

# Designing and Optimising the Parameters of Micro Channels

Rajesh Pant, Jasmeet Kalra, Pankaj Negi, Vijay Kumar

**Abstract:** This paper documents the optimization of different parameters of micro channel heat sink which enhance the heat transfer. The objective is to find the major thermal resistance in micro channel and its effect on other parameters. Water is used as a coolant and the initial values of convective heat transfer coefficient and volume flow rate are  $30000 \text{ W/m}^2\text{K}$  and  $1 \text{ lpm}$  respectively. Different graph are plotted between pressure drop, heat transfer co-efficient, pressure drop, thermal resistance and flow rate to finally achieve the optimized value of channel width and height, hydraulic diameter, thermal resistance and pressure drop. The result achieved are in good agreement with the previous researches.

**Keywords:** Micro channels, heat sink, MCC

## I. INTRODUCTION

Today the world is in need of energy packed system with smallest possible sizes to keep them portable and handy. Micro devices are the those systems whose dimension lies between  $1\mu\text{m}$  and  $1 \text{ mm}$ . The performance and size of electronic product plays an important role for its survival in market. With miniaturization and increases in the working capacity of a electronic product, heat dissipation became a big challenge. Microchannel heat sinks are designed to counter that problem. These channels efficiently dissipates heat from small area. These channels are machined in conductive solids like silicon or copper in which the liquid flows and removes heat. Heat capacity and heat transfer coefficient of fluid is an important parameter for enhanced cooling of any system.

## II. LITERATURE REVIEW

One of the most important parameter for designing an efficient micro channel is to reduce its overall thermal resistance. The pioneer work done by Tuckermann and Pease [1981] has shown that a micro flow passage having dimension  $50\mu\text{m}$  wide and  $302 \mu\text{m}$  deep has a minuscule thermal resistance of  $0.09 \text{ }^\circ\text{C/W}$ . Phillips [1988] provided an logical model to find out the thermal resistance of micro channel, outcomes of this model were in good accord. For a double chip power module Gilliot et al [1998] evaluated the thermal performance of a prototype of heat sink and demonstrated that it could dissipate  $230\text{-}350 \text{ W/cm}^2$  with a temperature rise of  $35 \text{ }^\circ\text{C}$ .

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On the experimental basis of Kawano et al (1998), Fedorov and Viskanta [2000] made a 3D model to inquire heat transfer in these channels and found that the average temperature of each wall in the direction of flow was same and this proved the dependency of thermal properties over temperature. Qu and Mudawar [2002] validated the use of Navier–Stokes equations for micro channels flow. According to Knight et al. [1992] the thermal resistance of Tuckermann and Pease [1981] can be decreased by 35% by allowing the flow to be turbulent. X.J. Wei et al. [2003] found that driving power increases by five times if flow is turbulent. Bau [1998] through a optimization study showed that thermal resistance and difference in temperature can be reduced by varying the diameter of micro channel. The main thermal resistance identified by X.J. Wei et al. [2003] in a micro channel are conductive, convective and heat/bulk resistances. The resistance offered by bulk motion in a micro channel is called bulk resistance.

## III. DESIGN THEORY

The measure of performance of microchannel is computed through its thermal resistance and it is denoted by  $R$  which is equal to

$$R = \frac{\Delta T}{Q} \quad (i)$$

$\Delta T$  is the temperature difference between laser material and the initial temperature of coolant and  $Q$  denoted the dissipated heat. In forced convection,  $R$  is almost independent of  $Q$ . The total thermal resistance  $R$  is equal to the sum of resistance due to conduction i.e.  $R_{cond}$ , due to convection i.e.  $R_{conv}$  and heat exchanger fluid temperature rise i.e.  $R_{heat}$ , hence

$$R = R_{cond} + R_{conv} + R_{heat} \quad (ii)$$

Conductive resistance can be made very small by containing the exchanger nearest to heat source. Copper has very high thermal conductivity i.e.  $398 \text{ W/mK}$  hence it is used as a substrate in microchannels.  $R_{heat}$  can also be reduced by using a liquid with better heat capacity  $\rho C_p$ , at a large flow rate  $f$  ( $R_{heat} = \frac{1}{\rho C_p f}$ ).  $\rho C_p$  of water is equal to  $4.18 \text{ J/}^\circ\text{Ccm}^3$  which is sufficiently high hence water will be a good selection as a coolant.

