

PERFORMANCE OF A MULTI-STACK MICROCHANNEL HEAT SINK

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Article history

Received

21 June 2016

Received in revised form

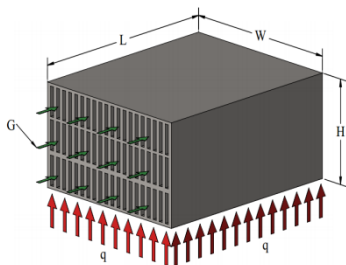
26 June 2016

Accepted

3 July 2016

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Graphical abstract



Abstract

The usage of a very large scale integrated circuits generate high heat fluxes and require an effective cooling system. A microchannel heat sink (MCHS) is one of the reliable cooling systems that had been applied. In terms of performance, a MCHS can be appraised by obtaining low total thermal resistance and pumping power. However, as the total thermal resistance decreases, the pumping power will increase. A few studies have been focused on the minimization of the thermal resistance and pumping power of a multi-stack MCHS. Optimization of two objective functions which are the total thermal resistance and pumping power has been done by using genetic algorithm. It is demonstrated that both objective functions can be minimized by optimizing two design variables which are the channel aspect ratio, α , and wall width ratio, β . It was found that the usage of a stacked configuration for the MCHS is able to reduce the total thermal resistance. From the optimization, it was found that the optimum number of stacks that can be implemented is three. With the three-stack configuration, the total thermal resistance found is 0.1180 K/W which is 21.8% less compared to the single-stack MCHS. However, the pumping power needed for the three-stack MCHS is increased by 0.17 % compared to single-stack which is 0.7535 W.

Keywords: Multi-stack, micro-channel heat sink, optimization, performance

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1.0 INTRODUCTION

Due to the rapid increase in the power density of electronic devices, effective thermal management becomes an important issue in the design process. Initially, space restriction in the design of electronic devices was not a concern. However, as the microelectronic mechanical systems (MEMS) keep developing, space reserved for each compartment in the system has become smaller. Consequently, the electronic chips become smaller and produce higher heat flux as the function in the chip probably increased or remains unchanged. Therefore, the application of a microchannel heat sink (MCHS) is extensively used for the MEMS as an effective cooling system.

For a better performance of the MCHS, it can be attained through the use of different types of structural material, different types of geometry and different types of coolant [1]. Optimization of the performance of a MCHS in terms of different materials used during fabrication process can be achieved by using a material with a high thermal conductivity.

Different channel geometries such as rectangular, circular, triangular and trapezoidal also affect the cooling process in a MCHS. Different geometry will have different total effective area which is important for the heat transfer process. However, the difficulties of the fabrication technique must be considered during the selection of the channel geometry.

The properties of a coolant play an important role during the selection of the working fluids. Different types of working fluids will have different properties

which is important for the heat removal process. The working fluids often used in a MCHS were air, water and refrigerants. Recently, nanofluids have been investigated as a potential coolant in a MCHS due to their high heat transfer capacity.

The research on the MCHS was first started by Tuckerman and Pease [2] associated with the heat transfer characteristics using theoretical and experimental work. During the investigation, the MCHS was fabricated directly onto the backside of a silicon wafer and able to remove heat flux up to 790 W/cm^2 with the total thermal resistance of $0.090 \text{ }^\circ\text{C/W}$ with water as the coolant. In their analysis, the total thermal resistance was obtained by including only the resistance due to base conduction, resistance due to convection between the channel surface and coolant and capacitive thermal resistance. By considering the results achieved as a standard, many researchers had studied the optimization of the MCHS.

Knight *et al.* [3, 4] performed an analysis of a MCHS by using the equation of fluid dynamics and the basics of conduction and convection heat transfer. The analysis covered the nature of flow for both laminar and turbulent regions. The results showed that as the pressure drop was small, thermal resistance found in the laminar region was much smaller than in the turbulent region. Conversely, when the pressure drop in the channel was large, turbulent region contributed to a smaller thermal resistance. By comparing the results found with the previous work by Tuckerman and Pease, the thermal resistance decreased in the range of 10 to 35 %.

Vafai and Zhu [5] started the concept of two stacks of a MCHS. By allowing a counter current flow of the coolant, thermal analysis and temperature distribution were analyzed. The results showed that the thermal performance and temperature distribution of two-stack have improved compared to a single-stack. With the same constraint set, the temperature difference for the one-layered MCHS was 15°C higher than the two-layered MCHS.

