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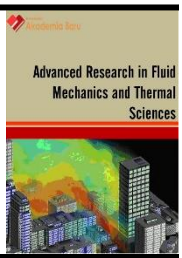

Numerical Study of Heat Transfer in A Microchannel Heat Sink with Hourglass Channel Profile

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Kok Cheong Wong[1,*], Yih Loong Chan[1], Boon Hoe Goh[2]

[1] Department of Mechanical, Materials and Manufacturing Engineering, Faculty of Engineering and Science, University of Nottingham Malaysia, 43500 Semenyih, Selangor, Malaysia
[2] Department of Civil Engineering, Faculty of Engineering and Science, University of Nottingham Malaysia, 43500 Semenyih, Selangor, Malaysia

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ABSTRACT

The present paper reports the numerical investigation of thermal and hydraulic performance of microchannel heat sink with hourglass profile (MCHS-HG). A 3-D conjugate heat transfer model was used to solve the problem with water as coolant and silicon as material of the heat sink. Performances of MCHS-HG were evaluated with 5 different contraction ratios, Rc (0.2, 0.4, 0.6, 0.8 and 1.0) and 3 different aspect ratios from 4, 5 and 6 with Reynolds number in the range of 300-1800. Thermal performance of MCHS-HG was compared with microchannel heat sink with straight rectangular channel (MCHS-R). Results showed that thermal performance of MCHS-HG is better than MCHS-R for any given aspect ratio. The study on the effect of contraction ratio show that the optimum thermal performance is given by the contraction ratio of 0.4. Microchannel with narrower throat of hourglass profile gives higher pumping power. The assessment on thermal-hydraulic performance of MCHS-HG shows that the MCHS-HG is less economical to be applied as compare to MCHS-R.

Keywords:
Microchannel Heat Sink; Hydraulic Performance; Thermal Performance; hourglass profile; contraction ratio

1. Introduction

Electronic devices are getting smaller in size in this few decades and it is expected to become even smaller in the future. One of the challenges in miniaturized electronic device is heats dissipation of the device as it operates under higher power density. The miniaturized microchannel heat sink (MCHS) was first introduced by Tuckerman and Pease [1] in 1981 which the microchannels were etched on the upper surface of a silicon chip. The idea of using microscopic heat sink was suggested by them and experiments were performed to evaluate its performance. They stated that the best way to significantly increase the heat transfer coefficient is to reduce the hydraulic diameter of the microchannel. In their experiments, lowest thermal resistance was recorded at 0.09 °C/W with heat

[*] *Corresponding author.*
E-mail address: Kok-Cheong.Wong@nottingham.edu.my

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flux of 790 W/cm^2. They also found that MCHS will be capable to dissipate 1300 W/cm^2 heat flux with temperature difference of 50°C. Qu and Mudawar [2] conducted both experimental and numerical study on pressure drop and heat transfer of a MCHS made of copper. They obtained good agreement between experimental and numerical results and concluded that conventional Navier–Stokes and energy equations can adequately predict the fluid flow and heat transfer characteristics of microchannel heat sinks.

Zhai *et al.,* [3] performed numerical and experimental work to validate previous analytical model used for predicting performance of microchannel heat sink with straight rectangular channel. An accuracy within 10% difference between analytical and empirical results was reported by them. Their results also showed that convective thermal resistance had a large effect on thermal performance of microchannel heat sink compared with capacitive and conductive thermal resistances. They also stated that vicious dissipation can be neglected if water is used as coolant. Research on heat transfer of microchannel heat sink has received considerable attention, including the study of substrate materials of MCHS such as ceramic [4] and the studies of microchannel geometrical effects [5-11].

Raghuraman *et al.,* [5] studied numerically the thermal performance of different channel aspect ratio of the channel in a MCHS-R. Aspect ratio of 20, 30 and 46.66 over a range of Reynolds number were investigated. They found that pressure drop was the highest with AR 20. On the other hand, pumping power was highest with AR 46.66 due to high flow rate. AR 20 also had a higher Nusselt number as compared with ARs of 30 and 46.66. MCHS with AR 30 was found to be ideal among those investigated as it gave highest heat transfer rate with pumping power maintained at optimum level.

