Numerical Modeling and Optimization of a V-groove Warm Water Cold-plate

Yaser Hadad, Bharath Ramakrishnan, Sami Alkharabsheh, Paul R. Chiarot, Bahgat Sammakia
State University of New York at Binghamton, Binghamton, NY, USA, 13850
yhadad1@binghamton.edu

Abstract

In electronics cooling, water is increasingly replacing air for applications requiring high heat flux. Water is the ideal substitute due to its high specific heat capacity and density. Indeed, high values of heat capacity (high density and specific heat capacity) enable water to receive, store and carry higher amounts of energy compared to air. Water's incompressibility and very low specific volume also requires smaller amounts of mechanical work for fluid circulation. Using warm water instead of chilled water makes the cooling process more economical, but requires more efficiently designed cold-plates. Our current work focuses on modeling and optimization of a V-groove mini-channel cold-plate using warm water as the coolant. Our results show that the performance of an impinging channel heat sink is significantly different compared to parallel channel designs. Dividing the flow into two branches cuts the fluid velocity and flow path in half for the impinging design. This reduction in the fluid velocity and flow length affects the developing thermal boundary layer and is an important consideration for a shorter length heat exchanger (where the channel length is comparable to the thermal entrance length). Distributing the coolant uniformly to every channel is a challenge for impinging cold-plates where there are strict limitations on size.

Keywords

Impinging Channel, Cold-Plate, Inverse Graetz Number, Reynolds Number, Tilt Angle, Thermal Resistance, Pressure Drop

Nomenclature

A	Channel cross section area
A_s	Channel heat transfer contact area
b	Channel width
D_h	Hydraulic diameter $(4A/P)$
f	Friction factor
Gz	Graetz number
H	Channel height
l^+	Hydraulic dimensionless channel length
l_{ch}	Channel length
l_{th}	Thermal entrance length
P	Channel cross section perimeter
Pr	Prandtl Number
Q	Coolant flow rate
$\dot{q}$	Heat input the cold-plate
R_{th}	Thermal resistance
Re_{D_h}	Reynolds number based on hydraulic diameter
T_{in}	Coolant temperature at inlet
$\bar{T}_b$	Cold-plate average base temperature
$\bar{T}_{out}$	Coolant average temperature at exit
V	Coolant average velocity
α	Thermal diffusivity
θ	Fin tilt angle
μ	Coolant dynamic viscosity
ξ	Channel cross section aspect ratio (H/b)
ρ	Coolant density
ΔP	Pressure drop
Ξ	Asymptotic solution (Equation (8))

1. Introduction

A low heat transfer coefficient coupled with a low specific heat makes air a poor choice in micro-channel passages. Liquids, especially water, as compared to air or gases, offer a very good alternative due to their higher heat transfer coefficient, higher specific heat and lower specific volume [1].

Cold-plates with rectangular, parallel channels draw greater attention compared to other geometries. Li et al. [2] simulated forced convection heat transfer occurring in a silicon based microchannel numerically. Using their rectangular parallel-channel 3D model, they studied the influence of geometric parameters and thermo-physical properties of the fluid on the flow and heat transfer. A detailed description of the local and average heat transfer characteristics using a numerical three-dimensional analysis of flow and heat in a rectangular micro-channel heat sink was presented by Qu and Mudawar [3]. The effects of Reynolds number and thermal conductivity of the solid substrate on the heat transfer process were scrutinized. They found that the location of the highest temperature point is immediately above the channel outlet. Much higher heat flux and Nusselt number values were observed near the channel inlet. Xie et al. [4] analyzed a mini-channel heat sink numerically for single-phase laminar flow of water for small hydraulic diameters and a constant heat flux boundary condition. They presented the effect of channel dimensions, channel wall thickness, bottom thickness and inlet velocity on pressure drop, thermal resistance and the maximum allowable heat flux. Their results indicate that a narrow and deep channel with thin bottom thickness and relatively thin channel wall thickness results in improved heat transfer performance with relatively high but acceptable pressure drop. Copeland [5] studied the effect of fin thickness and pitch on the pressure drop and thermal resistance of a parallel channel air cooled heat sink. Using the theory of extended surfaces and empirical heat and flow correlations presented in Ref. [6], these two geometrical parameters were optimized.

