alpha_c^4 - 0.2537\alpha_c^5$$

Measuring local pressure along the flow is difficult in microchannel, hence researchers generally measures pressure drop across the inlet and outlet manifolds. This pressure drop measurement represent the combined effect of entrance and exit losses, developing region effects and the core frictional losses. Thus, the measured pressure is sum of these components and is calculated by following equation.

$$\Delta p_{Total} = \frac{2\ f\,Re\ \mu u_m x}{d^2} + K\ x\ \frac{\rho u_m^2}{2} + K_c\left(\frac{\rho u_m^2}{2}\right) + K_e\left(\frac{\rho u_m^2}{2}\right)$$

Where, K_c and K_e are contraction and expansion loss coefficient due to area change.

The incremental pressure drop, K_c, K_e and friction factor are obtained from Kakak et al.[6] and Kandlikar, et al .[7]

V HEAT TRANSFER PARAMETERS

The dimensionless axial distance x^* is defined as

$$x^* = \frac{x/d}{Pe} = \frac{x/d}{Re\,Pr}$$

The thermal entrance length

$$L_{th} = 0.1 Re\,Pr\,d$$

The local Nusselt number for thermally developing region are obtained from Kandlikar, et al.[7]

VI OBJECTIVE

The objectives of present work are

1. To develop MATLAB program to analyze the heat transfer and pressure drop in a micro channels.
2. To present parametric study of micro channel geometry.

VII ASSUMPTIONS

The following assumptions are considered for analysis

1. Size of micro channel chip 10 X 10^{-3} m X 10 X 10^{-3} m.
2. Depth of micro channel, b is 300 X 10^{-6} m (300µm).
3. Constant heat flux of 300W/cm^2 is applied.
4. Inlet temperature of water is taken as 27^{o}C (300K).
5. Maximum mean temperature of the channel wall at outset is maintained below 360 K.
6. Constant properties are assumed for cooling fluid (water) and the channel wall material copper.
7. Number of channels is taken from 50 to 100 with increment of 10.
8. Fin spacing ratio is taken from 0.2 to 1 with increment of 0.1
9. Low starting value of Re is assumed e.g. Re=50 and required analysis is done.
10. Mass flow rate in a single channel m_c in kg/s is calculated for assumed Re.

VIII SOLUTION METHOD

One channel symmetric with center line of channel width is considered for analysis. Length of channel is divided into a certain number of equal divisions, in our analysis 10. Hence one symmetric channel is divided in ten slices from inlet to outlet as shown in Figure 2. Heat dissipated per slice Q_{div} is calculated for given data. Energy balance is applied for each slice starting from first slice where inlet temperature, per channel mass flow rate m_c, specific heat C_p and Q_{div} known and T_{out} is calculated for following equation.

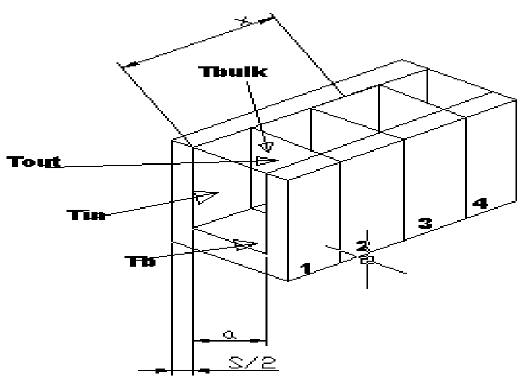

Fig. 2 Single channel is divided into smaller number of division for analysis.

$$T_{out} = \frac{Q_{div}}{m_c C_p} + T_{in}$$

This calculated T_{out} is used as T_{in} for next slice. The bulk temperature T_b for this slice is taken as the mean of T_{in} and T_{out}. Fluid properties required for further analysis are taken at this bulk mean temperature. The average surface temperature of this slice is calculated heat transfer by mode of convection from this slice by using equation

$$T_s = \frac{Q_{div}}{hA_{div}} + T_b$$

The temperature difference between this T_s and temperature of water at inlet 30^o C is calculated. If this is more than 60^o C then the assumed value of Re is increased by small increment and calculations are repeated. This process is repeated until required temperature difference of 60^o C is reached. For the converged value of Re mass flow rate and pressure drop is calculated. This process is repeated for different number of channels and fin spacing ratios. The results obtained are plotted as parametric plots.

IX VERIFICATION OF NUMERICAL ANALYSIS

Verification of numerical analysis is done by comparing experimental results obtained by Tuckerman and Pease with numerical results obtained for the same geometric, flow and thermal conditions. The results of total thermal resistance and pressure drop for three different cases are shown in Fig. 3 (a) & (b).

