The effectiveness of secondary channel on the performance of hybrid microchannel heat sink at low pumping power

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Abstract. High heat flux generated by advance miniature electronic devices with high power density is one of the factor that often adversely affects the device performance. Microchannels heat sink appears as a promising method which can removes the such heat flux. However, pressure drop generated by geometry in innovated MCHS consumes high pumping power in order to obtain the optimum overall performance of the MCHS. In this study, the combined effect of secondary channel with hybrid design (SD-RR-SC MCHS) has been analysed numerically. The result shows SD-RR-SC MCHS obtained highest overall performance over other designs for the low Re number (100 ≤ Re ≤ 450) and achieved PF of 1.65 at Re = 450. Besides that, the increases of pumping power consumption corresponding to CR MCHS for SD-RR-SC MCHS, SD-RR MCHS and RR MCHS at Re number of 450 are 71.2%, 101.7% and 351.4%, respectively. Means that, SD-RR-SC MCHS is suitable for an application which requires lower pumping power consumption. The existence of secondary channel geometry in SD-RR-SC MCHS has reduced the static pressure in its channel that contributes to the reduction of pumping power consumption. Furthermore, the design shows the most uniform velocity distribution which has contributed to the thermal performance enhancement.

1. Introduction
Over the past decade, the investigation of fluid flow and heat transfer characteristic induced by natural convection on thermal performance becomes a most interesting topic in a cooling system. The effectiveness of the cooling system in such application is very important to keep the temperature of a structure or electronic device from exceeding limits imposed by needs of safety and efficiency. In recent years, rapid growth in the electronic industry has witnessed a new generation high performing dense chip packages in many modern electronic devices. However, the development of more compact electronic devices that will operate at high power density causes the thermal management of electronic devices becomes a very critical issue in the electronics industry due to lack of efficient technique to remove heat from the devices [1].

During the past 30 years, many methods have been proposed in open literature in order to improve overall performance of microchannel heatsink. Generally, the methods can be categorized into two groups, active method and passive method. Most of researcher has widely used the passive method due to its low cost and absence of moving part compared to active method [2]. All of the studies reviewed in open literature demonstrates the heat transfer performance could be enhanced by using single technique of passive method [3]. However, pressure drop issue become the main constrain in the innovation of microchannel heatsink. Nowadays, many researchers have used multiple technique of passive method in single phase flow for enhancement of microchannel performance. In 2013, L. Gong
et al. [4] has analysed the performance of microchannel structured by dimple and wavy shape. They revealed that the presence of the dimple structure in wavy channel could enhance the heat transfer performance and not apparently increase the flow resistance. Li et al. [5] has presented a numerical study to investigate the combined effect of ribs and cavities on fluid flow and heat transfer characteristic in hybrid design MCHS. The analysis shows the design has obtained the optimum PF of 1.62 at Re = 500 with the friction factor ratio of 2.8 – 3.0. This characteristic is very important because the higher friction factor will require the higher pumping power due pressure drop produced by MCHS. After a year, Li and his research team [6] again study the performance of hybrid design with different geometry shape of cavities and ribs. The analysis shows all designs in their study obtained PF less than 1.45 for all Reynolds number. In 2017, Srivastava and his research team [7] try to investigate the combined effect of ribs and cavities in convergent-divergent hybrid microchannel. The result shows the hybrid design obtained the lower thermal resistance compare to hybrid design of rectangular channel with ribs and cavities. Pressure drop produced by the convergent-divergent hybrid microchannel is around around 7500 kPa – 10000 kPa for Re number of 400 to 500. Ghani et al. [8] studied the combined effect of ribs and secondary channels on hydrothermal performance in hybrid MCHS. The hybrid design obtained the PF of 1.9 – 2.0 with the friction factor of 6.0 – 7.0.

Most of hybrid designs presented in open literature shows the optimum overall performance of MCHS was obtained at high Re number which has consumed high pumping power. Hence, research gap existed since there has no researcher that study about the optimum overall performance enhancement at low Re number which contributes to reduction of pumping power consumption. The novelty of this research is to find the optimum hybrid design microchannel heat sink with secondary channel which can provide high heat transfer rate with low pumping power consumption. In this study, the effectiveness of secondary channel geometry on the performance of hybrid microchannel heat sink and pumping power consumption were numerically studied by comparing the fluid flow and heat transfer characteristic in each microchannel heat sink designs. In present study, we extend our previous work [9] by introducing secondary channels geometry in current design (SD-RR MCHS) in order to enhance the overall performance of the MCHS at low Re number with low pumping power consumption. Comparative analysis between proposed design (SD-
RR-SC MCHS) with related geometry such as CR MCHS, RR MCHS, SD MCHS and SD-RR MCHS was conducted in order to analyse the performance enhancement in the proposed design.

