

THERMAL AND HYDRODYNAMIC PERFORMANCE OF A MICROCHANNEL HEAT SINK COOLED WITH CARBON NANOTUBES NANOFLUID

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Article history

Received

21 June 2016

Received in revised form

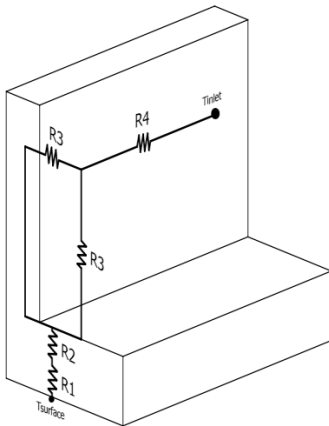
26 June 2016

Accepted

3 July 2016

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Graphical abstract



Abstract

The microchannel heat sink (MCHS) has been established as an effective heat removal system in electronic chip packaging. With increasing power demand, research has advanced beyond the conventional coolants of air and water towards nanofluids with their enhanced heat transfer capabilities. This research had been carried out on the optimization of the thermal and hydrodynamic performance of a rectangular microchannel heat sink (MCHS) cooled with carbon nanotube (CNT) nanofluid, a coolant that has recently been discovered with improved thermal conductivity. Unlike the common nanofluids with spherical particles, nanotubes generally come in cylindrical structure characterized with different aspect ratios. A volume concentration of 0.1% of the CNT nanofluid is used here; the nanotubes have an average diameter and length of 9.2 nm and 1.5 μm respectively. The nanofluid has a density of 1800 kg/m^3 with carbon purity 90% by weight having lignin as the surfactant. The approach used for the optimization process is based on the thermal resistance model and it is analyzed by using the non-dominated sorting multi-objective genetic algorithm. Optimized outcomes include the channel aspect ratio and the channel wall ratio at the optimal values of thermal resistance and pumping power. The optimized results show that, at high operating temperature of 40°C the use of CNT nanofluid reduces the total thermal resistance by 3% compared to at 20°C and consequently improve the thermal performance of the fluid. In terms of the hydrodynamic performance, the pumping power is also being reduced significantly by 35% at 40°C compared to the lower operating temperature.

Keywords: Carbon nanotube, nanofluid, microchannel, optimization, genetic algorithm

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1.0 INTRODUCTION

Ever since the invention by Tuckerman and Pease in 1981 [1], the microchannel heat sink (MCHS) had gained much interest throughout the world. The

MCHS device was demonstrated to cope with a heat flux of up to 790 (W/cm^2). Few decades had passed since the first invention of the MCHS. Since then, numerous operating parameters had been analyzed in order to optimize the performance of the MCHS. According to Adham *et al.* [2] the parameters consist

of the incorporation of the different coolant type, different material used in the fabrication and also different channel shape towards the MCHS. The use of different types of coolant has recently been introduced. Nanofluid as addressed by Saidur *et al.* [3] is a fluid with nanosized particles ranged within 1 to 100 nanometre suspended in a base fluid. A nano size particle is so small such that it cannot be seen with the naked eyes.

In 2007, the first ever nanofluid integrated into a MCHS was proposed by Tsai and Chein [4]. They demonstrated that the working fluid which is CNT-H₂O and Cu-H₂O provided a better heat transfer than pure water. The total thermal resistance for CNT-H₂O and Cu-H₂O were 0.0657 K/W and 0.0642 K/W respectively. Meanwhile, pure water exhibited only 0.086 K/W of the total thermal resistance.

A study by Maré *et al.* [5], showed that carbon nanotube (CNT) nanofluid is a highly potential candidate in the employment of the working fluid in a MCHS. This is due to the fact that the thermal conductivity of a CNT nanofluid is higher than common nanofluids thus providing a better performance to dissipate heat from the system.

Mohammed *et al.* [6], investigated the performance of a MCHS with alumina water as the working fluid. The study revealed that as the particle volume fraction increased, the thermal performance of the MCHS is improved. Another parameter that also increased with the increasing volume concentration was the heat transfer coefficient and the wall shear stress. The heat transfer coefficient increase was described by the presence of the nanoparticles in the base fluid. The nanoparticles enhanced significantly the thermal performance of the working fluid. In terms of the wall shear stress, it was increased as the volume fraction increased. The presence of the particles caused shear to occur at the wall of the MCHS. As the nanofluid increased to 5% volume fraction, the thermal performance of the MCHS could not be increased further. This was because the nanoparticle sedimented to the base of the MCHS. Thus, only the base fluid worked as the working fluid to carry the heat away from the MCHS.

