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Thermal Behaviour of Hybrid Nanofluids in Water: Bio Glycol Mixture in Cooling Plates of PEMFC

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ABSTRACT

Proton Exchange Membrane Fuel Cell (PEMFC) is a renewable technology application for vehicle power sources that is a viable challenger for a safe and efficient power generation. Nanofluids adoption is one of the advancements in PEMFC heat management. In addition to that, Bio Glycol is also introduced as a non-toxic, renewable fluid with 30% lower viscosity than regular petroleum-derived propylene glycol at low temperatures. The 0.5% volume concentration of hybrid nanofluids of Aluminium Oxide (Al_2O_3) and Silicon Dioxide (SiO_2) in water:BG at 60:40 volume ratio was investigated in this work. This paper investigated the heat transfer improvement and the pumping power effect in mini channel of PEMFC distributor and serpentine cooling plate with the adoption of hybrid Al_2O_3 and SiO_2 nanoparticles in water: Bio Glycol (BG). The simulation conducted using ANSYS Fluent, under laminar region of 300 to 1800 and constant heat flux of 6500 (W/m^2) to imitate the heat generation in a PEMFC bipolar plate. The $\mathrm{Al_2O_3}:\mathrm{SiO_2}$ ratios used were 10:90, 30:70, 50:50, and 70:30. The findings suggested that $\mathrm{Al_2O_3}:\mathrm{SiO_2}$ (30:70) in water: BG provides the highest improvement of 14.4% in the serpentine cooling plate and 20.9% in the distributor cooling plate at Re 1800. However, the pressure drop for both plates was increased up to 7 times greater than the base fluid. The advantage ratio was then calculated to assess the feasibility of nanofluids in PEMFC cooling plates. As a conclusion it was recommended that the serpentine cooling plate outperforms the distributor cooling plate in terms of both heat transfer and pumping power need.

1. Introduction

Global warming is a severe environmental issue with the potential to harm human and other living things. To solve the challenges, renewable energy supplies are being investigated to reduce the demand for fossil fuels. New energy sources have been investigated to prevent pollution and the depletion of natural fuel resources. Fuel cell is among the popular prospective energy that utilises hydrogen as the energy carrier in supplying power systems, transportation, thermal, industrial, and electrical services. Fuel cells plays a crucial role in the future low-carbon economy [1]. The fuel cell, according to Sharaf and Orhan [2], is a electrochemical device that transforms the chemical energy of a fuel into electrical energy. It has three active components: an anode that serves as a fuel electrode,

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a cathode that serves as an oxidant electrode, and an electrolyte. There are several types of fuel cell namely Solid oxide fuel cell (SOFC), molten carbonate fuel cell (MCFC), phosphoric acid fuel cell (PAFC), proton exchange membrane fuel cell (PEMFC), solid polymer membrane fuel cell (SPFC), and alkaline fuel cell (AFC) [3]. The maximum conductivity of the electrolyte material occurs only within specified temperature range thus optimal thermal management of fuel cell is critical in order to gain the maximum performance of the membrane.

PEM Fuel Cell is a form of fuel cell that uses oxygen and hydrogen to generate electricity and heat as a by-product. Because of its high conversion efficiency and environmental friendliness, it has a lot of advantage as an internal combustion engine (ICE) substitute. Their poor heat transfer rate was recognised as a barrier to enhance the fuel cell performance. The produced heat must be removed from the device, particularly the membrane, to prevent overheating. The new PEMFC requires a temperature range of 60 °C to 80 °C to function optimally [4-6]. The PEMFC can work at extremely low temperatures, which is the most important property of these cells since it allows them to give higher power density at these temperatures [7]. Thermal performance of nanofluids used in radiators that implemented in automotive PEMFC application has been recently studied by Bargal M *et al.,* [8].

Designing more compact stacks with conventional coolant of water- cooling systems for PEMFC stacks will require a bigger size of heat exchanger due to the small temperature difference [9] Numerous cooling systems have been developed to address this issue, with the air and liquid cooling system being the most employed. Liquid cooling is more effective than air cooling because it provides better heat transfer capabilities when compared to air [1, 10]. The cooling plates are used in a liquid cooling system between PEMFC cells to remove heat generated by the flowing coolant. Conventional coolants used is water and water: Ethylene glycol mixture. Currently, research was aggressively conducted on the use of nanofluids coolants to improve heat transmission and, as a result, reduce the number of cooling plates in the future for a more compact stack design [5, 11].