Wei and Joshi [6] started the investigation on a stacked MCHS for a fixed pressure drop, fixed volumetric flowrate and fixed pumping power. For a fixed pressure drop at 10kPa, the total thermal resistance resulted from a single-stack was $0.120 \text{ }^\circ\text{C}/(\text{W/cm}^2)$. Meanwhile, the total thermal resistance obtained from a two-stack was decreased up to $0.082 \text{ }^\circ\text{C}/(\text{W/cm}^2)$. For a fixed volumetric flowrate at $0.83 \times 10^{-6} \text{ m}^3/\text{s}$, the total thermal resistance for the two-stack was smaller than single-stack. However, as the number of stacks keeps increasing, the total thermal resistance tends to increase due to increasing total effective area and decreasing in heat transfer coefficient.

A few years later, Wei and Joshi [7] used genetic algorithm method to investigate the reduction of total thermal resistance of a stacked MCHS. During the investigation, the MCHS was fabricated using silicon with water as the working fluid. The pumping power

was set at 0.01W to 0.8W while the volumetric flowrate was maintained within $1.67 \times 10^{-5} \text{ m}^3/\text{s}$. The effects of the number of stacks and the pumping power were investigated to achieve minimum total thermal resistance. For their research, the total thermal resistance was found by considering conduction and convection thermal resistance in the base and channel of a MCHS, constriction and capacitive thermal resistance.

There are still few studies on the stacked MCHS, particularly on the effects of stacking on the thermal and hydrodynamic performances. This research focused on the minimization of the thermal resistance and pumping power of a multi-stack MCHS with a fixed height of the channel, H_c , in view of the constraint in space in miniaturized systems. In order to achieve an accurate result, the analysis has been done by considering a full channel as the domain of analysis, unlike previous studies which looked at half the channel width. The total thermal resistance was obtained by including all components of the thermal resistance which are conduction in the channel base and channel wall, convection from the channel base and channel wall, constriction and capacitive thermal resistance. The analysis also considered a transformation of the total resistance made of serial and parallel resistances into a simpler representation.

2.0 METHODOLOGY

Figure 1 and 2 illustrates the schematic diagram of a multi-stack MCHS model with their important dimensions. The overall dimensions of the heat sink are represented by the length, L , width, W , height, H and thickness, t . The channels are defined by the channel width, w_c , wall width, w_w , and channel height, H_c . The bottom of the MCHS experiences a uniform heat load, q , that came from electronic chips. The working fluid flows with a volumetric flowrate, G , passing through the channels in parallel flow.

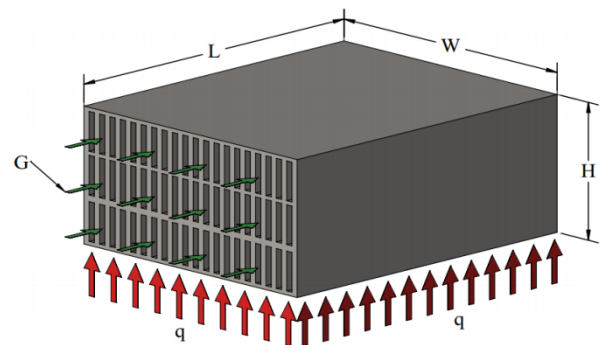


Figure 1 Schematic diagram of the multi-stack MCHS model

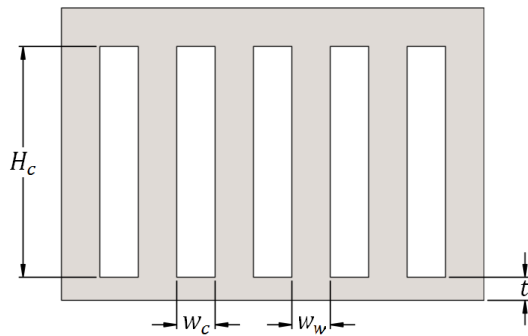


Figure 2 Important dimensions of a MCHS model

The assumptions made during the analysis are:

- Steady state flow and heat transfer
- Fully developed laminar flow
- One-dimensional analysis
- Thermal radiation is neglected
- Uniform heat flux experience at each stack
- Constant thermo-physical properties of coolant

There are two techniques provided in this research; a parametric study and optimization technique using Genetic Algorithm (GA). Table 1 lists the geometrical properties of a MCHS in this research. Table 2 lists the thermo-physical properties of the coolant used, water in this case.