Contrary to parallel channels, there are not many studies of liquid cooling for impinging channels. The advantages and disadvantages of replacing parallel heat sinks with impinging ones remains an open question. Zhuang et al. [7] studied the heat transfer characteristics under the impinging of transformer oil and FC-72 on two dimensional micro-channels. Local Nusselt numbers were obtained in stagnation and parallel flows. Duan and Muaychka [8] estimate the pressure drop through an impinging channel using analytical models for developing and fully-developed laminar boundary layers. The results were compared with experimental measurements and the accuracy of the pressure drop model was found to be within 20% of the experimental data. A numerical experiment and analytical scaling has been carried out to determine the pressure drop and thermal performance of single isothermal channels with variable width impinging flow by Biber [9]. They presented correlations for pressure loss coefficient and total heat transfer in a variable length channel with impinging flow.

The aim of this work is to develop numerical and analytical models for estimating the pressure drop and thermal performance of a manufactured impinging flow cold-plate. Using parallel channel numerical and analytical models, the effect of fins tilt angle on pressure drop thermal resistance is estimated.

2. Cold-Plate Geometry

The cold plate consists of two parts. The upper part is made of plastic and includes inlet and outlet elbows and ducts, as well as an inlet diffuser. The lower part is made of copper and possesses parallel channels connected by a V-groove (Figure 1).

Figure 1: Cold-plate with plastic and metal parts

In order to generate the geometry of the metal part for numerical modeling, micrographs were taken from different views. Figure 2 shows a micrograph taken at the exit of the channels showing the fins and their tilt angle.

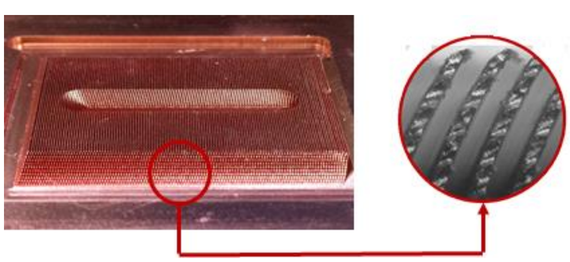

Figure 2: A micrograph of the channel network exit

The whole geometry of the cold-plate was divided to three parts (inlet, metal, and exit parts) in order to expedite the numerical modeling of the heat exchanger.

2.1. Inlet Part

The inlet part is the set of geometries that supplies the coolant to the micro-channels. The set consists of a circular cross section entrance, a curved diffuser (by 90°) with rectangular cross section and a duct that connects the entrance to the diffuser. Figure 3 shows the inlet geometry and the flow direction through them.

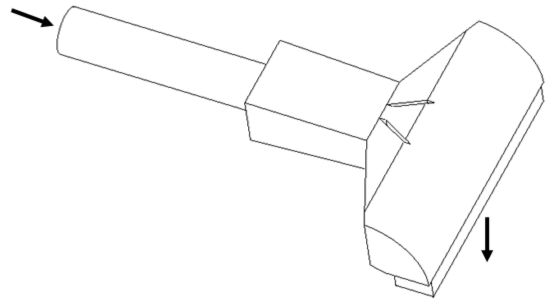

Figure 3: Geometry of the cold-plate inlet part

2.2. Metal Part

The metal part is copper and includes almost 200 impinging channels. Most of them are connected by a longitude V-shape cross section groove (Figure 2). Figure 4 shows a schematic of the metal part with associated flow direction and heat flux from the base.

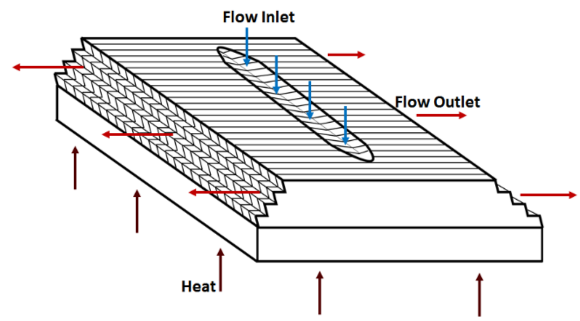

Figure 4: Flow and heat paths through the metal part

2.3. Outlet Part

In the outlet part, two ducts direct the flow from the exit of the channels to a miniature reservoir. Two subsequent rectangular and circular ducts direct the coolant from the reservoir to the exit hose (Figure 5).