Nanofluids are regarded as the most promising choice for an improved coolant for PEMFCs [12]. It can increase the thermal conductivity of the base fluid by spreading nanoparticles inside a base fluid which has a high thermal conductivity and increasing the total surface area of the particles. In comparison to the conventional PEMFC cooling system, the nanofluids in PEMFC is effective in terms of both heat transmission and cooling system simplicity by lowering the size of the radiator and perhaps removing the deionizer [5]. Nanofluids were initially implemented to increase coolant durability as the electrical conductivity requirement in PEMFC is very strict [13]. The preparation of nanofluids is critical in enhancing the thermo-physical characteristics of the base fluid. This is the first and most important step in experimenting. The researchers favour the two-step procedure since it is cost effective and can produce large-scale hybrid nanofluids [14]. However, according to Kumar *et al.,* [15], the stability of nanoparticle suspension in based fluids is more effective when nanofluid is prepared in a single step as compared to a two-step procedure.

It has also been discovered that Bio Glycol (BG) base fluid as an alternative glycol replacement which is derived from agricultural product is a non-toxic, with a viscosity 30% lower than propylene glycol [16]. However, Bio Glycol (BG) outperformed water by having a substantially lower freezing point and higher boiling point (46 °C to 177 °C). In addition, among other benefits of BG is that it has Furthermore, it also has better thermo-physical characteristics such as higher thermal stability as compared to propylene and ethylene glycols. It outperforms propylene glycol in terms of performance while being safer for the environment than ethylene glycol [17, 18].

The combination of both hybrid nanofluids in a greener base fluid of BG is as PEMFC coolants is currently not openly available in the literatures. Therefore, the study investigated the improvement of heat transfer and the pressure drop penalty caused by the use of hybrid nanofluids Al_2O_3:SiO_2 in 60:40 W:BG mixtures in a distributor and serpentine mini channel of a PEM fuel cell. The study was done numerically and throughout the simulation, the mixing ratios of the hybrid nanofluids were varied, and the values of Re number were varied from 300 to 1800 an increment of 300. The hybrid nanofluids are expected to outperform single nanofluids [19, 20].

2. Methodology

2.1 Properties of Fluid and Nanoparticles Used in the Analysis

Thermophysical properties of nanofluids are critical for predicting their heat transfer performance. In comparison to typical particles fluid suspension, millimetre and micrometre sized particles, nanoparticles have a huge potential to improve thermal transfer capabilities [21]. The simulated properties were referred based on few research studies. The hybrid of Al_2O_3 : SiO_2 was chosen to add variants of this hybrid ratios in different base fluid studies [22]. Furthermore, this hybrid nanofluids were proven for their superior thermal conductivity, stability, low cost, commercially available, with minimal effect of sedimentation. Table 1 displays the thermophysical characteristics of the nanoparticles and based fluid utilised in the study.

Table 1
Thermophysical properties of the substances

Substances	Density, ρ (kg/m^3)	Specific Heat, Cp (J/kg.K)	Thermal conductivity, k (W/m.K)	Viscosity, μ (mPa.s)	References
Water	999	4180	0.615	0.854	[23-25]
Al_2O_3	4000	765	36	-	[24][12]
SiO_2	2220	745	1.4	-	[25]

2.2 Mathematical Model Used in Mini Channel Cooling Plates

Ansys Fluent 16 was used to perform the numerical analysis for a serpentine and distributor cooling plate of a PEMFC. Hence, some assumptions have been made to facilitate the study:

i. The fluid is in steady-state incompressible and laminar flow.
ii. The dissipation of viscous fluid is neglected, and the fluid properties remain constant.
iii. Fluid phase and nanofluids have zero relative velocity in thermal equilibrium. The resulting combination is classified as a convectional single phase.
iv. All mini channels have the same heat transfer and fluid flow characteristics.

The following are the governing equations based on the preceding assumptions [1].

The continuity formula:

$$\nabla.\left(\rho_{nf}.V_m\right) = 0 \quad (1)$$

The momentum equation is as follows:

$$\nabla.(\rho_{nf}.V_m.V_M) = -\nabla P + \nabla.\left(\mu_{nf}.\nabla V_m\right) \quad (2)$$

Equation for coolant energy:

$$\nabla.(\rho_{nf}.C.V_m.T) = \nabla.\left(k_{nf}.\nabla T\right) \quad (3)$$

Conduction of heat through a solid wall:

$$0 = \nabla.(k_s.\nabla T_s) \quad (4)$$

The wall has a no-slip boundary:

$$\vec{V} = 0\ (@walls) \quad (5)$$

Boundary condition at channel inlet were assumed as:

$$\vec{V} = V_m\ (@inlet) \quad (6)$$

$$P = atmospheric\ pressure\ (@outlet) \quad (7)$$

The heat is transmitted through the solid and dispersed via the serpentine and distributor cooling plate by forced convection of fluid. With a steady heat flow, the bottom surface is uniformly heated.