Since the magnitude of  $R_{cond}$  and  $R_{heat}$  can kept rather low through the corrective steps discussed above but  $R_{conv}$  will be a dominant factor while designing microchannels heat sinks. Even when water will be allowed to flow over the surface to be cooled convective resistance will be of higher magnitude than other resistance. It is therefore necessary to study convective heat transfer theory.

Let us assume than there are parallel  $n$  channels of each length  $L$  imbedded on a substrate having equal length and width ( $W$ ). A coolant is allowed to flow in these channels absorbing constant amount of heat  $Q/nL$  from its walls. We have used various separate channels just to multiply the surface area by a factor of  $\alpha$  where  $\alpha$ = total wall surface area which is in contact with fluid/area of the slab. so convective heat transfer co-efficient i.e.

$$h=Q/nLp(T_s-T_f) \quad (iii)$$

Where  $T_s$  and  $T_f$  is surface temperature and liquid temperature,  $p$  is perimeter. Then  $R_{conv}=1/hnLp=1/h\alpha LW$ , Hence for keeping convective thermal resistance to be low, the value of  $h$  and  $\alpha$  should be kept large.  $\alpha$  directly depends on surface area i.e. using fins.

$$R_{conv}=1/hnLp \text{ and } R_{heat} = \frac{1}{\rho C_p f}$$

$h$  is calculated by:

$Nu = \frac{hd}{k_f}$ , the Nusselt number, a dimensionless heat transfer coefficients.

$Pr = \frac{\mu C_p}{k_f}$ , the Prandtl number, a property of the fluid ( $Pr=6.4$  for water at  $23^\circ C$ ).

$Re = \frac{vD\rho}{\mu}$ , the Reynolds number

Where  $D$  is called hydraulic diameter of channel and it is given by  $D=4*(\text{Cross sectional area})/\text{Perimeter}(P)$ . For rectangular channels of high aspect ratio  $D=2*W_c$  (channel width). Since  $W_c$  is kept very small, by keeping it close to slab so that  $R_{cond}$  remains small, we tentatively assume laminar flow (when  $Re \leq 2100$ ).  $Nu$  is monotonically decreasing function of  $x/(D*Re*Pr) \ll 2$ , where  $x$  varies between 0 and  $L$  ( $0 \leq x \leq L$ ). Asymptotic formulas are:

$$Nu \propto \left(\frac{x}{D*Re*Pr}\right)^{-1/3} \text{ for } x = x/(D*Re*Pr) \ll .02 \quad (iv)$$

$$X = \frac{L}{D_h Re_d h Pr}$$

Now for we have different values of Nusselt number depending upon the flow type of flow for fluid.

So from the above equation it is clear that for achieving very high convective heat transfer, we have to reduce  $d$  to very small width. But this size also depends on coolant viscosity. The volumetric flow rate of pump gets reduced due to the reduction in channel width which in return increases  $R_{heat}$ .

#### IV. CALCULATION OF MICROCHANNEL

The palner heat source is at substrate's front surface (Length  $L$ , width  $W$ ) and factor of multiplication  $\alpha = \frac{2z}{W_c+W_w}$  where  $W_c$  is coolant carrying width,  $z$  is depth and  $W_w$  is adjacent wall width. Finite wall conductivity can be accounted by a correlation factor denoted by  $\eta^{-1}$ , reciprocal of fin efficiency. For high aspect ration,  $D$  is taken as  $2w_c$ .

$$R_{conv} = \frac{2}{k_f Nu LW} (w_c \alpha^{-1} \eta^{-1}) \quad (v)$$

And efficiency of fin is given by

$$\eta = \frac{\tanh N}{N} \quad (vi)$$

Where

$$N = (2h/k_w w_w)^{1/2} z$$

$$N = (Nu k_f / k_w)^{1/2} \frac{w_c + w_w}{2(w_c + w_w)^{1/2}} \alpha \quad (vii)$$

$\eta$  is thus a consistently reducing function of  $N$ , with  $\eta \approx 1$  for  $N \ll 1$  and  $\eta \approx N^{-1}$  for  $N \gg 1$ .