2. Geometry parameter of microchannel heat sink
This microchannel heat sink is made by copper and consist of ten microchannels. However, in order to save the computational cost, only one symmetrical part of the microchannel heat sink is adopted in present simulation as shown in Figure 1(a). Table 1 shows parameter valued for the geometry that illustrated in Figure 1(b). In order to study the effectiveness of secondary flow on hybrid microchannel heat sink performance, design development is started from reference design (CR MCHS) to single passive technique design (RR MCHS and SD MCHS). Next, hybrid design (SD-RR and SD-RR-SC) is developed based on the strength and weakness that found in single passive technique designs.

![Figure 1](image-url)

**Table 1.** Geometry parameters of SD-RR-SC MCHS.

| Lt (µm) | Wt & Hc (µm) | Ht (µm) | Wc (µm) | Lr & L3 (µm) | Wr (µm) | L1, L2 & L5 (µm) | L4 (µm) | Wsc (µm) | Asc (°) | Asd (µm) |
|---------|--------------|---------|---------|-------------|---------|-----------------|--------|----------|--------|---------|
| 10000   | 300          | 400     | 150     | 250         | 45      | 500             | 75     | 40       | 15     | 75      |

3. Numerical method approach
In order to analyse the performance of proposed design (SD-RR-SC MCHS), a Computational Fluid Dynamic (CFD) software such as ANSYS FLUENT 17.0 is used to solve the three dimensional fluid flow and heat transfer equations by assuming fluid flow in all simulated designs are continuum due to Knudsen number, \((Kn)\) for the fluid flow is less than \((10^{-3})\) [10]. So, Navier-Stokes equation and non-slip boundary condition are applicable. Besides that, the fluid is assuming as Newtonian, incompressible and has a constant thermophysical properties. The laminar fluid flow and heat transfer are simulated in steady-state. Viscous dissipation, gravitational force and radiation heat transfer are neglected.

3.1. Governing equations
Based on the assumptions that made in the present study, governing equation for conservation of mass, momentum and energy equations can be written as equation (1), equation (2) – (4) and equation (5) – (6), respectively.

\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0
\]

(1)

Where \(u\), \(v\) and \(w\) are the velocity components in \(x, y\) and \(z\)-directions respectively. Momentum equation is written as:

\[
u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} = -\frac{1}{\rho_f} \frac{\partial p}{\partial x} + \frac{\nu}{\rho_f} \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right)
\]

(2)
\[
\begin{align*}
\frac{\partial \mathbf{v}}{\partial t} + (\mathbf{v} \cdot \nabla) \mathbf{v} &= -\frac{1}{\rho_f} \nabla p + \frac{\mu_f}{\rho_f} \left( \frac{\partial^2 \mathbf{v}}{\partial x^2} + \frac{\partial^2 \mathbf{v}}{\partial y^2} + \frac{\partial^2 \mathbf{v}}{\partial z^2} \right) \\
\frac{\partial w}{\partial t} + (\mathbf{v} \cdot \nabla) w &= -\frac{1}{\rho_f} \nabla p + \frac{\mu_f}{\rho_f} \left( \frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right)
\end{align*}
\]

Where \(\rho_f\) and \(\mu_f\) are the density and dynamic viscosity of the working fluid, respectively, and \(p\) is the fluid pressure. There are two energy equations that related to the present study such as energy equation for fluid region, equation (5) and energy equation for solid region, equation (6):

\[
\begin{align*}
\frac{\partial T_f}{\partial t} + \mathbf{v} \cdot \nabla T_f &= \frac{k_f}{\rho_f c_p} \left( \frac{\partial^2 T_f}{\partial x^2} + \frac{\partial^2 T_f}{\partial y^2} + \frac{\partial^2 T_f}{\partial z^2} \right) \\
\frac{\partial T_s}{\partial t} + \mathbf{v} \cdot \nabla T_s &= -k_s \left( \frac{\partial T_s}{\partial n} \right) = -k_f \left( \frac{\partial T_f}{\partial n} \right)
\end{align*}
\]

Where \(T_f, T_s, k_f, k_s\) and \(C_p_f\) are the fluid’s temperature, solid’s temperature, fluid thermal conductivity, solid thermal conductivity and fluid specific heat, respectively.