$$-k_{nf}.\nabla T = q\ (at\ bottom\ of\ cooling\ plate) \quad (8)$$

$$-k_{nf}.\nabla T = (at\ top\ of cooling\ plate) \quad (9)$$

2.3 Plates Heat Transfer and Fluid Flow Analysis

The heat transfer coefficient and Nu number were used to analyse heat transfer [26]. The inlet velocity value was determined using equation:

$$v = \frac{Re\mu}{\rho D} \quad (10)$$

Where v is the inlet velocity, Re is the reynolds number, μ is the dynamic viscosity, ρ is the density and D is the diameter. The heat transfer coefficient and the Nusselt number were determined as:

$$h = \frac{q}{T_p - \left(\frac{T_i - T_o}{2}\right)} \quad (11)$$

Where h is the heat transfer coefficient, q is the heat flux, T_p is the average plate temperature, T_i is the inlet temperature and T_i is the outlet temperature of the cooling plate.

$$Nu = \frac{hD_i}{k} \quad (12)$$

Where Nu is the nusselt number, D_i represent inlet diameter and k represent thermal conductivity.

The pressure drop is expressed as:

$$\Delta p = P_i - P_o \tag{13}$$

Where Δp, P_i and P_o is the pressure drop difference, inlet pressure and the outlet pressure respectively. The equation of pumping power was determined by using this equation:

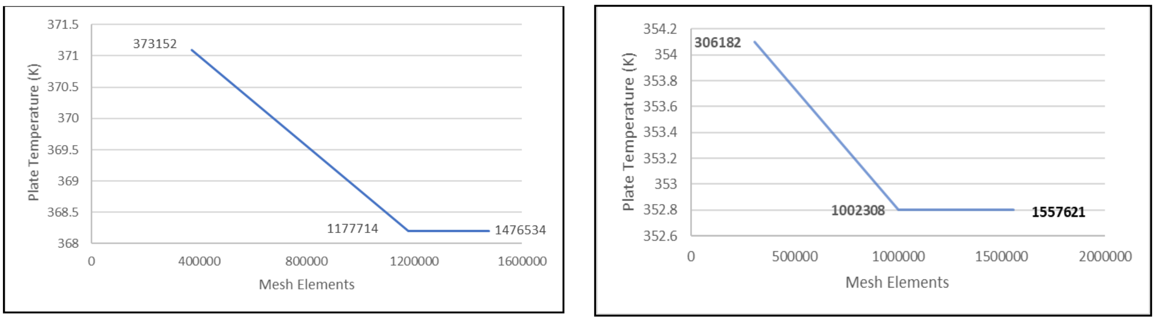
$$W_{pump} = \dot{Q} \times \Delta p \tag{14}$$

Where $\dot{Q}$ is the calculated volume flow rate. Advantage Ratio was calculated as:

$$AR = \frac{h}{\Delta p} \tag{15}$$

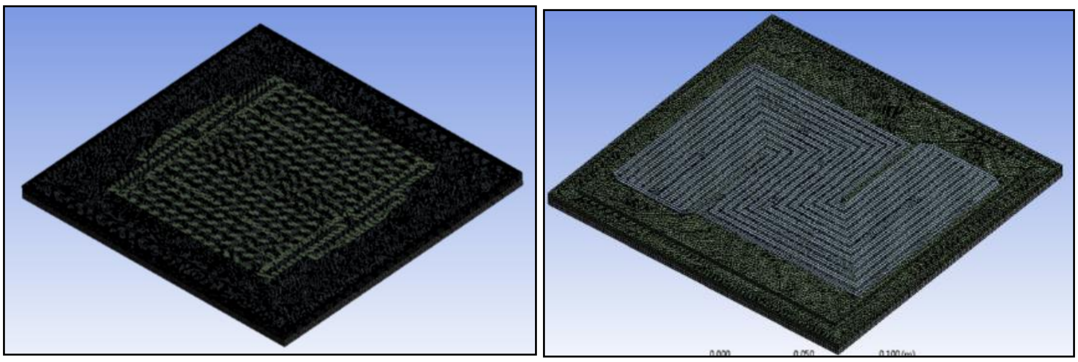
2.4 Grid Independent Test

The grid independent test was performed to ensure that the meshing element utilised was the optimum meshing, since a greater number of meshing elements increases simulation time, but a lesser number of meshing elements or inadequate meshing results in an inaccurate simulation result [27]. As indicated in the Figure 1 below, the ideal meshing size for analysis is 1177714 mesh elements for a distributor and 1002308 mesh elements for a serpentine cooling plate. Figure 2 showed the meshing for both serpentine and distributor cooling plates of PEMFC.