There were many interests in investigating thermal performance of MCHS with channel geometries other than rectangular shape. Wang *et al.,* [6] reported MCHS-R had lowest thermal resistance, followed by trapezoidal and triangular MCHS when number of channels, cross section area and hydraulic diameter were the same. MCHS-R gave the best performance when AR is in the range of 8.9-11.4. On the other hand, Tran *et al.,* [7] investigated numerically studied the thermal performance of MCHS with 5 different channel geometries i.e. circle, square, trapezium, concave, convex. The studies were conducted in the range of flow Reynolds number 700–2200 with hydraulic diameter of 200 microns. They reported that the best thermal performance was achieved by the geometry of circular channel which was able to dissipate up to 1500 W/cm^2 at Reynolds number of 2200 while square channel is the poorest among the 5 different geometries.

Attempts were made to enhance the heat transfer with different structures of MCHS. Examples are double layer MCHS by Vafai and Zhu [8] and MCHS with interrupted microchannels and triangular ribs by Wong and Lee [9]. Both studies showed positive results as compared to MCHS-R. The former has an increase in heat transfer surface area and the latter has a better mixing of flow and increase in heat transfer area that promote heat transfer rate. Both Zhou *et al.,* [10] and Sakanova *et al.,* [11] studied the thermal hydraulic performance of MCHS with wavy channel. Results in both studies showed an increase in heat transfer rate with as high as 2.8 times as compared to straight microchannel. In addition, Sakanova *et al.,* [11] also studied the effect of nanofluid in MCHS. Positive results were obtained in their study. They concluded that diamond-water nanofluid exhibits the best thermal performance with lowest thermal resistance.

Ghani *et al.,* [12] investigated numerically MCHS with rectangular ribs, sinusoidal cavity and combined features of both in the range of Reynolds number 100-800. They obtained results that shows the MCHS with combined features is superior over individual feature as the combined features able to eliminate stagnation and promote flow mixing. The combination effectively reducing the pressure drop through providing larger flow area compare with microchannel with rectangular ribs.

Yang *et al.,* [13] and Yu *et al.,* [14] had focused their studies in a pin-finned microchannel heat sink. The former studied five different pin fin configurations while latter focused in microchannel with piranha pin fin (PPF). They conducted experiment investigation and numerical simulation based on various geometries of pin fin. Yang *et al.,* [13] showed triangular pin fin has a greatest resistance to fluid flow while circular pin fin has the least resistance. They showed that hexagonal and circular pin fin has flow guided structure that proved to be vital in improving the cooling performance. Yu *et al.,* [14] stated the friction factor of PPF is comparable with straight channel and much lower than other micro pin fin heat sinks. In their study, they concluded fluid extraction effect of PPF leads to the enhancement of heat transfer while reducing pressure drop.

Huang *et al.,* [15] conducted numerical simulation for microchannel heat sink with slotted flow passages. MCHS with different configurations i.e. parallel, staggered-slots and trapezoidal staggered channels were investigated. These configurations showed a better thermal-hydraulic performance than traditional straight rectangular channel. In addition, MCHS with larger slot width was found to give higher Nusselt number. They concluded that staggered-slot channel provides better thermal performance than parallel-slot channel, but trapezoidal staggered channel showed lower pressure drop.

Riera *et al.,* [16] numerically and experimentally studied the performance of the stepwise varying width microchannel cooling device. They found that the thermal performance of the microchannel cooling device is three-fold the performance of the minichannel for the same flow rate.

Xia *et al.,* [17] experimentally and numerically investigated the heat transfer and fluid flow characteristics in MCHS with complex corrugated profile. Comparisons were made against MCHS with rectangular profile. The results showed that the corrugated profile developed a higher pressure drop but a better heat transfer performance as the corrugated profile caused interruption and redevelopment of boundary layer.

Betz and Attinger [18] studied the use of segmented flow in enhancing heat transfer of MCHS. Segmented flow was created by injecting air bubbles periodically into water filled channels of MCHS. Results showed that segmented flow is capable of enhancing the Nusselt number by more than 100% provided mass velocity is in range of 330−2000 kg/m^2s.

Luo *et al.,* [19] was inspired by leaf vein structure that transports water in leaf to investigate thermal performance of leaf-vein-inspired microchannel. Three different channel structures based on the mid rib leaf structures of 3 selected plants were investigated. Numerical results showed that pitted wall thickening patterns of one of the leaf structure had the best heat and mass transfer characteristics. These structures were applied and modeled into microchannel heat sink and their results showed that they provide excellent heat transfer characteristic.