Mean Flow velocity can be calculated by  $v = w_c^2 P / 12 \mu L [1]$ .

The flow rate is given by:  $f = \frac{1}{2} v W w_c \alpha$ . Whence

$$R_{heat} = \frac{1}{\rho C_p f} = \frac{24 \mu L}{\rho C_p P W} (w_c^{-3} \alpha^{-1}) \quad (viii)$$

To reduce convective resistance, design variables  $w_c, w_w$  and  $\alpha$  must be optimized, which means  $w_w = w_c$ .

Hence we reach at a conclusion that for the maximum efficiency of the heat transfer through the fins we must have the channel width and the gap between the channels to be equal. Both  $R_{conv}$  and  $R_{heat}$  with increasing  $\alpha$ , however the fin efficiency  $\eta$  rolls of as  $\alpha^{-1}$  for large  $\alpha$ , hence  $R_{conv}$  asymptotically approaches a lower limit. If one design the micro channel for aspect ratio= 5 than;

**Pressure Drop**:- low aspect ratio and high pressure drop increases heat transfer coefficient.

The pressure drop  $\Delta p$  is given by:

$$\Delta p = \frac{2CL\rho v^2}{ReD} \quad (ix)$$

Where  $C$  is Poiseuille number:  $C = f * Re$  where  $f$ =friction factor

And  $C$  can be given by:

$$C = 24(1 - 1.3553\alpha + 1.9467\alpha^2 - 1.7012\alpha^3 + 0.9564\alpha^4 - 0.2537\alpha^5)$$



From the above relation of Nusselt number i.e....

$$\text{Where } X = \frac{L}{D_h Re_d \epsilon_d Pr}$$

It is clearly visible that the Nusselt number is constant for the value of  $X \geq 0.1$  the designing value of the  $x$  should be less than  $0.1 (\leq 0.1)$ .

From the design calculations it is clearly visible that for the aspect ratio of 5, the value of  $x$  is always  $< 0.1$  for the minimum flow rate of  $1.00 \times 10^{-5} \text{ m}^3/\text{sec}$  i.e. 0.6 Lpm.

### V. OPTIMIZATION OF FLOW RATE AND HYDRAULIC DIAMETER

Now again from the study of figure 1 we can easily see that the heat transfer coefficient comes to be  $31800 \text{ W/m}^2\text{K}$  for the flow rate of  $6.68 \times 10^{-5} \text{ m}^3/\text{sec}$  or 4 lpm and the value of  $W_c = W_w = 100 \mu\text{m}$ ,  $z = 500 \mu\text{m}$  and the hydraulic diameter comes to be  $170 \mu\text{m}$ . But we have to optimize it for the thermal resistance and the pressure drop.