(a) Distributor (b) Serpentine

Fig. 1. Graph of Grid Independent Test



(a) Distributor (b) Serpentine

Fig. 2. Geometry of Meshing for both Cooling Plates

3. Results and Discussion

3.1 Average Plate Temperature

The average plate temperature serves as the basic findings in this study. The average plate temperature was shown in Figure 3. It was observed that as the reynold number was increased, the the average plate temperature for both cooling plates reduced as well. The maximum reduction of plate temperature was experienced by Al_2O_3 : SiO_2 (30:70) hybrid nanofluids at Re 1800 with reduction of by 1.23 % and 2.15 % in serpentine and distributor plate respectively as compared to base fluid of W:BG (60:40). The hybrid ratios of Al_2O_3 : SiO_2 (50:50) was also 1.07 % and 1.96 % reduction from the base fluids in both serpentine and distributor plate. The Al_2O_3 : SiO_2 (70:30) and Al_2O_3 : SiO_2 (10:90) hybrid nanofluids performed equally as Al_2O_3 and SiO_2 single nanofluids. The base fluid has the highest plate temperature. This is expected due to the superior thermal conductivity of hybrid nanofluids over single nanofluids and base fluid, the cooling plate with hybrid nanofluids has a lower average plate temperature than single nanofluids and base fluid. The findings agreed well with studies by Zakaria *et al.,*[22]. The research showed that the thermal conductivity value was highest in Al_2O_3 : SiO_2 (30:70) hybrid nanofluids followed by Al_2O_3 : SiO_2 (50:50) hybrid nanofluids and the rest of the ratio.

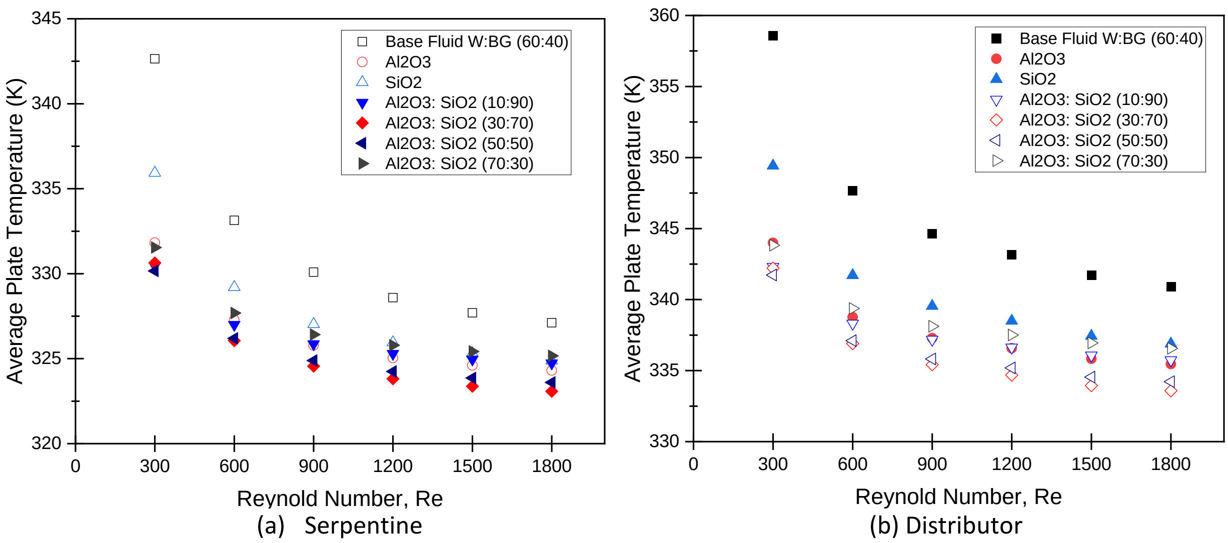
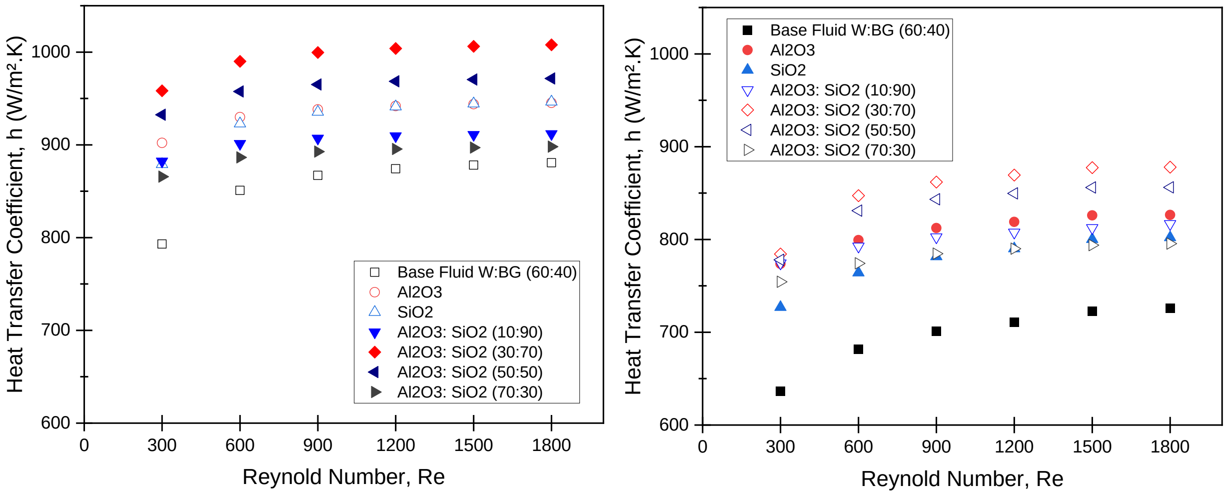