Recently, several studies [20,21] related to wavy microchannel heat sink have been conducted. Study by Lin *et al.,* [20] found that a novel design heat sink with wavy microchannel was capable of enhancing the heat transfer. In the work done by Zhu *et al.,* [21], they employed a three-dimensional fluid-solid conjugate model to investigate the heat transfer performance of the left-right and up-down wavy microchannel heat sinks for various wavy amplitudes, wavelengths, channel aspect ratios, and width ratios of channel to pitch. The up-down wavy design if found to give better performance for small wavelengths.

Thermal and hydraulic performance of micro-channel heat sink with hourglass profile (MCHS-HG) are investigated in the present study numerically. To the knowledge of the authors, research studies related to thermal performance of hourglass profile in MCHS can hardly be found. The performance is investigated numerically with Reynolds number in the range of 300-1800 and aspect ratio in the range of 4-6. MCHS-HGs have different solid-fluid interface area as compared to MCHS-R and affect the heat exchange. The hourglass profile is expected to affect the flow resistance. Therefore, it is interesting to understand the effect of the hourglass profile on thermal performance by varying its throat size with various contraction ratio defined hereafter. In addition, the pumping power and performance enhancement factor will be evaluated.

2. Numerical Methodology

MCHS with overall dimensions of $10\ mm \times 10\ mm \times 480\mu m$ is employed for this numerical study. The dimensions correspond to the overall length L, overall width w and overall height H of the heat sink as illustrated in the physical model in Figure 1(a). The overall dimensions used are referenced to the dimensions used by Tuckerman and Pease [1]. The heat sink consists of repeated sections, and only a single section will be used as computational domain. Figure 1(b) illustrates cross-section of the repeated section, showing hourglass channel profile that is used for investigation with w_w, w_c, H_{c1}, H_w and r equal to 50, 50, 50, 80 and 25 μm, respectively.

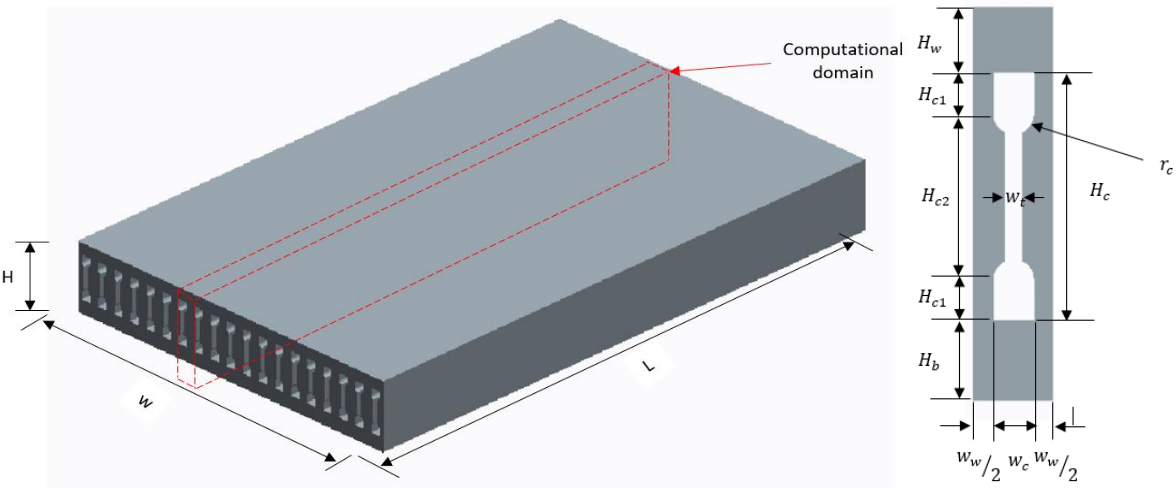

Fig. 1. (a) Schematic view of microchannel heat sink for study with computation domain clearly shown (b) Cross section of single channel used for simulation