A recent study by Halefadi *et al.* [7] utilized aqueous carbon nanotube (CNT) nanofluid and it improved the thermal performance of the MCHS. In the study involved, the weight concentration of the CNT nanoparticles was only 0.01%. Here, the convective heat transfer of the fluid increased as the temperature increased. It was reported that the convective heat transfer at 20°C, 30°C and 40°C increased by 2%, 12% and 13% respectively. From these data obtained, CNT nanofluid is very compatible to be implemented as a working fluid in a MCHS because the operating temperature of a MCHS is higher than room temperature. A very large system integration (VLSI) of microchips produces more heat as it processes more information. Therefore, for higher end applications, CNT nanofluid provides a gateway to further improve the thermal

performance of the MCHS. This research was completed to investigate the thermal and hydrodynamic performance of a MCHS cooled with CNT nanofluid. The properties had been obtained experimentally [8], today the availability of which is limited. The CNT type is termed N2 with average diameter and length of 9.2 nm and 1.5 μm respectively. N2 has a density of 1800 kg/m³ with carbon purity 90% by weight having lignin as the surfactant. Experimentally obtained properties are crucial due to their variation at different operating temperatures. The performance of the particular CNT nanofluid in this research is only being studied here for a MCHS application. The optimization was done with multi-objective genetic algorithm (MOGA), an optimization tool that has recently been proven to be fast and effective in the optimization of a MCHS [7,9-13].

2.0 METHODOLOGY

2.1 Mathematical Model

There are several approaches to analyze the performance of a MCHS. Previous studies showed that methods that had been used consisted of the thermal resistance model [7], porous medium model [4] and also three-dimensional numerical model [6]. The thermal resistance model is chosen to analyze the performance of a MCHS in this study with CNT as the working fluid. The thermal resistance model is chosen to evaluate the performance of a MCHS because of its simplicity and accuracy. A previous study by Liu and Garimella [14] showed that the thermal resistance model provided an acceptable result compared with the fin, two fin-liquid coupled and porous medium models.

A MCHS consists of channels and fins confined by an adiabatic cover plate on top. There are several parameters to be taken into account to carry out the analysis. The parameters are its length, width, channel height, substrate thickness, channel width and wall width denoted by L , W , H , H_c , t , w_c , and w_w respectively. The parameters described are represented in Figure 1 and Figure 2.

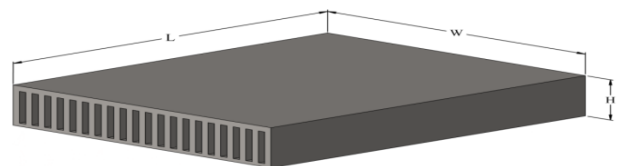


Figure 1 A schematic diagram of a MCHS

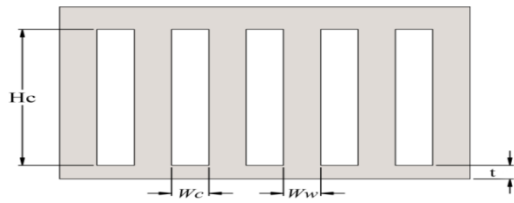


Figure 2 Closed up view of a MCHS

The thermal resistance model utilizes the physics of the thermal resistance network to solve for the thermal performance of a MCHS. Figure 3 illustrates an overview of the thermal resistance network involved in series.



Figure 3 The thermal resistance involved

The thermal resistance network is based on Wen and Choo [15]. In this study, the approach offered by Adham *et al.* [9] was considered in evaluating the performance of a MCHS. The thermal resistance R1, R2, R3 and R4 represent conduction, constriction, convection and capacitive respectively. The conduction thermal resistance, R_{cond} , occurred at the substrate region. As the heat travels up to the base fin of the MCHS it experiences constriction thermal resistance, R_{cons} . Then, there is the convection thermal resistance, R_{conv} , from the base and the wall of the fin. For simplicity of the evaluation, both the areas involved are defined as an effective area. Finally, the heat is carried away by the coolant. As the heat is carried away, it experiences capacitive thermal resistance, R_{capa} . These four elements are very important in carrying out the analysis based on the thermal resistance model.