Fig. 3. Graph of Average Plate Temperature against Re for both Cooling Plates

3.2 Heat Transfer Coefficient

The plate temperature was then further investigated to get the heat transfer coefficient value. Figure 4 depicts the heat transfer coefficient for all fluids investigated for Serpentine and Distributor cooling plates. It is shown that the heat transfer coefficient of Al_2O_3 : SiO_2 (30:70) hybrid nanofluids has the highest heat transfer coefficient value with 14.4% in serpentine cooling and 20.9% in distributor cooling plate as compared to base fluid. This demonstrated that hybridization has significantly increased the cooling fluid's heat transfer coefficient. The findings are in good agreement with findings of Idris *et al.,* [26] and Zakaria *et al.,* [5]. It is shown that the distributor cooling plate shows better thermal management than the serpentine cooling plate in terms of heat transfer requirement.

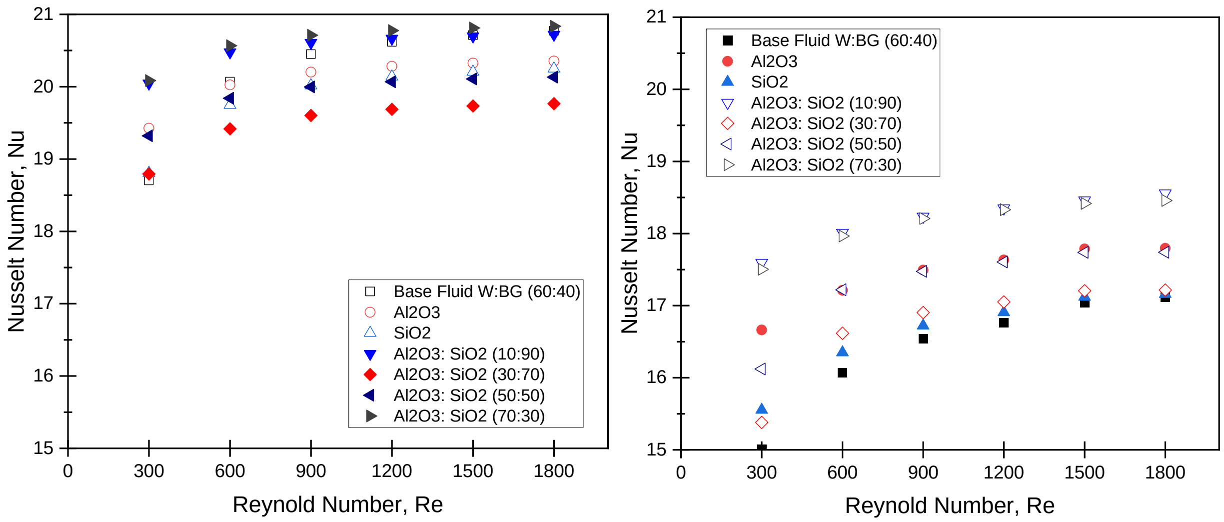


(a) Serpentine (b) Distributor

Fig. 4. Graph of Heat Transfer Coefficient for both Cooling Plates

3.3 Nusselt Number

The convective heat transfer coefficient was then analysed to derive the non-dimensionalized Nusselt number as shown in Figure 5. In general, the Nusselt number increased linearly as the Reynolds number increased. The nusselt number was also observed to be higher in serpentine as compared to the distributor cooling plate. This was due to greater convective heat transmission effect as compared to the conductive heat transfer. In serpentine cooling plate, the greatest Nusselt number was obtained for $Al_2O_3 : SiO_2$ (70:30) hybrid