3.2. Boundary Condition

Boundary condition is a condition for hydrodynamic and thermal that we applied on the simulated geometries in the present study. Table 2 shows the details for the boundary conditions.

**Table 2.** Boundary condition.

| Boundary       | Location | Condition                                      |
|----------------|----------|-----------------------------------------------|
| Hydrodynamic   |          | No-slip and no penetration                     |
|                |          | \( u = v = w = 0 \)                           |
| At the fluid-solid interface |          | \(-k_f \left( \frac{\partial T_f}{\partial n} \right) = -k_f \left( \frac{\partial T_f}{\partial n} \right)\) |
|                |          | \( u_f = u_{in} \) (uniform velocity)         |
| At inlet,      | \( x = 0 \) | \( v = w = 0 \)                              |
| At outlet,     | \( x = L_t = 10 \text{mm} \) | \( p_f = p_{out} = 1 \text{atm} \) |
| Thermal        |          | \( T_f = T_{in} = 300 \text{K} \) (for water) |
|                | \( x = 0 \) | \(-k_f \left( \frac{\partial T_f}{\partial x} \right) = 0 \) (for solid) |
| At outlet,     | \( x = L_t = 10 \text{mm} \) | \(-k_f \left( \frac{\partial T_f}{\partial x} \right) = 0 \) (for solid) |
| At top wall,   | \( z = Ht = 0.4 \text{mm} \) | \( u = v = w = 0 \); \(-k_s \left( \frac{\partial T_s}{\partial z} \right) = 0 \) |
| At bottom wall,| \( z = 0 \) | \(-k_s \left( \frac{\partial T_s}{\partial z} \right) = q = 100 \text{W/cm}^2 \) |
At side wall,
\[ y = 0 \]
\[ \frac{\partial}{\partial y} = 0 \text{ (symmetry)} \]

At side wall,
\[ y = Wt = 0.3\text{mm} \]
\[ \frac{\partial}{\partial y} = 0 \text{ (symmetry)} \]

3.3. Grid independence and CFD simulation

Mesh structure quality and number of grid are very important element that will contribute to the convergence of numerical solution and numerical computation stability. In this paper, ANSYS ICEM is used to generate hexahedral meshing structure due to faster solution time with better accuracy than tetrahedral meshing structure. Figure 2 illustrates the hexahedral mesh structure in SD-RR-SC design. Finite volume method is utilized to discretize the governing equation. The SIMPLE algorithm was adopted to accomplish the pressure-velocity coupling. At the same time, the second order upwind scheme is used for convective term and second order central difference scheme is applied for diffusion term. Furthermore, convergence criterions are set to be less than 10^{-6} for continuity and less than 10^{-9} for energy.

![Figure 2. Computational grid of SD-RR-SC MCHS: a) Isometric view, b) Top view at x-z plane.](image)

Grid independence test is a process to find the optimum mesh structure for all designs which can produce accurate result with the lower time cost. The process is started with a fine mesh structure to a coarse mesh structure. In the present study, the total number of element or grid for the fine mesh is 1.5 million for Reynolds number of 800. With the same Reynolds number, the number of element is reduced up to 0.7 million. By considering Nusselt number and pressure drop for each element number, the optimum mesh structure is obtained by calculating the relative error using following equation:

\[
e^{\%} = \left| \frac{J_2 - J_1}{J_1} \right| \times 100 \quad (7)
\]

Where \( J_1 \) represents the value of Nusselt number and pressure drop for the finest mesh structure while \( J_2 \) represents the value of Nusselt number and pressure drop for other element number. Based on the relative error presented in Table 3 for each element number, it clearly observed that element number of 1.3 million has a reasonable accuracy compared with other element numbers. Consequently, the number element of 1.3 million is selected for all cases in current study.
### Table 3. Grid independence test.

| Grid number (X10^6) | Nu   | e%  | Pressure drop (Pa) | e%  |
|--------------------|------|-----|---------------------|-----|
| 1.5                | 10.23| -   | 41572.54            | -   |
| 1.3                | 10.24| 0.10| 41524.38            | 0.12|
| 1.1                | 10.24| 0.10| 41482.03            | 0.22|
| 0.9                | 10.25| 0.20| 41394.09            | 0.43|
| 0.7                | 10.26| 0.29| 41189.63            | 0.92|