In the present study, thermal and hydraulic performances of MCHS-HG are investigated. The hourglass geometry is constructed based on a rectangular channel with the center pinched to form a uniform throat with curves connecting the throat and upper and lower parts of channel. For a given AR, the channel height H_c and channel width w_c are fixed, and the hourglass profile is varied by contraction ratio, R_c which is defined as:

$$R_c = \frac{w_t}{w_c} \tag{1}$$

where w_t is the width of throat. Higher value of R_c means wider throat size and it becomes a rectangular channel when R_c=1. The aspect ratio AR is given as follows:

$$AR = \frac{H_c}{w_c} \tag{2}$$

The effect of AR is investigated with values 4, 5 and 6. For each value of AR, the thermal performance of the MCHS with hourglass profiles are investigated with Rc = 0.2, 0.4, 0.6, 0.8, 1.0. Silicon is used as the material of heat sink and water is used as coolant for the heat sink. Uniform heat flux of 790 W/cm^2 [1] is applied at base of MCHS with inlet temperature of 296K. The front and back solid surfaces, as well as the top surface are assumed to be adiabatic. The side walls are set to be symmetrical. The inlet velocity in is set at the channel inlet, and pressure outlet is set at the channel outlet. The heat flux and temperature are conserved at the fluid-solid interface. Few assumptions were made to simplify the simulations.

a. Fluid is steady, incompressible and laminar over the whole length of MCHS.
b. Properties for water and heat sink remain constant.
c. Heat transfer to surrounding is negligible.

With these boundary conditions and assumptions, the problem is solved using the following governing equations:

Continuity equation:

$$\nabla \cdot V = 0 \tag{3}$$

where V is velocity vector.

Momentum equation for fluid:

$$\rho_f (\mathrm{V} \cdot \nabla) V = -\nabla p + \mu_f \nabla^2 V \tag{4}$$

where ρ_f and μ_f is the density and dynamic viscosity of fluid respectively, and p is the pressure of the fluid.

Energy equation for fluid:

$$\rho_f c_{p,f} \left(V \cdot \nabla T_f \right) = k_f (\nabla^2 T_f) \tag{5}$$

where $c_{p,f}$, T_f and k_f is the specific heat capacity, temperature and thermal conductivity of fluid, respectively.

Energy equation for solid:

$$k_s (\nabla^2 T_s) = 0 \tag{6}$$

where k_s and T_s is the thermal conductivity and temperature of solid, respectively.

SIMPLEC algorithm is used to for pressure velocity coupling with pressure using PRESTO scheme, momentum and energy using power law scheme. The convergence residual criteria are set to be lower than 10^{-5} for continuity and velocity while convergence residual criteria for energy is set to be lower than 10^{-8}. Performance of MCHS is will be evaluated over a range of Reynolds number from 300-1800 for each case. Reynolds number is defined as:

$$Re = \frac{\rho_f u_{in} D_h}{\mu_f} \tag{7}$$

where u_{in} and D_h is the inlet velocity and hydraulic diameter of the MCHS respectively.

Hydraulic diameter, D_h is defined as:

$$D_h = \frac{4A}{P} \tag{8}$$

where A and P are the area and perimeter of cross section of channel, respectively.

Pressure drop is defined as:

$$\Delta p = \overline{p_{in}} - \overline{p_{out}} \tag{9}$$

where $\overline{p_{in}}$ and $\overline{p_{out}}$ is the average pressure of inlet and outlet, respectively

The average heat transfer coefficient, $\bar{h}$ is defined as follow:

$$\bar{h} = \frac{\dot{q''}}{\bar{T}_w - \bar{T}_f} \tag{10}$$

where $\bar{T}_w$ and $\bar{T}_f$ is the average temperature of the heated base and average bulk fluid temperature, respectively. $\dot{q''}$ is the heat flux at base of MCHS. $\bar{T}_w$, $\bar{T}_f$ is calculated with formula below:

$$\bar{T}_f = \frac{\int \rho_f T_f \, dV}{\int \rho_f \, dV} \tag{11}$$

$$\bar{T}_w = \frac{\int T_w \, dA}{\int dA} \tag{12}$$

Pumping power for the fluid flow through the MCHS is defined as,

$$\Omega = \dot{V} \Delta p \tag{13}$$

where $\dot{V}$ is the volumetric flow rate of coolant.

3. Results

Present study is validated against experimental results obtained by Tuckerman and Pease [1]. Comparison presented in Table 1 shows a maximum deviation of 2.182%, which shows that the numerical model is in good agreement with the experiment results [1].

Table 1
Validation with Tuckerman and Pease [1] experimental results for numerical method used

w_c (μm)	w_w (μm)	H_c (μm)	Heat flux (W/cm^2)	Results (K/W) Experiment [1]	Present study	Deviation (%)
56	44	320	181	0.110	0.1076	2.182
55	45	287	277	0.113	0.1131	0.088
50	50	302	790	0.090	0.0905	0.556

Grid independence test has been carried out and the results are shown in Table 2. Mesh with 1.3 million elements showed a difference of 0.6247% with 3.4 million elements which is negligible. Therefore, mesh size of about 1.3 million is applied throughout all the simulations.