nanofluids with 7.4% enhancement as compared to base fluid whereas the lowest was reported for the $Al_2O_3 : SiO_2$ (30:70) hybrid nanofluids with 0.5% enhancement. Meanwhile, as for the distributor cooling plate, it shows that $Al_2O_3 : SiO_2$ (10:90) hybrid nanofluids has the highest Nusselt Number with 8.4% enhancement as compared to base fluid of water:BG. The lowest Nusselt was given by base fluid of w:BG. This agrees to the effect of higher value of thermal conductivity dominates the heat transfer in distributor as compared to convective heat transfer thus giving a lower Nusselt nu values as compared to serpentine cooling plate.



(a) Serpentine (b) Distributor

Fig. 5. Graph of Nusselt Number for both Cooling Plates

3.4 Pressure Drop

The pressure differential between the input and outlet fluids was used to assess the fluid flow of nanofluids in a cooling plate, as illustrated in Figure 6. The highest pressure drop was observed at Reynolds number 1800 for Al_2O_3 : SiO_2 (10:90) hybrid nanofluids with up to 7 times higher for both distributor and serpentine cooling plates when compared to base fluid. The base fluid of water:BG had the lowest pressure drop. A high-pressure drop was predicted in nanofluids since the coolant has higher density and viscosity which makes it harder to flow through the small channels of the cooling plate. Hybrid nanofluids have a greater viscosity value than single nanofluids and base fluid, resulting in a much larger pressure drop. This was well aligned with findings by Khalid *et al.,* [12].

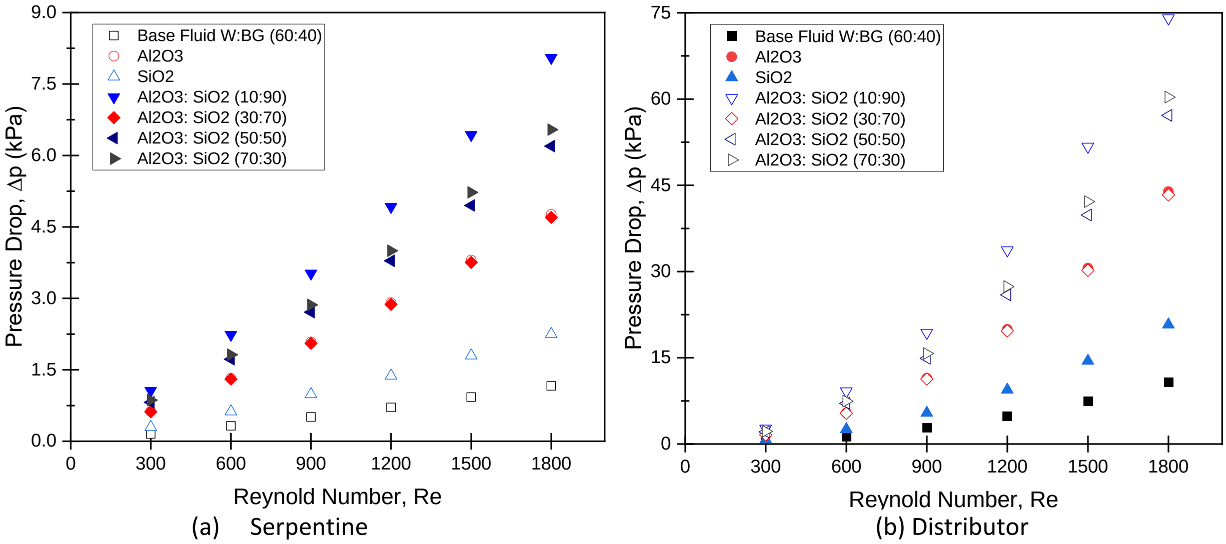
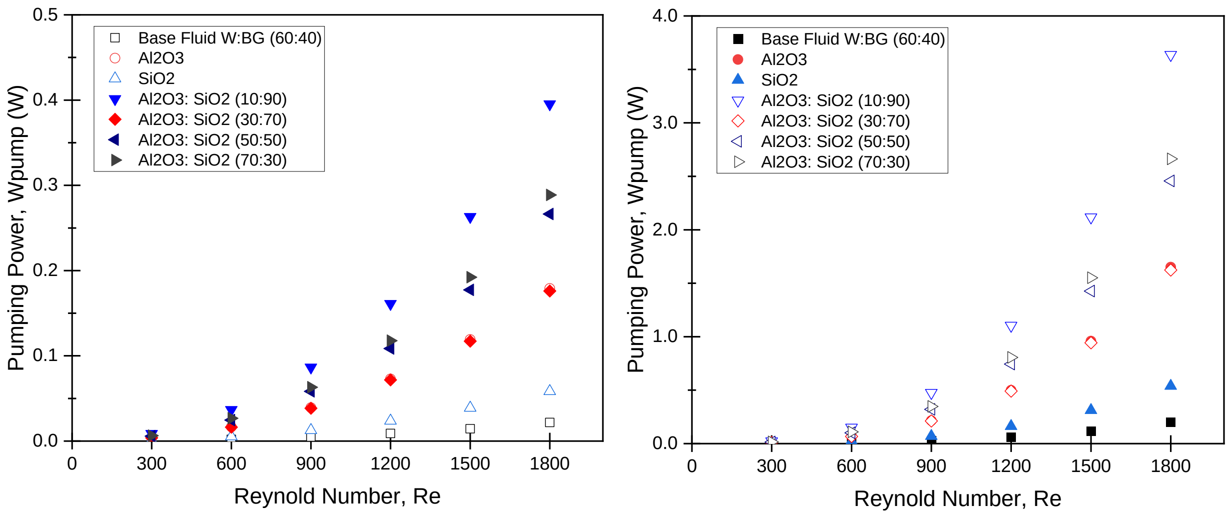