3.4. Data reduction

This section presents the relevant expressions that used to calculate the characteristics of heat transfer and fluid flow in MCHS. Re number, hydraulic diameter and apparent friction factor are expressed in equation (8), equation (9) and equation (10), respectively.

\[
Re = \frac{\rho u_m D_h}{\mu} \tag{8}
\]

\[
D_h = \frac{2H W_c}{H_e + W_c} \tag{9}
\]

\[
f_{app,ave} = \frac{2D_h \Delta P}{L t \rho u_m^2} \tag{10}
\]

Where \( D_h \), \( L_t \) and \( \Delta P \) are hydraulic diameter, total length of microchannel and pressure drop across microchannel, respectively. Average heat transfer coefficient and the average Nusselt number are given by:

\[
h_{ave} = \frac{q_w A_{film}}{A_{cond}(T_{w,ave} - T_{f,ave})} \tag{11}
\]

\[
Nu_{ave} = \frac{h_{ave} D_h}{k_f} \tag{12}
\]

Where \( q_w, A_{film}, A_{cond}, T_{w,ave} \) and \( T_{f,ave} \) are the heat flux per unit area, heated area, convection heat transfer area, average temperature of wall and average temperature of fluid, respectively.

4. Result and discussion

4.1. Validation of conventional rectangular design (CR MCHS) based on the previous correlation predictions

In order to verify the accuracy of the simulation model approach, data that obtained from simulation analysis such as local Nusselt number and pressure drop for reference design (CR MCHS) are validated with the previous correlation developed by former researcher. The local Nusselt number is validated with Philips [11] correlation as shown below:

\[
Nu_x = 1.0958 \left[ \frac{23.315 + 27038(X) + 1783300(X)^2}{1 + 3049(X) + 472520(X)^2 - 35714(X)^4} \right] \tag{13}
\]

\[
X = \frac{x}{D_h Re Pr} \tag{14}
\]

Pressure drop is validated with Steinke & Kandlikar [12]:

...
\[ \Delta p = \frac{2(f \, \text{Re})\mu_m L_t}{D_b^2} + \frac{K \rho_f u_m^2}{2} \]  
\[ f \, \text{Re} = 24\left(1 - 1.3553\alpha_c + 1.9467\alpha_c^2 + 1.7012\alpha_c^3 + 0.9564\alpha_c^4 + 0.2537\alpha_c^5\right) \]  
\[ K = 0.6797 + 1.2197\alpha_c + 3.3089\alpha_c^2 - 9.5921\alpha_c^3 + 8.9089\alpha_c^4 - 2.9959\alpha_c^5 \]  

Where \( \alpha_c \) is channel aspect ratio of width to height and \( K \) is Hagenbach’s factor. Validation of the simulation model with the correlation has been illustrated in Figure 3. It clearly shows the simulation result has a good agreement with the correlation. This agreement indicates that the simulation model approach can be adopted to predict the fluid flow and heat transfer characteristic in present study for all designs.

\[ \text{Figure 3. Model validation: a) Local Nusselt number according to Philips [11], b) Pressure drop according to Steinke & Kandlikar [12].} \]

4.2. Fluid flow and heat transfer analysis in hybrid design with secondary channel (SD-RR-SC MCHS)

4.2.1. Velocity distribution analysis. Velocity gradient developed in laminar flow is one of the factors that create thermal resistance along microchannel heat sink. So, uniformity of fluid velocity will give a significant effect on the thermal resistance. In order to provide more uniform velocity distribution along microchannel, rib geometry was introduced in RR MCHS, SD-RR MCHS and SD-RR-SC MCHS. Figure 4 illustrates the effect of ribs geometry on velocity profile in those microchannel heat sinks. The figure shows rib geometry have guided flow at the centre portion of channel which has a high velocity to a side wall which has a low velocity due to friction on the side wall. For designs without ribs geometry (CR MCHS and SD MCHS), it clearly shown there has no redevelopment of boundary layer thickness in the fluid flow velocity profile. Besides, SD MCHS created the hot spot in cavity area due to vortexes flow in stagnation zone. When the Re number increased from 200 to 450, SD MCHS design shows the enlargement of vortexes flow in cavities. This condition will increase the hot spot area. For RR MCHS, maximum velocity focused on the sidewalls of rib. By comparing all designs, SD-RR MCHS and SD-RR-SC show the most
uniform velocity distributions compare to other designs. If we compare between these two designs, SD-RR-SC MCHS has a better velocity uniformity than SD-RR MCHS at the central portion of channels. Secondary flow provided by secondary channel in SD-RR-SC MCHS helps to create disturbance at the main flow and thus provide uniform velocity in the main channel.