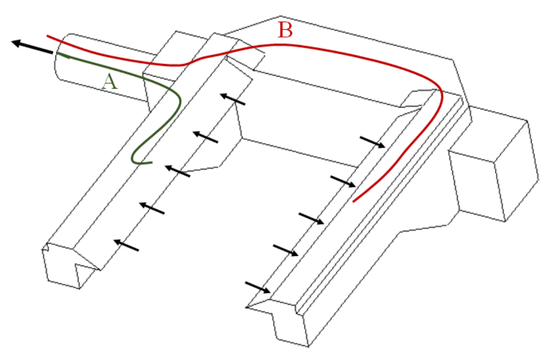

Figure 5: Outlet part of the cold-plate

3. Numerical Solution

In the case of the metal part, due to the periodic geometry and uniform heat flux boundary condition, a grid study was performed on one single channel and then extended to the whole domain. A structured grid with prismatic elements for the groove and hexahedral ones for the rest of the domain (fins, channels and base) was generated. Heat transfer consists of conduction in the base and fins and forced convection in the channels. The continuity, Navier-Stokes and energy equations were solved in the laminar regime ($Re_{D_h} < 200$) using the finite volume method with SIMPLE algorithm.

Unstructured grids were generated for inlet and outlet parts due to their more complex geometries. The energy equation was not solved for these parts because they do not participate in heat transfer. For high flow rates, the flow regime becomes locally turbulent ($Re_{D_h} > 2300$). Therefore, for these two parts, the standard k-ε turbulent model with standard wall functions was applied to find the solution.

4. Fins tilt angle

As shown in Figure 2, the fins (or channels) of the cold-plate are not vertical. For the current cold-plate, the tilt angle is about 65°. The effect of this angle on pressure drop and cooling performance was studied by developing a numerical model of a parallel channels cold-plate.

Micro or mini-channel cold-plates which are used for liquid cooling in electrical devices and other technologies are commonly recognized as short length heat exchangers ($l_{th} \sim l_{ch}$). In addition to the Reynolds number and heat transfer surface area, when designing and analyzing short heat exchangers, the Graetz number also plays an important role due to its strong effect on heat transfer coefficient, regardless of thermal boundary conditions (temperature or heat flux constant).

We consider fins with varying tilt angle, but constant height. Figure 6 shows how the cross section of a channel changes with tilt angle at constant height. The Inverse Graetz number can be calculated as follows:

$$Gz^{-1} = \frac{l_{ch}}{D_h Re_{D_h} Pr} \tag{1}$$

Where Re_{D_h} can be written as:

$$Re_{D_h} = \frac{2\rho Q}{\mu(b + \frac{H}{\sin(\theta)})} \tag{2}$$

Substituting Re_{D_h}, Pr, $D_h = 4A/P$ and $Q = VA$ in (1) we obtain:

$$Gz^{-1} = \frac{l_{ch}\alpha P^2}{16QA} \tag{3}$$

Replacing P and A versus θ, Gz^{-1} can be expressed as:

$$Gz^{-1} = \frac{l_{ch}\alpha}{4bHQ}(b + \frac{H}{\sin(\theta)})^2 \tag{4}$$

The heat transfer surface area also changes with tilt angle. If we assume that the upper wall of the channel is adiabatic (as it is, in most cold-plates), it can simply be expressed as:

$$A_s = l_{ch}(b + \frac{2H}{\sin(\theta)}) \tag{5}$$

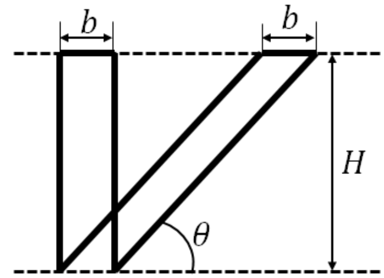

Figure 6: Change of channels cross section with tilt angle

In order to have an analytic estimation for the effect of tilt angle on pressure drop, a common asymptotic theoretical model developed by Muzychka and Yovanovich [10] for rectangular cross section channels is used. Here, their model is extended for channels with a parallelogram cross section. In their model, pressure drop can be calculated by:

$$\Delta P = \frac{1}{2}\rho f \frac{l_{ch}}{D_h} V^2 \tag{6}$$

where

$$f = \frac{\Xi}{Re_{D_h}} \tag{7}$$

the asymptotic solution for Ξ is:

$$\Xi = 4[\left(\frac{3.44}{\sqrt{l^+}}\right)^2 + (fRe_{D_h})^2]^{1/2} \tag{8}$$

the dimensionless hydraulic length l^+ is expressed as:

$$l^+ = \frac{l_{ch}}{D_h Re_{D_h}} \tag{9}$$

The fully developed term is expressed as the first term of analytical solution series [6]:

$$fRe_{D_h} = \frac{24}{[1 + (\frac{1}{\xi})]^2[1 - \frac{192}{\pi^5 \xi}\tanh\left(\frac{\pi\xi}{2}\right)]} \tag{10}$$