Fig. 6. Graph of Pressure Drop for both Cooling Plates

3.5 Pumping Power

The pressure drop readings were then converted to pumping power data to determine the influence of nanofluids on a system's pumping requirements. Figure 7 depicts the pumping power data for all fluids investigated. The Al_2O_3 : SiO_2 (10:90) hybrid nanofluids demonstrated the greatest pumping power requirement with 3.6 W as compared to water:BG of 0.2 W at Re 1800 for distributor cooling plate. Meanwhile, 0.4 W for Al_2O_3 : SiO_2 (10:90) hybrid nanofluids was recorded as the highest pumping power requirement for serpentine cooling plate while the lowest one was water:BG of 0.02 W. This was then followed by a different ratio of hybrid nanofluids to single nanofluids. The pumping power requirement was 9 times higher in distributor plate as compared to serpentine plate. It was also known that the pumping power value increased as the Reynolds number was increased. The inclusion of nanoparticles has resulted in increased internal friction and flow resistance, which eventually raises the necessary pumping power [27]. The improvement was intended to allow more viscous nanofluids to circulate through the cooling system. Higher pumping power is less desirable since it results in a parasitic loss. However, further judgement on the increase in parasitic losses need to be performed in order to find the feasibility of the adoption.

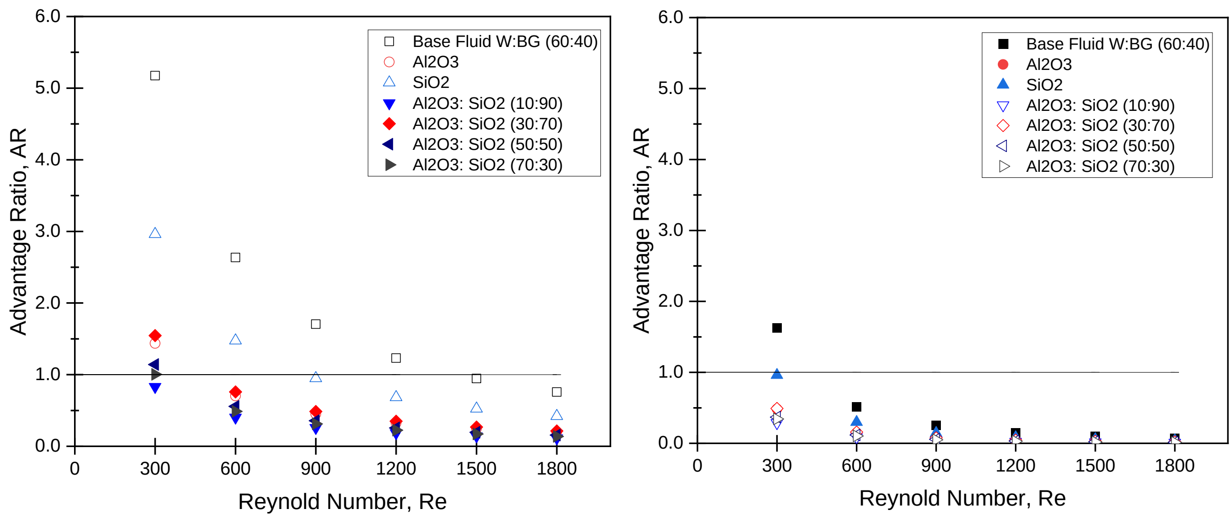


(a) Serpentine (b) Distributor

Fig. 7. Graph of Pumping Power for both Cooling Plates

3.6 Advantage Ratio

The advantage ratio was used to assess the feasibility of using hybrid nanofluids in PEMFC cooling. The advantage ratio weighs the benefits of heat transfer improvement against the value of the additional pressure drop encountered by nanofluids. Figure 8 depicts the advantage ratio of applied nanofluids. The AR 1 standard was used to assess the feasibility