Table 1 Geometrical properties of a MCHS [2]

Material	Silicon
Length, L (m)	0.01
Width, W (m)	0.01
Substrate Thickness, t (m)	213×10^{-6}
Thermal Conductivity, k ($W/m \cdot K$)	150
Channel Height, H_c (m)	320×10^{-6}

Table 2 Thermo-physical properties of the coolant [8]

Coolant	Water
Temperature, T ($^{\circ}C$)	23
Density, ρ (kg/m^3)	997.4
Specific Heat, C_p ($J/kg \cdot K$)	4180.8
Thermal Conductivity, ($W/m \cdot K$)	0.6034
Dynamic Viscosity, μ ($kg/m \cdot s$)	0.9354×10^{-3}
Volumetric Flowrate, G (m^3/s)	4.7×10^{-6}

2.1 Total Thermal Resistance for a Multi-Stack MCHS

Figure 3 shows the thermal resistance network for a multi-stack MCHS with different types of resistance across a MCHS. Heat generated by the electronic chips will first face R_1 which is the sum of resistances due to conduction and constriction at the base of a MCHS. In the channel wall, the heat will encounter R_2 which is the resistance due to the wall conduction. In the channels, the heat will experience R_3 and R_4 which are the resistances due to the base and wall convection respectively. At the same time, the working fluid will absorb the heat and carry it away. The process of removing heat in the fluid involves R_5 which is the capacitive thermal resistance.

In order to solve the total thermal resistance for a multi-stack MCHS, the total thermal resistance of the upper layer ($N = 1$) must be obtained first and can be expressed as

$$R_{total}^N = R_1 + \left[\frac{(R_2 + R_4)(R_3)}{(R_2 + R_4 + R_3)} \right] + R_5 \quad (1)$$

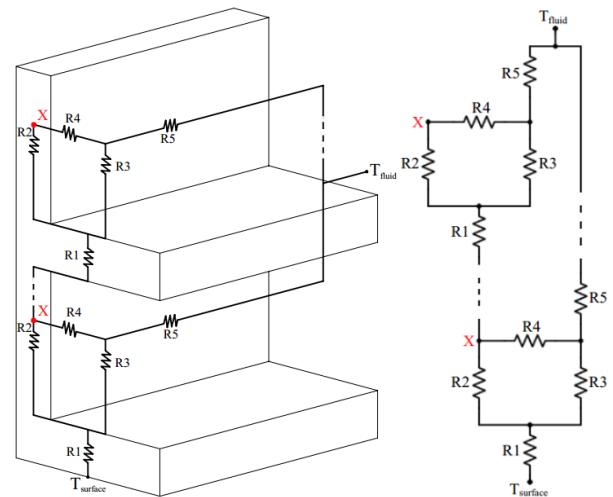


Figure 3 Thermal resistance network for a multi-stack MCHS

However, after solving for the upper layer, the thermal resistance network was found to be in non-series-parallel arrangement. Therefore, a transformation known as delta-to-wye transformation was done to convert the non-series-parallel to the series-parallel arrangement. Figure 4 shows the thermal resistance network for the non-series-parallel and series-parallel arrangement.

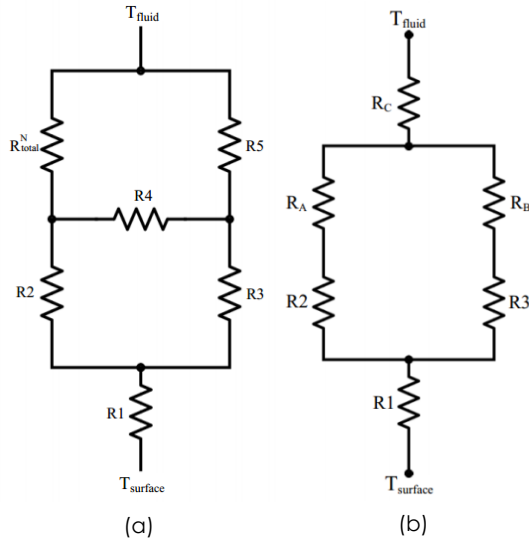


Figure 4 Thermal resistance network for (a) Non-series-parallel and (b) Series-parallel