Table 2
Results of grid independence test

No. of elements	$\bar{h}$ (W/m^2K)	Deviation (%)
330 thousand	23.25	3.165
629 thousand	23.67	1.416
1.3 million	23.86	0.6247
2.45 million	23.94	0.2915
3.4 million	24.01	-

The thermal performance of the MCHS with hourglass profiles has been investigated with Rc = 0.2, 0.4, 0.6, 0.8, 1.0, and the results of average heat transfer coefficient $\bar{h}$ against Reynolds number are presented in Figure 2(a), Figure 2(b) and Figure 2(c) for aspect ratio 4, 5 and 6, respectively. Generally, Figure 2 shows that, the value of $\bar{h}$ increases with Re regardless of the value of AR. For a given Reynolds number and Rc, increasing AR results in higher value of $\bar{h}$ [5]. For the effect of varying Rc, it can be observed in Figure 2(a), Figure 2(b), and Figure 2(c) that, Rc = 0.4 provides highest value of $\bar{h}$ except the case of Re = 300 at which Rc = 0.6 is slightly higher in $\bar{h}$. For the range of Reynolds number investigated, the highest enhancement in $\bar{h}$ of about 28% can be achieved at the conditions of Rc = 0.4 at Re = 1800 as compared to MCHS-R.

It can be observed in Figure 3 that, the channel geometry with contration ratio of Rc = 0.2 possesses the highest solid-fluid interface area. Despite this, channel geometry with Rc = 0.2 do not give highest value of $\bar{h}$ but mediocre. The value of $\bar{h}$ decreases as Rc increases from 0.6 to 1.0. This means there could be other factors that govern the heat transfer other than the solid-fluid interface area.

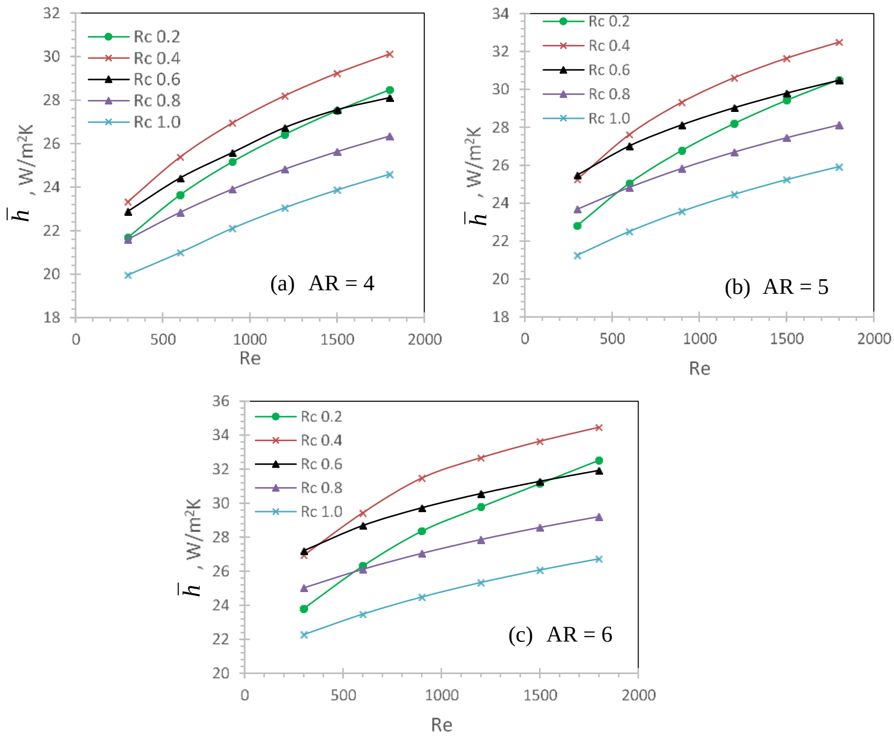

Fig. 2. Variation of $\bar{h}$ with Re for (a) AR = 4, (b) AR = 5 and (c) AR = 6

To further analyse the effect of contraction ratio on thermal performance, the temperature distribution of MCHS are presented in Figure 3 for different values of Rc at Re = 1800 and AR = 5. Note that the temperature distribution is presented at the middle section between inlet and outlet.