In the optimization process, the design variable, α and beta, β plays a major role to determine the overall performance of a MCHS. The definition of these two variables are described as follows:

$$\alpha = H_c / w_c$$

$$\beta = w_w / w_c$$

For the thermal resistance model, according to Choquette *et al.* [16], the total thermal resistance of a MCHS is the ratio between the temperature difference of the maximum surface temperature and the temperature of the inlet coolant to the heat flux applied. Figure 4 shows the relationship between the temperature of the surface and the coolant.

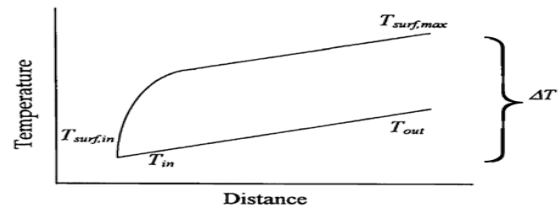


Figure 4 The temperature of the surface and coolant

As shown in Figure 4, the total thermal resistance of a MCHS can be expressed as

$$R_{total} = \frac{T_{surface,max} - T_{inlet}}{q} \quad \left(\frac{^{\circ}\text{C}}{\text{W}} \right) \quad (1)$$

where $T_{surface,max}$ is the maximum temperature of the substrate, T_{inlet} is the initial temperature of the coolant and q is the heat flux. The total resistance in a MCHS is contributed by the conductive, convective, capacitive and also constrictive thermal resistance and it can be expressed as

$$R_{total} = R_{cond} + R_{conv} + R_{capa} + R_{cons} \quad \left(\frac{^{\circ}\text{C}}{\text{W}} \right) \quad (2)$$

The thermal resistance for conduction due to the substrate of a MCHS is a function of the thickness of the substrate, thermal conductivity of the material and the area experienced by the heat flux. It can be expressed as

$$R_{conduction} = \frac{t}{k_{hs}(WL)} \quad \left(\frac{^{\circ}\text{C}}{\text{W}} \right) \quad (3)$$

where t is the thickness and k_{hs} is the thermal conductivity of the substrate. The thermal resistance for convection is a function of the heat transfer coefficient and also the effective area. It can be expressed as

$$R_{convection} = \frac{1}{h_{av}(WL)} \frac{1 + \beta}{1 + 2\alpha\eta} \quad \left(\frac{^{\circ}\text{C}}{\text{W}} \right) \quad (4)$$

where h_{av} is the convective heat transfer coefficient and η is the fin efficiency whereby the channel thin walls are treated as fin. In the analysis of the heat transfer coefficient, the Nusselt number is given by

$$Nu = \frac{h_{av} D_h}{K_f} \quad (5)$$

where K_f and D_h are the thermal conductivity of the fluid and the hydraulic diameter respectively. At this point only the parameter K_f is obtained from the properties of the fluid. For the evaluation of the Nusselt number, Nu is obtained from Kim and Kim [17].

The correlation proposed is valid for rectangular geometry and in laminar flow.

$$Nu = 2.253 + 8.164 \left(\frac{\alpha}{\alpha + 1} \right)^{1.5} \quad (6)$$

The volumetric flow, G , for n channels is given by

$$G = nH_c w_c V \quad \left(\frac{m^3}{s} \right) \quad (7)$$

where V is the velocity of the coolant inside the MCHS. The working fluid to carry the heat away experiences capacitive thermal resistance. It is described by

$$R_{capacitance} = \frac{1}{\rho_f C_{pf} G} \quad \left(\frac{^\circ C}{W} \right) \quad (8)$$

Meanwhile, heat from the base of the fin experiences constrictive thermal resistance [18]. It is described by

$$R_{constrictive} = \frac{w_w + w_c}{\pi k_{hs} (WL)} \ln \left[\frac{1}{\sin \frac{\pi w_w}{2(w_w + w_c)}} \right] \quad \left(\frac{^\circ C}{W} \right) \quad (9)$$