The delta-to-wye transformation [9] can be completed by the following equations

$$R_A = \frac{R_{total}^N \cdot R_4}{R_{total}^N + R_4 + R_5} \quad (2)$$

$$R_B = \frac{R_4 \cdot R_5}{R_{total}^N + R_4 + R_5} \quad (3)$$

$$R_C = \frac{R_5 \cdot R_{total}^N}{R_{total}^N + R_4 + R_5} \quad (4)$$

Therefore, the total thermal resistance for the next layer (N+1) can be found as

$$R_{total}^{N+1} = R_1 + \frac{(R_2 + R_A)(R_3 + R_B)}{(R_2 + R_A + R_3 + R_B)} + R_C \quad (5)$$

In order to obtain the total thermal resistance of the following layer, the value from Eq. (5) should be assigned into R_{total}^N in Eqs. (2, 3 and 4) and Eq. (5) must be executed again. The value N starts with one and undergoes an increment of one after the execution of Eq. (5).

In the optimization process of a multi-stack MCHS, there are two design variables that play an important role in order to determine the geometrical properties of a multi-stack MCHS. They are defined as

$$\alpha = \frac{H_c}{w_c} \quad (6)$$

$$\beta = \frac{w_w}{w_c} \quad (7)$$

All the components of the thermal resistance in terms of both design variables can be expressed as

$$R_{base\ cond} = \frac{t}{k_{hs}(WL)} \quad (8)$$

$$R_{cons} = \frac{1 + \beta}{\pi k_{hs}(WL)} \ln \left[\frac{1}{\sin \frac{\pi\beta}{2(1+\beta)}} \right] \alpha H_c \quad (9)$$

$$R_{wall\ cond} = \frac{H_c(1 + \beta)}{k_{hs}(WL)} \quad (10)$$

$$R_{wall\ conv} = \frac{1 + \beta}{2h_{conv}\alpha\eta(WL)} \quad (11)$$

$$R_{base\ conv} = \frac{1 + \beta}{h_{conv}(WL)} \quad (12)$$

$$R_{capa} = \frac{L}{C_p \mu_f (WL)} \frac{2}{Re} \frac{1 + \beta}{1 + \alpha} \quad (13)$$

where h_{conv} is the heat transfer coefficient which is evaluated based on the correlation proposed by Kim and Kim [10], η is the fin efficiency and Re is Reynolds number which are expressed as

$$h_{conv} = 2.253 + 8.164 \left(\frac{\alpha}{\alpha + 1} \right)^{1.5} \frac{k_f}{D_h} \quad (14)$$

$$D_h = \frac{4H_c w_c}{2(H_c + w_c)} = \frac{2}{1 + \alpha} H_c \quad (15)$$

$$\eta = \frac{\tanh(mH_c)}{mH_c} \quad (16)$$

$$m = \sqrt{\frac{2h_{conv}}{k_{hs}w_w}} \quad (17)$$

$$Re = \frac{\rho_f V D_h}{\mu_f} = \frac{2\rho_f G}{\mu_f n H_c} \frac{\alpha}{\alpha + 1} \quad (18)$$

$$n = \frac{W}{w_w + w_c} \quad (19)$$

The total thermal resistance obtained from Eq. (5) was used as the first objective function for the optimization purpose

2.2 Pumping power for a Multi-Stack MCHS

The hydrodynamic performance is referred to as the total pressure drop or the pumping power. The modified total pressure drop [11] can be expressed as

$$\Delta p = f \frac{(1 + \alpha)L}{2H_c} \rho_f \frac{V^2}{2} + \left[1.79 - 2.32 \left(\frac{1}{1 + \beta} \right) + 0.53 \left(\frac{1}{1 + \beta} \right)^2 \right] \rho_f \frac{V^2}{2} \quad (20)$$

where f is the friction factor and V is the velocity of the working fluid. The first term is the pressure drop due to the channel friction losses and the second term is due to the entrance and exit channel losses [12]. The friction factor and velocity of the working fluid can be expressed as

$$f = \frac{64}{Re} \tag{21}$$

$$V = \frac{G}{n H_c w_c} \tag{22}$$

The pumping power needed is defined by

$$P_p = \Delta p \cdot G \tag{23}$$

The pumping power of Eq. (23) was used as the second objective function for the optimization purpose. It is common knowledge that a decrease in the thermal resistance increases the pressure drop, subsequently the pumping power. Thus, it is imperative that an optimal condition is found whereby simultaneous minimization of the thermal resistance and pumping power is achieved.