In Figure 3(a) (Rc = 0.2), it is clearly shown that, near the throat region, the fluid temperature is relatively closer to the solid surface temperature. Comparing among the values of Rc, the case of Rc = 0.2 has relatively narrower throat size, therefore the fluid that flows through the narrower throat has relatively closer proximity to the solid surface. The fluid will be more effectively absorbing heat from the solid surface. However, since it is the narrowest throat, the amount of fluid flow is relatively lesser, and this limit the capacity to carry away heat. This could be the reason why Rc = 0.2 do not give highest heat transfer coefficient despite having highest fluid-solid interface area and close proximity of fluid to the solid.

For the case of Rc 0.4, it can be observed in Figure 3(b) at the fluid-solid interface region that, the fluid temperature is highest near the solid wall, but develop a significant temperature gradient towards the center of the throat. In Figure 3(b), it also shows that, the fluid at the center has the lowest temperature as it is considered far for the heat from the solid to reach. Considering larger amount of fluid flowing through the throat as compared to the case of Rc = 0.2, the case Rc = 0.4 should have a higher capacity to transfer the heat away but not expected to be very much different as effective heat transfer is only occurring through the fluid near the interface. Further observation for other cases of MCHS-HG (Rc = 0.6, 0.8) in Figure 3(c) and Figure 3(d) reveals similar characteristics that heat is only transferred effectively to the fluid near the interface. Large portion of liquid near the center of the fluid remains at low temperature. This means increasing the volume of fluid flow by having larger size of throat (higher value of Rc) enhances the heat transfer but not significantly. On the other hand, based on the hourglass profile, larger size of throat gives smaller solid-fluid interface area which is not favourable to the heat transfer. From the above, it implies that, the MCHS-HG with Rc = 0.4 gives highest heat transfer coefficient simply due to the compromise between the interfacial heat transfer area and the amount of fluid flow that subject to the size of the throat (hourglass profile).

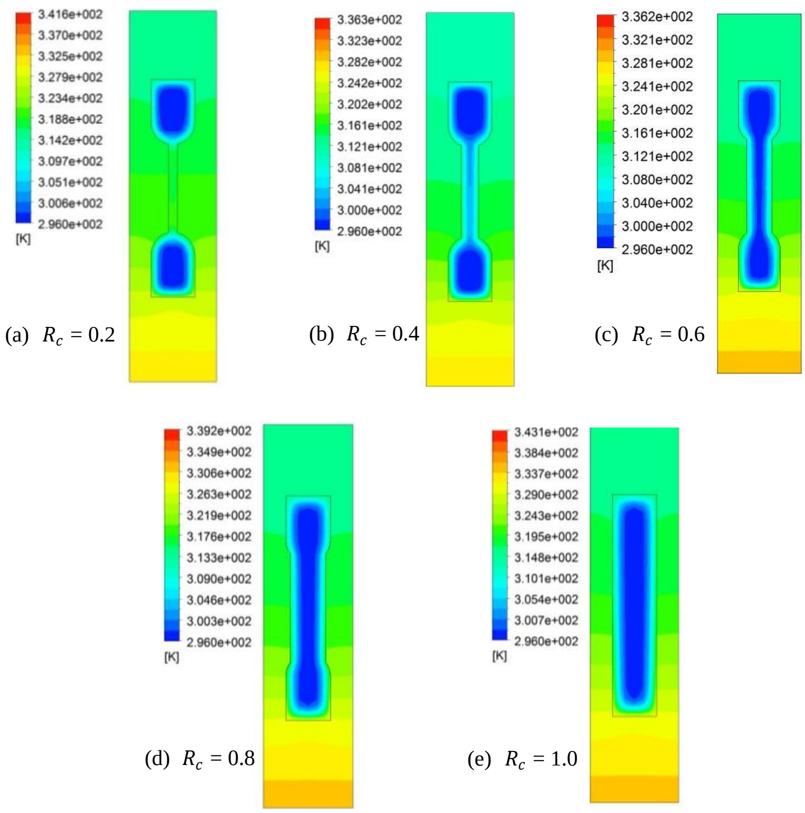

Fig. 3. Temperature distribution of MCHS midway between inlet and outlet for AR=5 and Re=1800

Despite the better thermal performance of MCHS-HG, increase in pumping power need to be considered for the practicability of MCHS-HG for future applications. To understand the hydraulic performance, the results of pumping power for different Reynolds numbers and different values of Rc have been obtained. The results are presented in Figure 4(a), Figure 4(b), and Figure 4(c) for AR = 4, 5 and 6, respectively. The results show that, for any given value of Rc, Ω increases with AR or Re. This could be due to higher flow resistance as a results of higher aspect ratio [5]. Figure 4 also shows that, for any given values of Reynolds number, Ω increases with decrease in Rc. Literally, it means narrower throat requires higher the pumping power to drive the fluid. This is attributed to the increase in pressure drop when the throat is getting narrower. The large pressure drop could be due to the flow constriction at narrower throat and larger fluid-solid interface area.