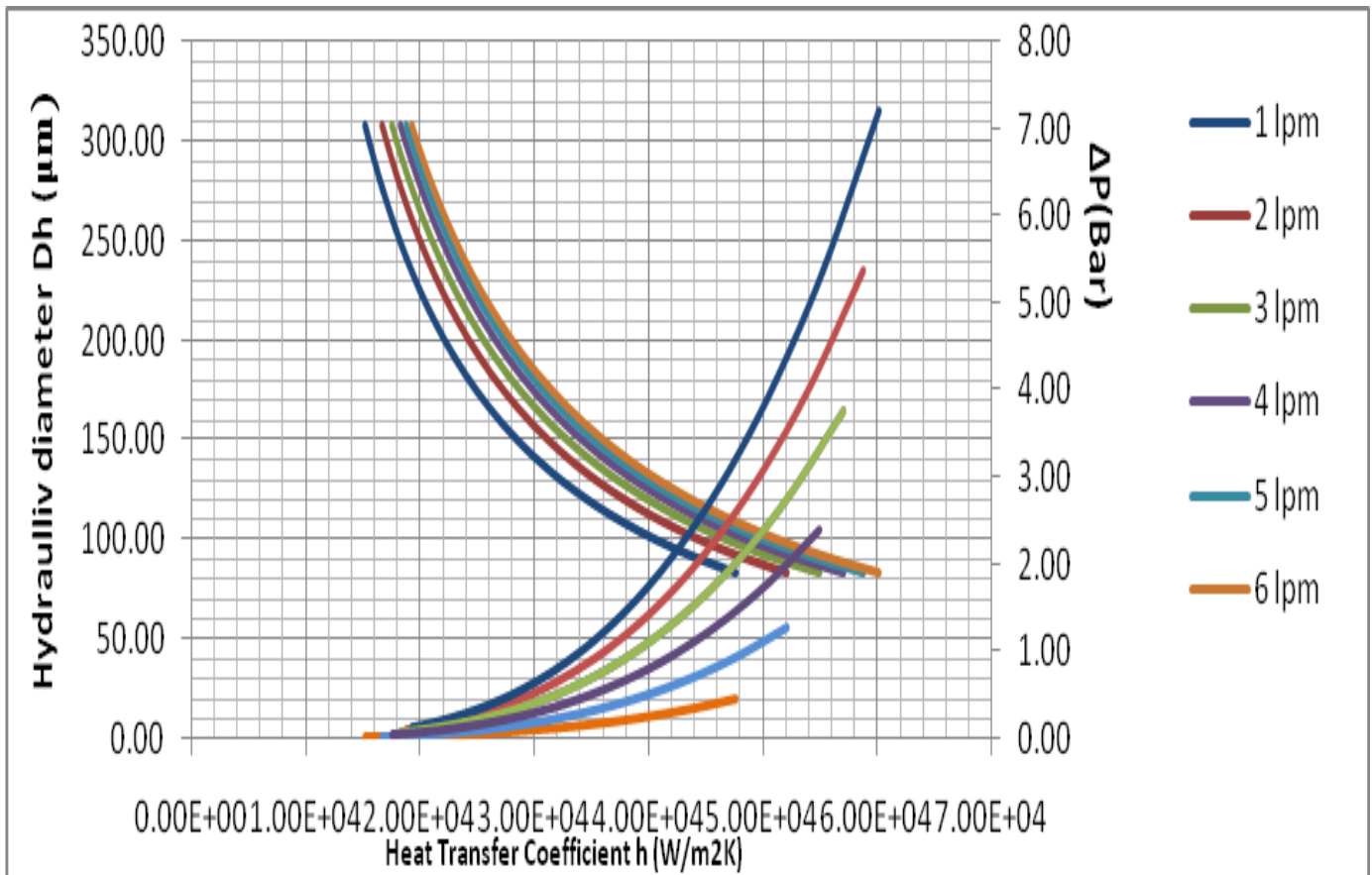


Figure 1: Hydraulic Diameter Vs Heat transfer coefficient plot

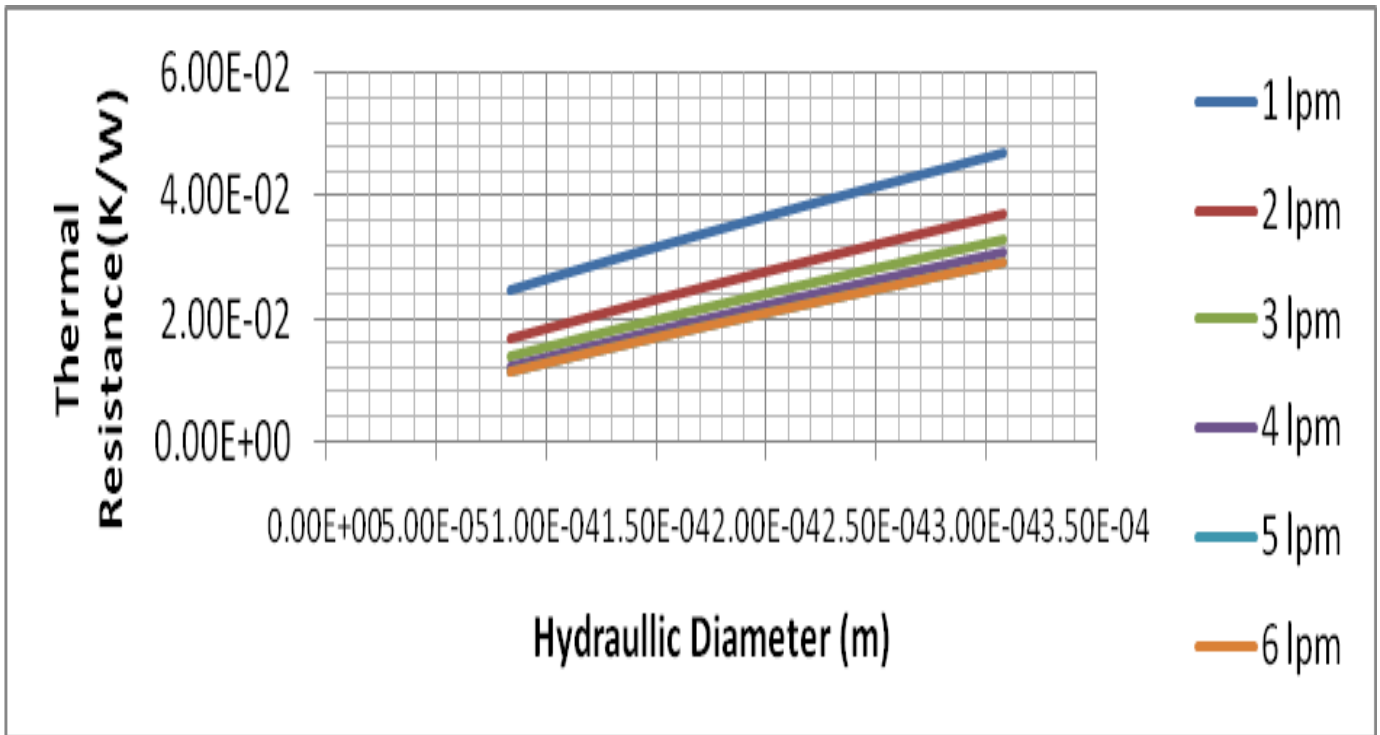


Figure 2: Thermal resistance Vs Hydraulic Diameter

**Estimation of Pressure Drop:** A secondary curve of pressure drop is plotted against the heat transfer coefficient. The value of pressure drop in the channels comes to be around 0.48 bar i.e. 6.96 psi.

**Estimation of Thermal Resistance:** Curve between the variations in the thermal resistance with the hydraulic diameter is plotted for the different flow rates. From the study of plots the thermal resistance comes to be 0.033 K/W for 4 lpm and 170µm hydraulic diameter.

VI. RESULTS AND DISCUSSION

Table 1: Results

S.No.	Wc(µm)	Ww(µm)	Z(µm)	Dh(µm)	h	f (lpm)	ΔP(bar)	R <sub>conv</sub>
1	80	80	400	133	31800	1	0.14	0.050
2	90	90	450	150	31700	2	0.2	0.043
3	95	95	475	158	31600	3	0.34	0.037
4	100	100	500	170	31500	4	0.48	0.033
5	105	105	525	175	31100	5	0.6	0.027
6	110	110	550	180	30800	6	0.74	0.024

Results obtained for aspect ratio 5. Now for an optimum design we will go with the 4 lpm flow rate. As discussed above, we know that as the aspect ratio increases the pressure drop and the thermal resistance decreases in the micro channels. Hence for the optimization in the same

parameters we will increase the value of the aspect ratio from 6 to 7 and study the behavior of the plot. There is negligible changes in heat transfer coefficient but a large drop in thermal resistance can be seen.