2.3 Model Validation

A parametric study was first carried out in order to validate all the equations defined. The validation was made by comparing the results from the current model with that of Wei and Joshi [7]. The result of the validation is tabulated in Table 3. Only a minor difference of the thermal resistance is found from the comparison which is 1.4%. The different results are due to the different sources of the thermo-physical properties of the coolant used. Wei and Joshi did not specify the source of their data while in this research, the properties have been obtained from Cengel and Ghajar [8].

Table 3 Model validation with Wei and Joshi [7]

	Wei and Joshi	Current Model
N	2	2
G (cm^3/s)	4.5	4.5
H_c (μm)	400	400
w_c (μm)	107	107
w_w (μm)	41	41
R_{th} (K/W)	0.213	0.210
Relative uncertainty R_{th} (%)	-	1.4

2.4 Optimization Technique

In the current study, the optimization was completed using Genetic Algorithm (GA) in the toolbox provided in the Matlab R2012a software [13]. Figure 5 shows the

flowchart of the multi-objective optimization process using GA. In order to execute GA correctly, there are several parameters that should be specified. Table 4 lists the parameters involved.

Table 4 Parameter selection for optimization process

Objective Function	Total Thermal Resistance and Pumping Power
Design Variable Limit	$1 \leq \alpha \leq 10$ $0.1 \leq \beta \leq 1$
Population Size	200
Selection Function	Tournament
Mutation	Constraint Dependent
Crossover Function	Intermediate
Stopping Criteria	Function Tolerance : 1×10^{-6}
Plot Function	Pareto Front

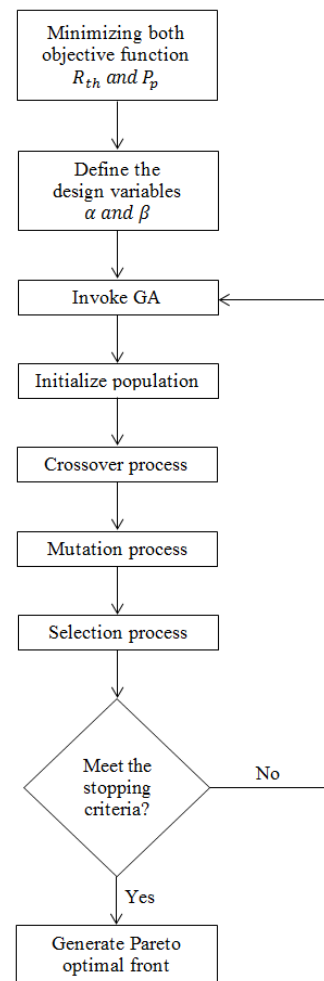


Figure 5 Flowchart of the optimization process

3.0 RESULTS AND DISCUSSION

Figure 6 shows the optimized total thermal resistance with the corresponding pumping power required for different stacks. As for a single-stack MCHS, the optimized total thermal resistance is found to be 0.1509 K/W with the corresponding pumping power 0.7522 W . When the number of stack increases to two-stack, the optimized total thermal resistance is decreased by 18% compared to the single-stack which is 0.1234 K/W . The equivalent pumping power for the two-stack is 8% less than single-stack which is 0.6910 W . As for the three-stack MCHS, the optimized total thermal resistance is 22% less than single-stack which is 0.1180 K/W . However, the pumping power needed for the three-stack increases by 0.2% compared to single-stack which is 0.7535 W . As for the following stacks, the optimized total thermal resistance gives a minor difference. However, there is a significant increase with their corresponding pumping power. Therefore, the optimum number of stacks that can be implemented is three.