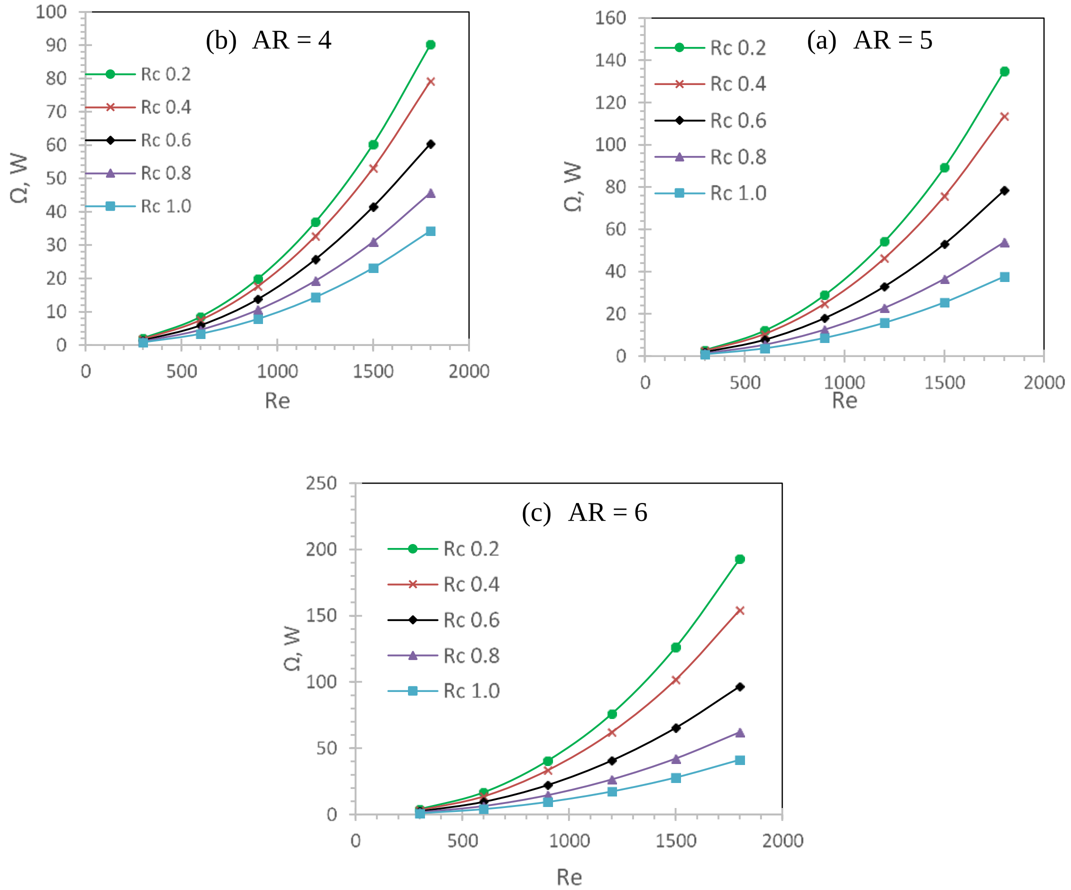

Fig. 4. Variation of pumping power with Re

The results discussed above reveal that the MCHS-HG offer better thermal performance at the expense of the pumping power. Performance enhancement factor, η is used to evaluate the thermal hydraulic performance to assess whether it is economical to be implemented. η is defined as follow:

$$\eta = \frac{\bar{h}/\bar{h}_o}{\Omega/\Omega_o} \tag{13}$$

which subscript 'o' denotes MCHS-R.

Performance enhancement factor, η is the ratio of dimensionless heat transfer coefficient to dimensionless pumping power with reference to the rectangular channel. If it is larger than unity, enhancement in heat transfer is considered greater than the increase in pumping power, and therefore achieving a better thermal hydraulic performance than MCHS-R. Figure 5 showed all MCHS-HG has values of η lower than 1 with Rc of 0.2 being the poorest at high Reynolds number. Increase in $\bar{h}$ of MCHS-HG is at relatively higher expense of pumping power, and therefore MCHS-HG is not economical to be implemented. Figure 5 also shows that the most economical MCHS-HG are those design with lower value of AR and high value of Rc.