In terms of the hydrodynamic performance, which is also investigated here, it is evaluated by using the pressure drop and pumping power [9]. The total pressure drop across the MCHS can be expressed as

$$\Delta P_{total} = F1 + F2 \quad (\text{Pa}) \quad (10)$$

$$F1 = f_{hs} \frac{(1 + \alpha)L}{2H_c} \rho_{nf} \frac{V^2}{2}$$

$$F2 = \left(1.79 - 2.23 \left(\frac{1}{1 + \beta} \right) + 0.53 \left(\frac{1}{1 + \beta} \right)^2 \right) \rho_{nf} \frac{V^2}{2}$$

f_{hs} is the friction factor for laminar flow. The correlation can be described as

$$f_{hs} = \frac{64}{Re} \quad (11)$$

In terms of the evaluation for the pumping power required to drive the coolant, the equation involved is given by

$$P_p = \Delta P_{total} \times G \quad (\text{W}) \quad (12)$$

Minimization of equations (2) and (12) are the two objectives to be achieved in this optimization but a decrease in the former is followed by an increase in the latter. Thus, optimization is crucial in obtaining a balance outcome from both objectives.

2.2 Mathematical Model Validation

A parametric study is carried out first for the total thermal resistance and pressure drop and the outcomes compared with the experimental data from Tuckerman and Pease [1]. In the current research, the evaluation of the total thermal resistance omitted the constrictive thermal resistance in order to compare with the previous study [1]. The properties of CNT nanofluid used are given in Table 1 [8]. Results from the current model are tabulated in Table 2. It shows that the current model exhibits only slight disparities compared with the experimental results from Tuckerman and Pease [1] with 9% for thermal resistance and 3% for pressure drop. The difference might be in terms of the different approach used in both models. Besides that, the working fluid used in the previous study was deionized water while for the current study properties of common water were used. The difference in terms of thermophysical properties of the working fluid does affect both the thermal and hydrodynamic performance of a MCHS.

Table 1 Thermophysical properties of CNT nanofluid

T(°C)	k(W/m.K)	ρ (kg/m ³)	C_p (J/kg.K)	μ (Ns/m ³)
20	0.5907	1001.119	4178.663	0.001252
30	0.6317	999.720	4178.663	0.0009969
40	0.6531	996.7212	4177.663	0.0006907

2.3 Optimization Procedure

The method of the optimization is based on Adham *et al.* [9]. First of all, the objective function has to be clearly stated. Here the thermal performance and hydrodynamic performance are being evaluated which are reflected by the lowest value of the total thermal resistance and pumping power. The main variables in the current analysis are α and β . In the optimization using genetic algorithm, the population is initialized at the preliminary step. Population initialization is very important as it will affect the out where selected chromosomes are reproduced for the next population. Finally, the solution for the objective function is generated. Many attempts must be completed to determine the most appropriate combinations of population size, percentage each for cross-over, mutation and selection. Generalization for the GA approach and its parameter involved is described in Figure 5.

2.3.1 Optimization Validation

The thermal resistance and the pumping power is first optimized by using water under the same operating conditions as Tuckerman and Pease [1] and the results are tabulated in Table 3. This validation is necessary to prove that the model and algorithm are

accurate. The current optimization that had been carried out clearly shows a reduction of total thermal resistance by 15%, 24% and 32 % for channel heights of 320, 302 and 287 micrometers respectively.

The reduction in the total thermal resistance proves that the genetic algorithm approach can be used to determine the lowest thermal resistance with

the optimum design variable conditions. The ability to determine the lowest value at a fast rate is very convenient compared to the parametric study where the evaluation needed to be done involves repeated discrete variations.

Table 2 Parametric study of current model against Tuckerman and Pease [1]

Models	Parameters									Relative uncertainty	
	W (cm)	L (cm)	H _c (mm)	α	β	G (cm ³ /s)	R (°C/W)	ΔP (psi)	Pumping power (W)	R (%)	ΔP (%)
Tuckerman and Pease [1]	1	1	0.32	5.714	0.7857	4.7	0.11	15	0.49	-	-
Current model	1	1	0.32	5.714	0.7857	4.7	0.10	13	0.42	9	3