Case 1: a=56µm, s=44 µm,b=320 µm,f=4.7cm^3/s and q=181W/cm^2, Case 2: a=55µm, s=45 µm,b=287 µm,f=6.5cm^3/s and q=277W/cm^2, Case 3: a=50µm, s=50 µm,b=302 µm,f=8.6cm^3/s and q=790W/cm^2.

Numerical results are seen to be in good agreement with experimental results.

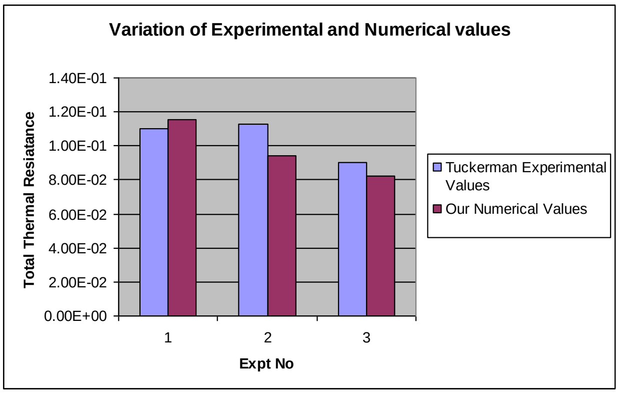

Fig 3 (a) Variation of total thermal resistances for three different cases.

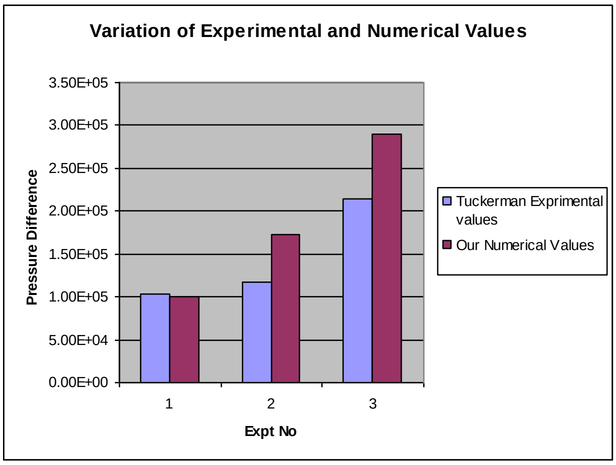

Fig 3 (b) Variation of pressure drop for three different cases.

X RESULT AND DISCUSSION

Fig. 4. Shows variation of h with respect to number of channels and fin spacing ratio. It shows that h increases with increase in number of channels and β_f. As we know that Nu is proportional to product of h and d. Fig. 5 shows variation of d with respect to number of channels and fin spacing ratio. Highest value of h is 45000 W/m$^{2\,o}$ K at n=100 and β_f=1 where value of d is lowest.

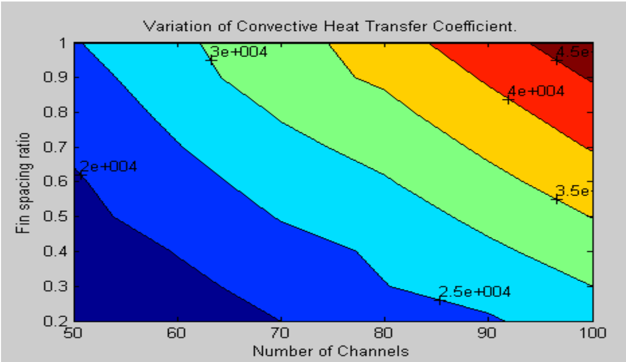

Fig. 4 Variation of convective heat transfer coefficient with number of channels and fin spacing ratio.

Fig. 6 shows variation of pressure drop in kPa with respect to number of channels and fin spacing ratio. Highest value of pressure drop of 30 kPa is obtained at n=100 and β_f=1. Value of pressure drop is directly depends on resistance to flow and resistance to flow is directly depends on hydraulic diameter available for flow.

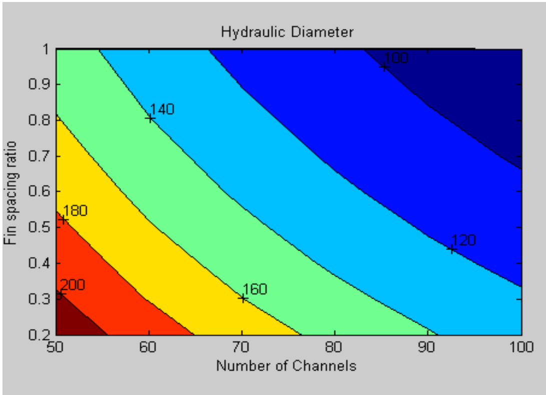

Fig. 5 Variation Hydraulic diameter with respect to number of channels and fin spacing ratio