![Velocity distribution contour and streamlines](image)

**Figure 4.** Velocity (m/s) distribution contours and streamlines at x-z plane (y = 0.25mm): a) At Re = 200, b) At Re = 450.

4.2.2. Pressure distribution analysis. This analysis was conducted in order to study the effect of geometry design on pressure drop created in channels. High pressure drop will consume high pumping power to flow fluid in the channels. According to Figure 5, SD MCHS obtained the lowest static pressure due to no ribs geometry at the centre portion of channels and has a large flow area in cavities. However, this design is not the best for thermal performance due to high thermal resistance created by boundary layer thickness. The highest static pressure is shown by RR MCHS due to flow-blocking effect in the small area of straight channel provided by ribs geometry. As consequence, the design has produced the highest pressure drop and dominates the thermal performance enhancement obtained by the design. However, the ability of ribs geometry to redevelop boundary layer has contributed to the overall performance enhancement of SD-RR MCHS and SD-RR-SC MCHS due to the availability of larger flow area provided by cavities. Surprisingly, the existence of secondary channel geometry in SD-RR-SC MCHS obtained the lower static pressure than SD-RR MCHS as illustrated in Figure 5 at location x = 5.5mm in cavities area. From this analysis we can see that, combination of secondary channel geometry in hybrid design (SD-RR-SC) has contributed to the reduction of pressure drop by providing the lower static pressure along MCHS.
4.2.3. Temperature distribution analysis. Temperature distribution analysis was conducted in order to illustrate the performance of heat transfer among each designs. Figure 6 illustrates temperature distribution from a top view at x-z plane (y=0.25mm) for Re number of 450. The figure shows a conventional design, CR MCHS shows the highest temperature distribution compare to other designs. For an enhanced MCHS, SD MCHS shows the highest temperature distribution compared with RR MCHS, SD-RR MCHS and SD-RR-SC MCHS. As we can see, even temperature distribution profile of SD MCHS quite similar with CR MCHS which has two distinct region of temperature (low-temperature region at the central portion and high-temperature region at the sidewalls), SD MCHS shows the lower temperature distribution than CR design due to heat transfer area provided by SD MCHS is greater than CR MCHS. However, heat transfer performance in SD MCHS is not good as other enhanced MCHS due to vertexes flow in stagnation zones in cavities of SD MCHS increased fluid temperature on the hot spot area. However, combination of cavities and ribs geometries in SD-RR MCHS and SD-RR-SC MCHS can eliminate the vertex flow in the stagnation zone experienced by SD MCHS. Both designs show the lower temperature distribution than CR MCHS, SD MCHS and RR MCHS. By comparing SD-RR MCHS and SD-RR-SC MCHS at location x = 4.5mm, the existence of secondary channel reduces the temperature distribution on the channel walls of SD-RR-SC MCHS more than SD-RR MCHS. Flow disturbance and flow mixing provided by ribs and secondary channels geometry leads to the thermal performance enhancement in SD-RR-SC MCHS.
4.2.4. **Performance Evaluation Characteristic (PEC) analysis.** Figure 7 and Figure 8 show the variation of Friction factor ratio \((f / f_o)\) and Nusselt number ratio \((Nu / Nu_o)\) for all designs corresponding to CR design, respectively. The Subscript \((o)\) is refer to parameter of reference geometry (CR MCHS). The lowest friction factor ratio and the highest Nusselt number ratio will contribute to the overall performance enhancement in MCHS. Figure 7 shows SD MCHS obtained the lowest friction factor ratio while the highest friction factor ratio observed in RR MCHS. Surface roughness method that implemented in SD MCHS by using cavity geometry has reduced the friction losses in its channels. For the hybrid designs, SD-RR-SC MCHS obtained the lower friction factor ratio than SD-RR MCHS due to extended of flow area provided by secondary channel geometry. Generally, a single passive technique will have a drawback behind its performance. As shown in Figure 8, even the friction factor ratio of SD MCHS is the lowest one, the design shows the lowest Nusselt number ratio which will affect the overall performance enhancement. For RR MCHS, even the geometry shows the highest friction factor ratio, the design obtained the highest Nusselt number. However, due to pressure drop and pumping power consumption, the RR MCHS is not a good design for enhanced performance MCHS. For hybrid designs MCHS, SD-RR-SC MCHS obtained the higher Nusselt number ratio than SD-RR MCHS for the Re number less than 400. However, at Re ≥ 400, SD-RR MCHS dominates the Nusselt number ratio obtained by SD-RR-SC MCHS.