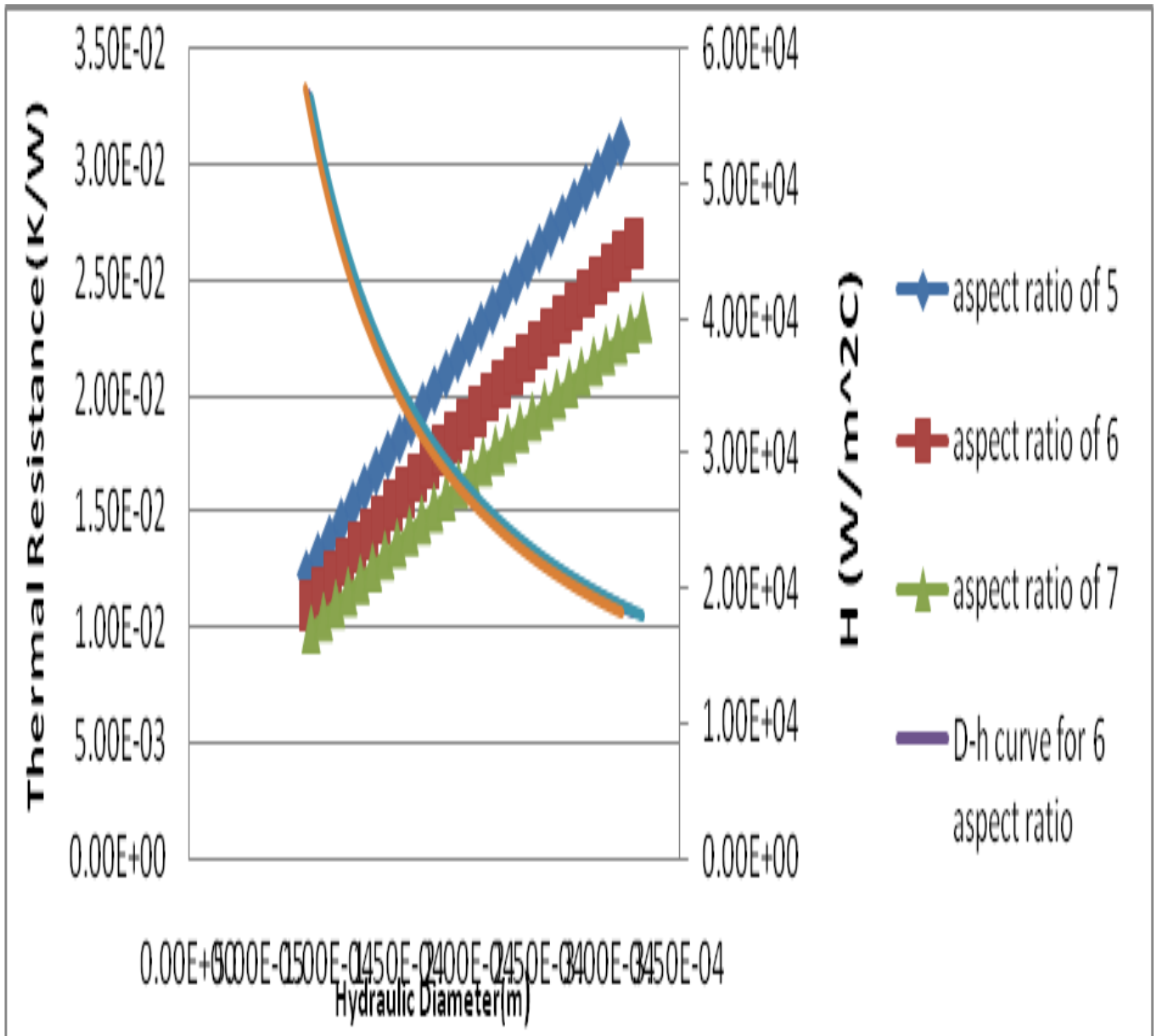


Figure3: Thermal resistance Vs Convective heat transfer Vs Hydraulic diameter

The value of heat transfer coefficient for aspect ratio 7 is somewhere slightly closer to 30000 hence we will opt for aspect ratio of 6 to be on safer side the value of heat transfer coefficient for 6 aspect ratio goes to 31200W/m<sup>2</sup>K but the value of pressure drop and the thermal resistance becomes 0.34 bar or 5 psi and 0.022 K/W. Hence the design is optimized and the values of Hydraulic diameter also increase to 175 μm.

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