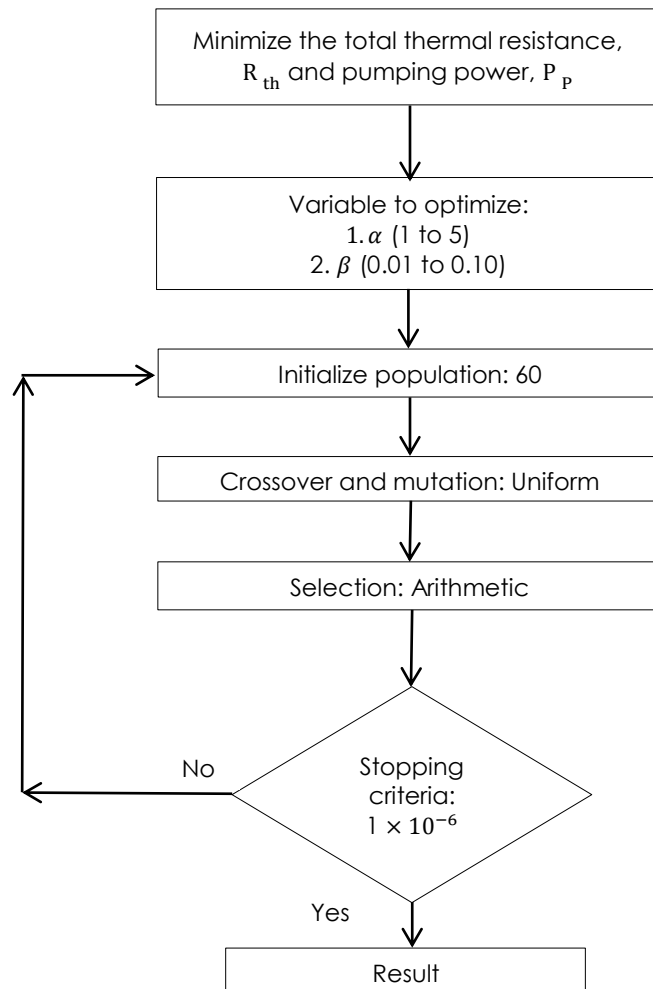


Figure 5 Complete optimization procedure for current study (adapted from [10])

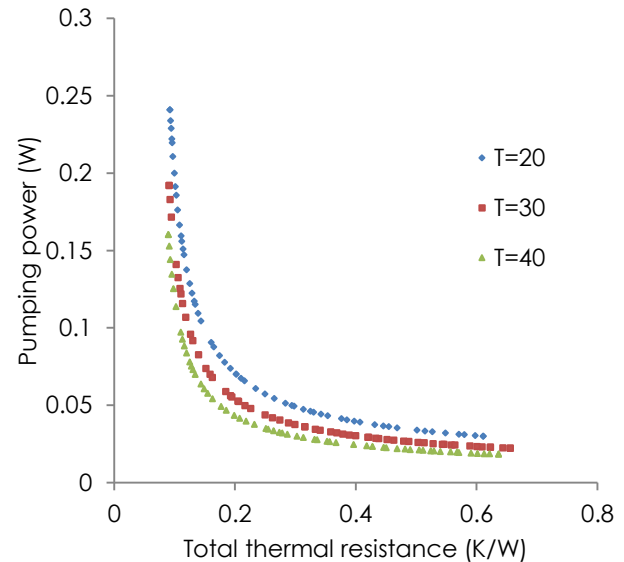
Table 3 Comparison of the current result with [1]

		Tuckerman and Pease [1]				Optimization result (Current study)				
G (cm ³ /s)	H _c (mm)	α	β	R(°C/W)	P _p (W)	α	β	R(°C/W)	P _p (W)	$\frac{\Delta R}{R}$ (%)
4.7	320	5.71	0.78	0.11	0.486	4.937	0.0312	0.093	0.184	15
8.6	302	6.01	1	0.09	1.838	4.9472	0.0426	0.068	0.7418	24
6.5	287	5.218	0.818	0.113	0.762	4.8332	0.224	0.077	0.4674	32

3.0 RESULT AND DISCUSSION

In this research, a volume concentration of 0.1% for CNT nanofluid is being used and optimization outcomes compared with water. The optimized results are being compared at three different operating temperatures, selected based on [7], and they are tabulated in Table 4. The results obtained clearly show that at a high operating temperature of 40°C, this particular CNT nanofluid has a low total thermal resistance compared to water. The reduction of total thermal resistance for CNT nanofluid is 3% from the temperature of 20°C to 40°C. Meanwhile, for water the reduction in thermal resistance is 2%.