**Figure 7.** Variation of Friction factor ratio with Reynold number for all designs corresponding to CR design.

**Figure 8.** Variation of Nusselt number ratio with Reynold number for all designs corresponding to CR design.

Performance factor \(PF\) that written in equation (18) is used as indicator for the enhancement of overall performance. If the \(PF\) value is greater than 1, means the enhancement of heat transfer performance obtained by the enhanced MCHS is within acceptable pressure drop. If the \(PF\) value is less than 1, means the enhancement of pressure drop in the enhanced MCHS dominates the augmentation of heat transfer performance. Figure 9 shows hybrid designs (SD-RR MCHS and SD-RR-SC MCHS) obtained the highest performance factor than SD MCHS and RR MCHS. At the lower Re number (Re ≤ 450), SD-RR-SC MCHS shows a superior overall performance over all designs in present study. However, at the higher Re number (Re > 450), performance factor of SD-RR MCHS dominates the overall performance obtained by SD-RR-SC MCHS.

In order to meet the objective of this paper and industrial pressure limit for low Re number (100≤Re≤400) [13], pumping power consumption is considered in the design development. Equation (19) is used to calculate the pumping power consumption. Figure 10 shows the pumping power consumption for two channels of each design. It clearly observed the SD-RR-SC MCHS shows the lower pumping power consumption than SD-RR MCHS for all Re number due to lower friction factor obtained in the channels of SD-RR-SC MCHS. In general, SD-RR-SC MCHS shows a superior performance factor over other designs at low Re number (100≤Re≤450) with less pumping power consumption than SD-RR MCHS and RR MCHS. The increases of pumping power consumption corresponding to CR MCHS for SD-RR-SC MCHS, SD-RR MCHS and RR MCHS at Re number of 450 are 71.2%, 101.7%.
and 351.4%, respectively. Means that, Secondary channel geometry in hybrid design (SD-RR-SC MCHS) is suitable for the application which requires a low pumping power consumption.

\[ Pf = \frac{Nu/Nu_0}{(f/f_0)^{0.5}} \]  
\[ P_{pump} = nu_c A \Delta p \]  

5. Conclusion
Objective of this research to study the effectiveness of secondary channel geometry on hybrid microchannel heat sink (SD-RR-SC MCHS) performance and pumping power consumption. Comparative analysis between proposed design (SD-RR-SC MCHS) with relevant geometry (SD MCHS, RR MCHS and SD-RR MCHS) was conducted in order to analyse the significant effect of each geometry on fluid flow and heat transfer characteristic. From the analysis and results, the following conclusions appeared:

1) SD-RR-SC MCHS shows the highest performance factor over other designs at the low Re number (100≤Re≤450) with less pumping power consumption than SD-RR MCHS and RR MCHS. The increases of pumping power consumption corresponding to CR MCHS for SD-RR-SC MCHS, SD-RR MCHS and RR MCHS at Re number of 450 are 71.2%, 101.7% and 351.4%, respectively. Means that, Secondary channel geometry in hybrid design (SD-RR-SC MCHS) is suitable for the application which requires a lower pumping power consumption.
2) The highest performance factor obtained by SD-RR-SC MCHS is 1.62 at Re number of 450 (For the range of 100≤Re≤450)
3) SD-RR-SC MCHS shows the most uniform velocity distribution profile compare to other design. This characteristic has contributed to the heat transfer performance in microchannels. The existence of secondary channel has reduced a residence time of fluid in cavities which experienced by SD-RR MCHS.
4) The existence of secondary channel in SD-RR-SC MCHS has reduced static pressure in its channel which has contributed to the reduction of pressure drop and thus reduces pumping power consumption.

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