5. Results

Impinging cold-plates are expected to have some advantages compared to a parallel configuration. They will have a lower pressure drop through their channels due to fact that they slit the flow in two branches and cut the velocity and path length in half. However, distributing the flow uniformly to every channel is the main challenge of this kind of heat sink. In order to prevent this likely maldistribution, both the diffuser (inlet part) and collector should be properly designed. As can be seen in Figure 5, flow entering the left branch of the collector travels a shorter and more straight forward path (path B) to the exit compared to the right branch. In other words, flow along path A undergoes more friction and minor losses due to the collector asymmetry. Consequently, we calculate a higher coolant flow rate in the channels terminating at the left branch of the collector. This maldistribution is amplified by the inlet diffuser. Because of its curvature, the inlet diffuser makes the velocity profile asymmetric with its maximum nearer to the outer wall (Figure 7). Indeed, it pushes more flow to the channels which terminate to the left branch of the collector (Figure 5).

5.1. Pressure drop

Figure 8 shows the results of pressure drop through the cold-plate by matching the velocity profile at the exit of the diffuser with the inlet of the metal part (V-groove). The accuracy of the model is validated with experimental measurements of pressure drop.

The effect of tilt angle on the pressure drop is determined using the numerical parallel channel model. The analytical model mentioned in Section 4 (Equations (5)-(9)) is also

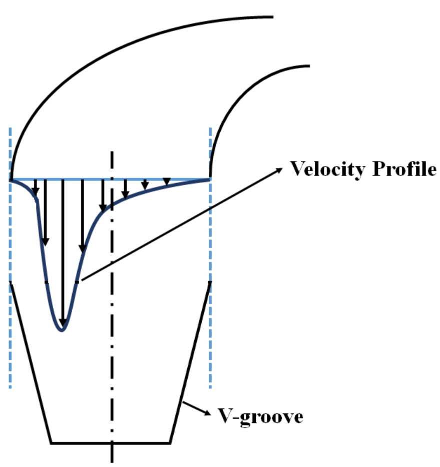

Figure 7: Schematic of diffuser flow maldistribution in 2D

extended for the parallelogram cross section channels and is used for validation. Figure 9 displays the results of pressure drop versus tilt angle. There is excellent agreement between the numerical and analytical model. Both predict that the pressure drop decreases for increasing tilt angle and reaches its minimum value at 90 degrees. It is worth mentioning that both numerical and analytical models used here consider the hydraulic entrance length automatically.

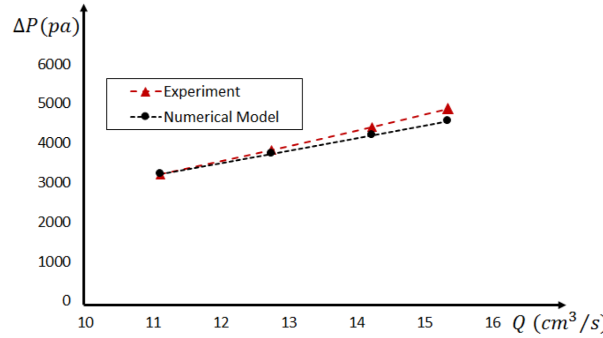

Figure 8: Pressure drop through the cold-plate

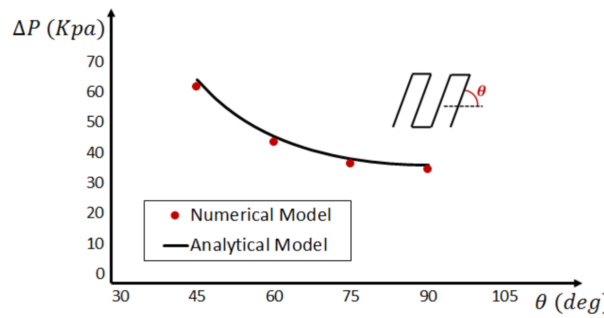

Figure 9: Effect of the tilt angle on the pressure drop

5.2. Thermal performance

The experimental and numerical results of outlet temperature are compared in Figure 10.