The relationship between pumping power and the thermal resistance at different operating temperatures is presented in Figure 6. It can be inferred that as the operating temperature increases, the total thermal resistance decreases. The lowest thermal resistance is accompanied with the highest pumping power. As the operating temperature increases, the thermal conductivity of the CNT nanofluid increases. The increment of the thermal conductivity enhances the heat transfer coefficient. The enhancement of the heat transfer coefficient plays a major role in reducing the convective thermal resistance. Besides thermal conductivity, the density of the fluid decreases as the temperature increases. Therefore, it follows that at a temperature of 40°C the pumping power is at the lowest value compared to 20°C and 30°C.

**Figure 6** Optimized pumping power against total thermal resistance at different operating temperature

In geometrical properties, the main parameters that are being discussed are ratio of channel height to width channel, α , and ratio of wall width to channel width, β . The effects of both parameters towards the total thermal resistance and pumping power are presented in Figures 7 through 10. The effect of α towards the total thermal resistance at 40°C is presented in Figure 7. It is clearly shown that as α increases, the total thermal resistance decreases. The highest channel aspect ratio results in the lowest thermal resistance for both CNT nanofluid and water. The trend of the graph shows that at low α , the total thermal resistance for CNT nanofluid is lower compared to water, 1% lower at 40°C. The total thermal resistance keeps decreasing significantly until it reaches a value of 3.5. After that, the total thermal resistance only drops slightly. Notice that the variable α reaches the optimum condition as it approaches near the value of 5.0.

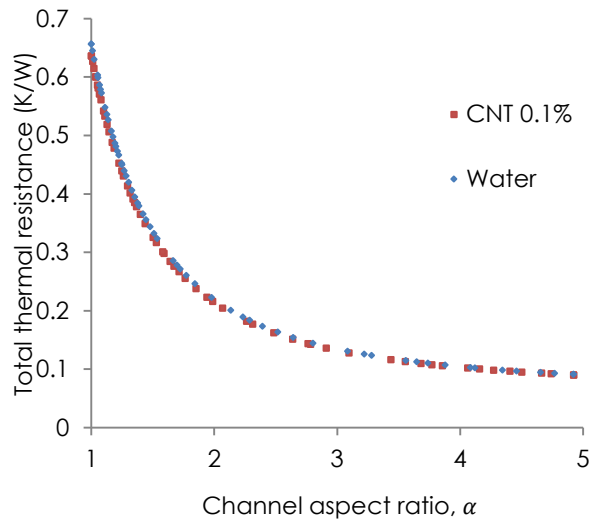


Figure 7 Optimized thermal resistance against channel aspect ratio

The optimum design variable is necessary to determine the lowest thermal resistance in a MCHS. Any increment of α does not give much difference in lowering the total thermal resistance. α is the ratio of channel height over channel width. Since the height of the channel is kept constant throughout the process, the increment of channel aspect ratio leads to the decrement of channel width. The decrement of the channel width yields the reduction of the convective thermal resistance. Since convective thermal resistance plays a major impact on the total thermal resistance, the design variable α will give a major impact towards the reduction of the total thermal resistance.

Table 4 Current optimization result for CNT nanofluid and water

Optimized result		CNT nanofluid 0.1%			Water		
		T = 20°C	T = 30°C	T = 40°C	T = 20°C	T = 30°C	T = 40°C
H _c =320 micrometer	R(°C/W)	0.093	0.092	0.090	0.093	0.092	0.091
	P _p (W)	0.246	0.187	0.159	0.196	0.155	0.127
	α	4.952	4.863	4.943	4.972	4.94	4.953
	β	0.026	0.011	0.018	0.018	0.018	0.011

The graph for α against pumping power is presented in Figure 8. As the value of α increases, the pumping power also increases. The system that implements CNT nanofluid as the working fluid experiences a higher pumping power compared to water. This is because the CNT particles increase the density of the fluid which requires a higher power to drive the coolant. The highest pumping power yielded by the system is when α is near 5.0. At 5.0, the channel width is the smallest. This explains the highest value of pumping power because of the narrow channel to drive the coolant through. Unlike in Figure 7, the pumping power does not reach a steady state after the value of 3.5