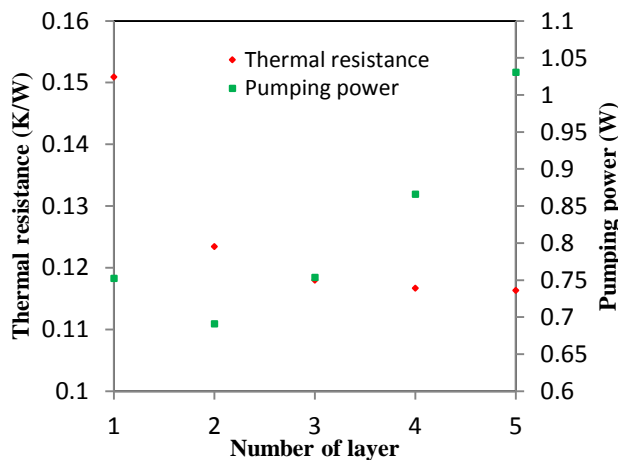


Figure 6 Optimized thermal resistance with corresponding pumping power for different stacks

The results from the optimization of both objective functions of a multi-stack MCHS is shown in Figure 7. From the results obtained, it can be inferred that as the number of stacks increased up to 5 stacks, the total thermal resistance keep decreasing to 0.1163 K/W . The decreasing of the thermal resistance is due to the increasing number of channels as the number of stacks increases and thus increasing the total effective area. Increasing the total effective area will cause a decrease in convective thermal resistance which is a major contributor to the total thermal resistance. However, it should be noted that as the total thermal resistance decreases, more pumping power is needed as the number of stacks increases. The increasing in pumping power is necessary in order to maintain the volumetric flowrate at each stack.

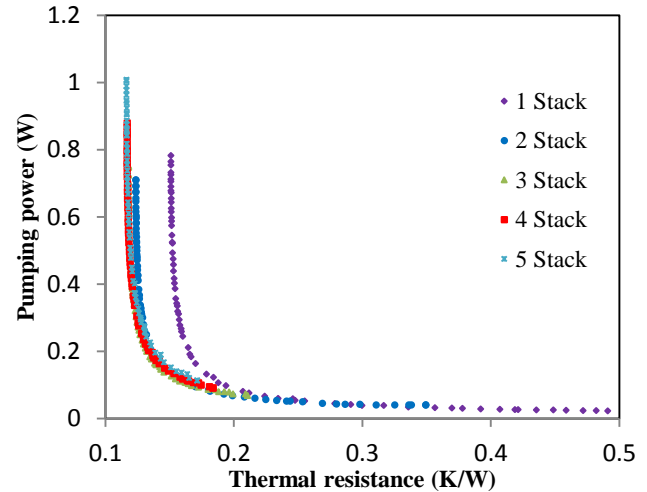


Figure 7 Optimized pumping power against thermal resistance for different stacks

The relationship between the total thermal resistance and both design variables which are channel aspect ratio and wall width ratio for different stacks is presented in Figure 8 and 9 respectively. The increase in both design variables results in decreasing the channel width as the channel height has been kept constant during the optimization process. As the channel width decreases, the number of channels that can fit along the width of the heat sink increases. Increasing the number of channels causes the total effective area to increase and improves the convective thermal resistance. At the same time, a decrease in the channel width also causes the heat transfer coefficient to increase and thus again improves the convective thermal resistance. Therefore, it can be clearly seen that as both design variables increases, the total thermal resistance decreases exponentially. As expected that an increase in the number of stacks also leads to the decreasing in total thermal resistance. However, as the number of stacks keeps increasing after three-stack, there is only a minor difference in total thermal resistance. This strongly supports the statement stated earlier that the three-stack is the optimum number of stacks.