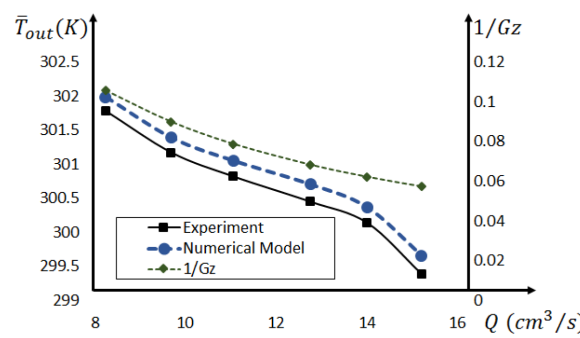

Figure 10: Average outlet temperature versus flow rate

As seen in Figure 10, there is a dramatic decrease in the outlet temperature at a threshold value of coolant flow rate ($Q = 14\,cm^3/s$) in both the numerical and experimental results. At this flow rate, the Inverse Graetz number approaches 0.06 and the thermal entrance length equals the channel length. At this point, the temperature at the center of the exit plane is equal to coolant inlet temperature. This region gets larger by increasing flow rate. It should be noted that this sudden drop cannot happen if we consider the mean temperature.

In order to study the effect of fins tilt angle on thermal performance, a numerical model of one channel with constant heat flux from the base and periodic boundary conditions on the side walls was developed. The thermal resistance is defined as:

$$R_{th} = \frac{\overline{T}_b - T_{in}}{\dot{q}} \tag{11}$$

It is calculated for different tilt angles and the results are shown in Figure 11.

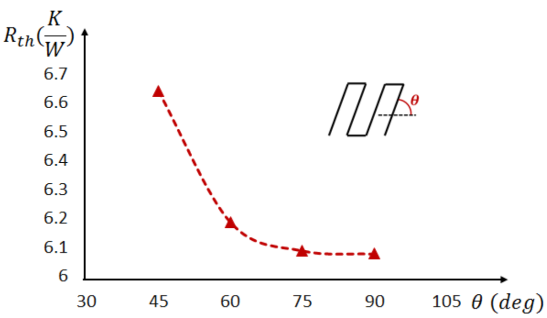

Figure 11: Effect of tilt angle on thermal resistance

As it is seen in Figure 11, similar to pressure drop, thermal resistance decreases with increasing tilt angle and is a minimum at 90 degrees.

Increasing the heat transfer surface area is one of the most effective ways for enhancing the thermal performance of heat exchangers. However, in the case of short heat exchangers, which are more common in electrical cooling, the inverse Graetz and Reynolds numbers determine the thermal performance.

Figure 12 depicts the variations of these three parameters with the tilt angle for a mini-channel with identified dimensions ($b = 0.2\ mm$, $H = 2mm$ and $l_{ch} = 20mm$) and constant water flow rate ($Q = 0.2\ cm^3/s$).

As opposed to thermally fully-developed flow, in the entry region the Nusselt numbers increases with decreasing inverse Graetz number. Therefore, according to Figure 12 (A) the maximum value of Nusselt number occurs at 90 degrees. The amount of heat transfers in a channel rises with increasing flow velocity and Reynolds number. As Figure 12(B) shows, the Reynolds number of the channel increases with increasing tilt angle and reach its maximum for vertical walls. For the case of heat transfer surface area in Figure 12 (C), it decreases with increasing tilt angle. For the short heat exchangers, the effect of Reynolds and inverse Graetz number dominant over the reduction of the contact surface area.

6. Conclusions

A numerical model of a water cooled cold-plate was developed to estimate pressure drop and thermal performance. It was found that the designs of the distributor and collector are very important. Inappropriate design of these parts leads to non-uniform distribution of coolant through the channels and ultimately causes high flow and thermal resistance.

The effect of tilt angle on pressure drop and thermal resistance was studied using numerical and analytical models. In the case of pressure drop, a developed analytical model for