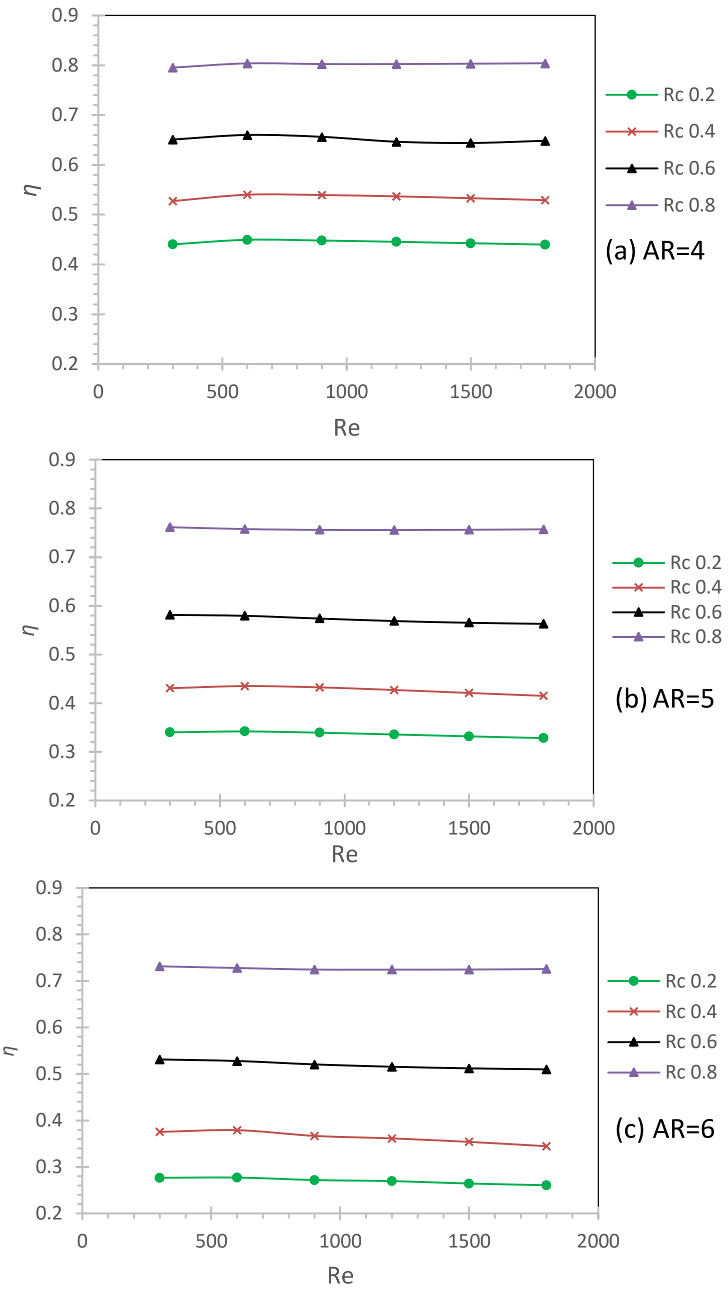

Fig. 5. Variation of η with Re for different AR

4. Conclusions

A numerical investigation has been carried out to study the thermal and hydraulic performance of microchannel heat sink with hourglass channel (MCHS-HG) and compared to microchannel heat sink with straight rectangular channel (MCHS-R). A 3-D conjugate heat transfer model was employed to solve the problem. The numerical method used has been validated against the experimental work of a literature. MCHS-HG with 5 different values of contraction ratio and 3 different values of aspect ratio were investigated over a range of Reynolds number of 300-1800. The effect of these two parameters on the thermal and hydraulic performance has been investigated and the results can be concluded as follow:

i. The value of contraction ratio of 0.4 gives optimum heat transfer coefficient, capable of achieving 28% enhancement as compared to the MCHS-R.
ii. Lower values of contraction ratio requires higher pumping power.
iii. Increase in aspect ratio of the MCHS-HG results in higher heat transfer coefficient but requires higher pumping power for MCHS-HG has much lower thermal-hydraulic performance than the MCHS-R and therefore not economical to be implemented.

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