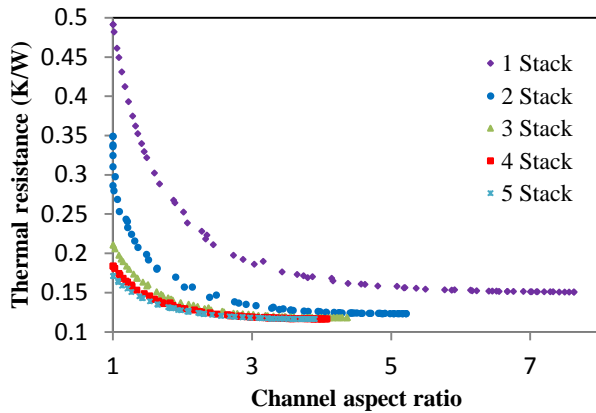


Figure 8 The impact of the channel aspect ratio on the thermal resistance

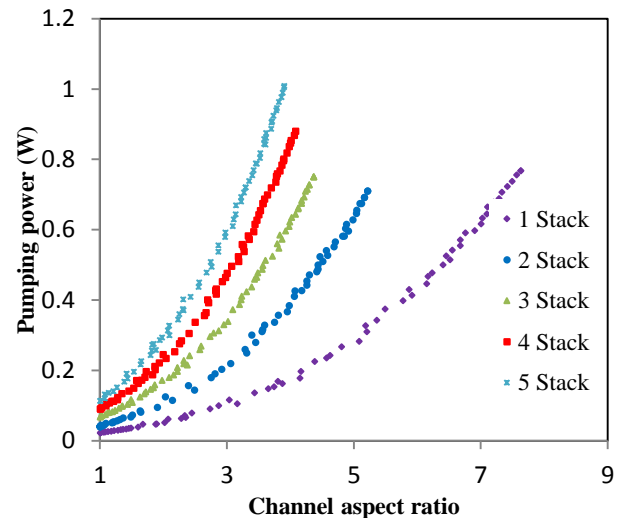


Figure 10 The impact of the channel aspect ratio on the pumping power

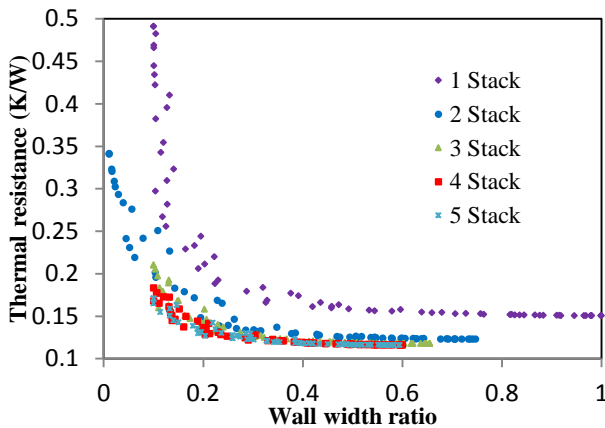


Figure 9 The impact of the wall width ratio on the thermal resistance

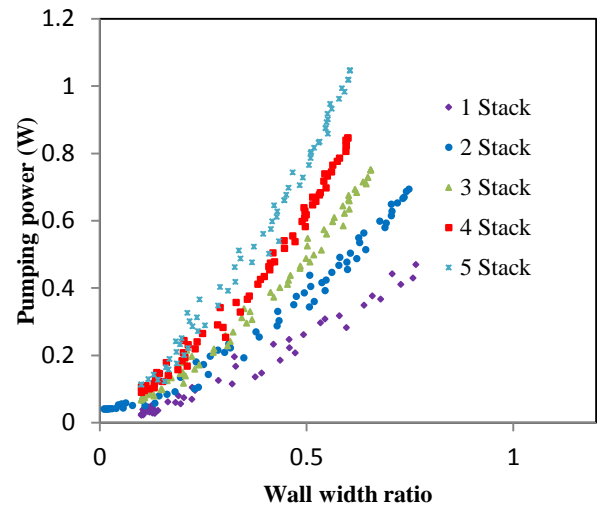


Figure 11 The impact of the wall width ratio on the pumping power

The relationship between the pumping power and both design variables which are channel aspect ratio and wall width ratio for different stacks is presented in Figure 10 and 11 respectively. In general, the pumping power increases exponentially as both design variables increases. Even though the velocity of the working fluid decreases with the increasing of both design variables, there are some parameters that cause the increase in pumping power. Decreasing in velocity of the working fluid causes the Reynolds number to decrease and thus increasing the friction factor. As the friction factor increase, the pressure drop across the MCHS increase and thus increasing the pumping power. The increasing in the number of stacks also causes the pumping power to be increased in order to maintain the volumetric flowrate at each stack.

4.0 CONCLUSION

The performance of a multi-stack MCHS has been theoretically investigated and has been optimized. A parametric study completed on the current model in this study shows a good agreement with previous published work. The optimization of the two objective functions which are total thermal resistance and pumping power was dependent on the two design variables which are channel aspect ratio and wall width ratio. It was found that the implementation of a multi-stack MCHS is able to reduce the total thermal resistance. However, reducing the total thermal resistance causes the pumping power to be increased.

From the optimization process, it was found that the optimum number of stacks that can be implemented is three. With the three-stack configuration, the total thermal resistance found is 0.1180 K/W which is 22% less compared to single-stack. However, the pumping power needed for the three-stack increases by 0.2% compared to single-stack which is 0.7535 W .

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