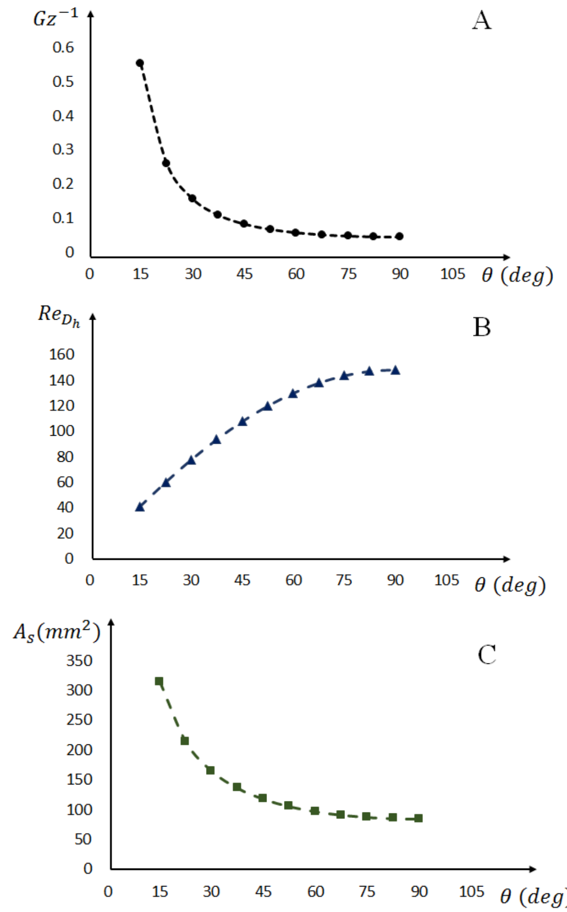

Figure 12: Variation of (A): Graetz number, (B): Reynolds number and (C): Heat transfer surface area with tilt angle

rectangular channels was extended for parallelogram-cross sections. Our results show that for thermal performance and pressure drop, channels with vertical walls are best.

In future work, we plan to optimize other geometrical parameters such as channel aspect ratio, thickness of fins, base thickness and groove cross section shape. The current cold-plate possesses an almost V-shape groove, but there is no reason for that to be the optimum. The results for impinging cold-plates can be compared with those of traditional parallel ones, provided that the models have similar geometric (channel width, fin thickness, fin height, etc.) and flow parameters (flow rate, pump power and pressure drop).

Acknowledgments

Authors would like to acknowledge M. Seymour (from Future Facilities), R. Tipton (from Emerson) and T. Craft (from CommScope) for their useful guidance through the course of this study.

References

1. Upadhye, H. R. and G. S. Kandikar, "Optimization of Micro-Channel Geometry for Direct Chip Cooling Using Single Phase Heat Transfer", ASME 2nd International Conference on Micro-Channels and Mini-Channels, pp. 670-685, 2004.

2. Li J., G.P. Peterson and P. Cheng, “Three-Dimensional Analysis of Heat Transfer in a Micro- Heat Sink with Single Phase Flow”, International Journal of Heat and Mass Transfer, Vol. 47, pp. 4215–4231, 2004.
3. Qu, W. and I. Mudawar, “Analysis of Three-Dimensional Heat Transfer in Micro-Channel Heat Sinks”, International Journal of Heat and Mass Transfer, Vol. 45, pp. 3973–3985, 2002.
4. Xie, X. L., Z. J. Liu, Y. L. He and W. Q. Hao, “ Numeical Study of Laminar Heat Transfer and Pressure Drop Characteristics in a Water-Coold Mini-Channel Heat Sink”, Applied Thermal Engineering, Vol. 29, pp. 64-74, 2009.
5. Copeland, D., “Optimization of Parralell Heat Sinks for Forced Convection”, 16th IEEE Semi-Therm Symposium, pp. 226-272, 2000.
6. Shah, K. R. and A. L. London, Lamiar Flow Forced Convection in Ducts, Advances in Heat Transfer, Supliment 1, Academic Press (1978).
7. Zhuang, Y., C. F. Ma and J. D. Dale, “Experimental Study on Local Heat Transfer with Liquid Impinging Flow in Two Dimensional Micro-Channels”, International Jouranl of Heat and Mass Transfer, Vol. 40, pp. 4055-4059, 1997.
8. Duan, Z., Y. S. Muaychka, “Pressure Drop of Impinging Air Cooled Plate Heat Sinks” Transactions of ASME, Vol. 129, pp. 190-194, 2007.
9. Biber, R. C., “Pressure Drop and Heat Transfer in an Isothermal Channel with Impinging Flow”, IEEE Transactions on Components, Packaging and Manufacturing Technology-Part. A, Vol. 20, No. 4, pp. 458-462, 1997.
10. Muzychka, Y. S., M. M. Yovanovich, “Pressuer Drop in Laminar Developing Flow in Noncircular Ducts: Scaling and Modeling Approach”, Jouranl of Fluid Engineering Transaction of ASME, Vol. 131, 111105 (11 pages), 2009.