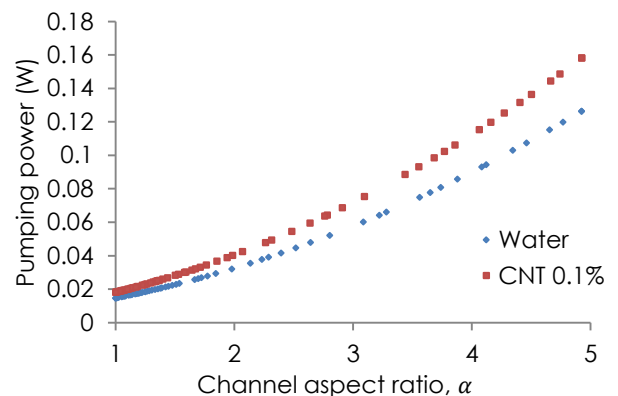


Figure 8 Optimized pumping power against channel aspect ratio

Effect of β on the total thermal resistance is presented in Figure 9. As β increases, the total thermal resistance decreases. This behavior shows a similar trend with Figure 7 with the drop in total thermal resistance occurred at low value of β . The lowest thermal resistance occurred when β is near 0.022. The value of total thermal resistance drops slightly as the value of β passes the value of 0.017. The reason for this small value of β is that the MCHS is more sensitive towards α rather than β . Hence, the

value of β reaches an optimum state at a low value of β . For a particular channel width, the increment of β will decrease the channel width. Reduction in channel width gives effect towards the convective thermal resistance.

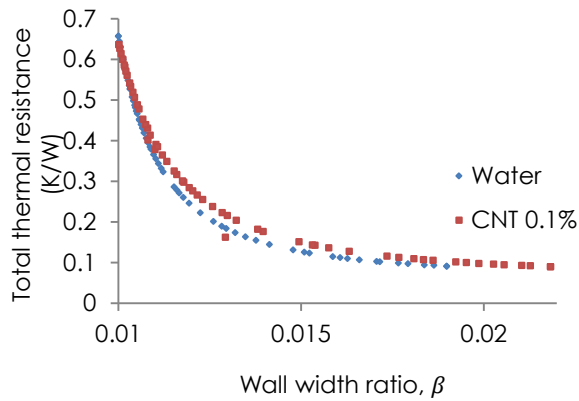


Figure 9 Optimized total thermal resistance against wall width ratio

Meanwhile, the effect of β towards the pumping power is displayed in Figure 10. CNT nanofluid experiences a higher pumping power compared to water. The trend exhibits a similar pattern with Figure 8 which is when the wall width ratio increases, the pumping power also increases. At a particular wall width, the increasing of β will result in a narrow channel. Therefore, a higher pumping power is required to drive the coolant.

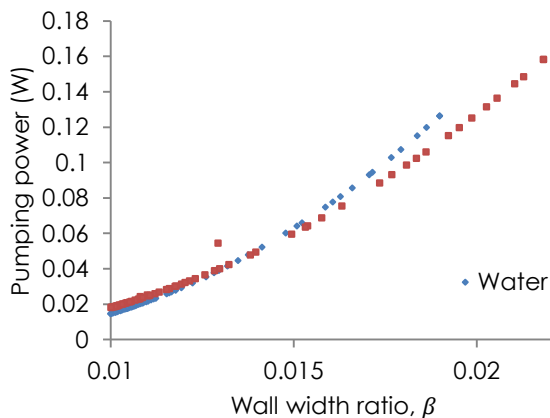


Figure 10 Optimized pumping power against wall width ratio

4.0 CONCLUSION

The effects of the implementation of 0.1% volume concentration of a particular CNT nanofluid, this with an average diameter, length, density, and carbon purity of 9.2 nm, 1.5 μm , 1800 kg/m^3 , 90% respectively, as a coolant in a MCHS has been presented. In general, the optimized total thermal resistance for

the MCHS for 0.1% volume concentration of the CNT nanofluid is much lower compared to water. Besides that, at high operating temperature the optimized total thermal resistance of the CNT nanofluid at 40°C is lower than that of 20°C by 3%. Since MCHS does operate at high operating temperature, the implementation of the CNT nanofluid as the working fluid is an attractive approach to remove the heat and thus provide an effective method for the removal of high heat flux. Due to the different characteristics of different CNT nanofluids, accurate property data is important and generalization should be avoided for misrepresentation of the expected performance of any particular CNT nanofluid.

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