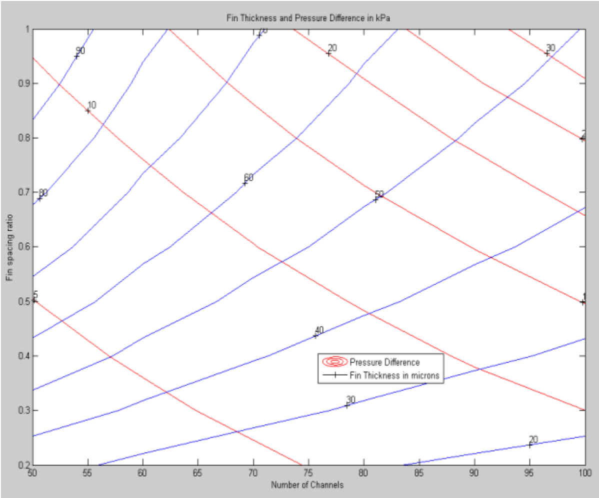

Fig. 6 Contour plot of fin spacing ratio β vs number of channel with pressure drop (red lines) and fin thickness in µm as parameter for water flow in plane rectangular micro channels at a heat flux of 3MW/m2.

From the above plots we can conclude that

1. **Performance with maximum pressure drop:** Channels with hydraulic diameter less than 100 µm, number of channels from 85 to 100 and fin spacing ratio of 0.8 to 1. Convective heat transfer rate in this range is above 40000 $W/m^{2\,o}K$ and pressure drop is around 25 kPa.
2. **Performance with minimum pressure drop:** Channels with hydraulic diameter slightly greater than 180 µm, number of channels from 50 to 65 and fin spacing ratio of 0.2 to 0.5. Convective heat transfer rate in this range is above 18000 $W/m^{2\,o}K$ and pressure drop is below 5 kPa.

Two special cases in the above mentioned two categories are compared to study effect of channel geometry on thermal resistances. The variation different thermal resistances for these two cases are shown in Fig 7(a) and (b).

Case 1) number of channels 90 and fin spacing ratio of 0.9 Case 2) number of channels 60 and fin spacing ratio of 0.3. In this case convective thermal resistance is more because of lower convective heat transfer coefficient where as conduction and heat resistances are almost same. Therefore total thermal resistance in this case is more than case no 1 hence in case 2 same cooling effect is obtained with slight higher temperature difference.

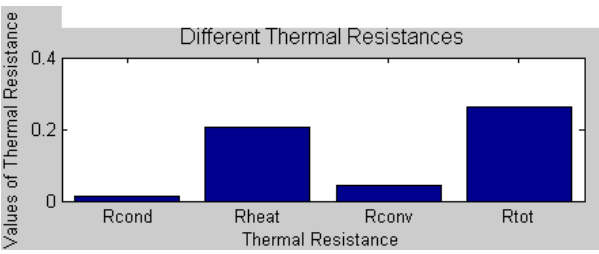

Fig. 7(a) Case 1) number of channels 90 and fin spacing ratio of 0.9 Rconduction=0.0117, Rheat=0.207, Rconvection= 0.0428, Rtotal= 0.261 and Tbase=64.7oC

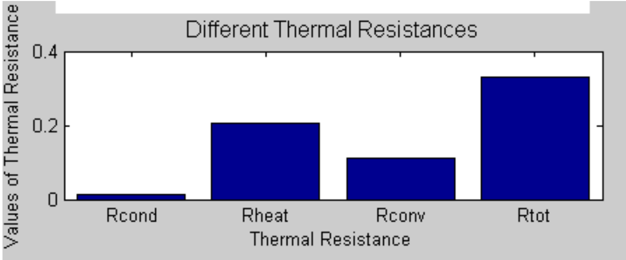

Fig. 7 (b) Case 2) number of channels 60 and fin spacing ratio of 0.3.Rconduction=0.0117, Rheat=0.207, Rconvection= 0.111, Rtotal= 0.33 and Tbase=65.2Oc

CONCLUSION

Numerical analysis and parametric study of microchannels fabricated on 10mmX10mm chip size, 300 W/cm^2 heat flux and 60^{o}C junction to ambient temperature difference is studied. Results are presented in the form of contour plots. This parametric study is used for selection of channel geometry according to system requirements. The results shows that for the same cooling requirements you have a choice of selecting channel geometries for:

1) Channel with hydraulic diameter of 100 µm having high convective heat transfer coefficient of 40000 W/m$^{2\,o}$ K and maximum pressure drop of 25kPa.

2) Channel with hydraulic diameter of 180 µm having low convective heat transfer coefficient of 18000 W/m$^{2\,o}$ K and minimum pressure drop of 5kPa.

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