of hybrid nanofluid adoption. Advantage ratios greater than one should be possible for applications that consider higher heat transfer performance over pressure drop penalty [26, 28]. The greater the advantage ratio, the more practicable the adoption is. The most viable coolant for PEMFC, as indicated in the graph, was the base fluid of Water:BG (60:40), followed by single nanofluids, and finally hybrid nanofluids. In general, advantage ratio is better in serpentine as compared to distributor because the advantage ratio values for serpentine mostly above 1 except for hybrid nanofluids of Al_2O_3 : SiO_2 (10:90). Meanwhile, the advantage ratio for distributor were less than 1 except the base fluid of Water:BG (60:40) at Reynolds number of 300. The advantage ratio was determined to be largest at the lowest Reynolds number because lower Reynolds numbers experience less pressure drop than higher Reynolds numbers [29].



(a) Serpentine (b) Distributo

Fig. 8. Graph of Advantage Ratio for both Cooling Plates

3.7 Coolant Contour Analysis

In this study, the distributor and serpentine cooling plate were investigated. Figure 9 and 10 show the heat distribution on coolant flow for base fluid in comparison to all nanofluids. The lowest plate temperature was spotted on Al_2O_3 : SiO_2 (30:70) hybrid nanofluids for both cooling plates. There was no red spot seen in the contour for Al_2O_3 : SiO_2 (30:70) hybrid nanofluids and Al_2O_3 : SiO_2 (50:50) hybrid nanofluids in both plates. In contrast, the heat distribution for base fluid and single nanofluids displayed some small red spots on Serpentine and Distributor plate. The hottest plate temperature was recorded in base fluid at the same Re 600 for both cooling plates. The cooling plate temperature steadily dropped as the nanofluids were applied.

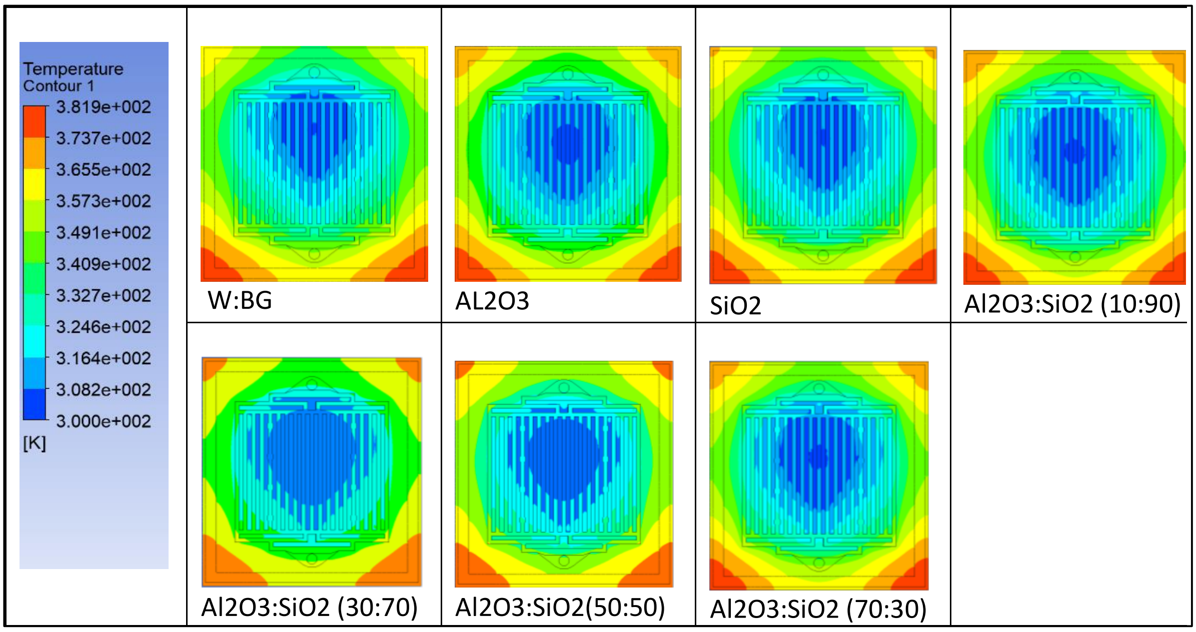


Fig. 9. Temperature Contour at Re 600 for Distributor Cooling Plate

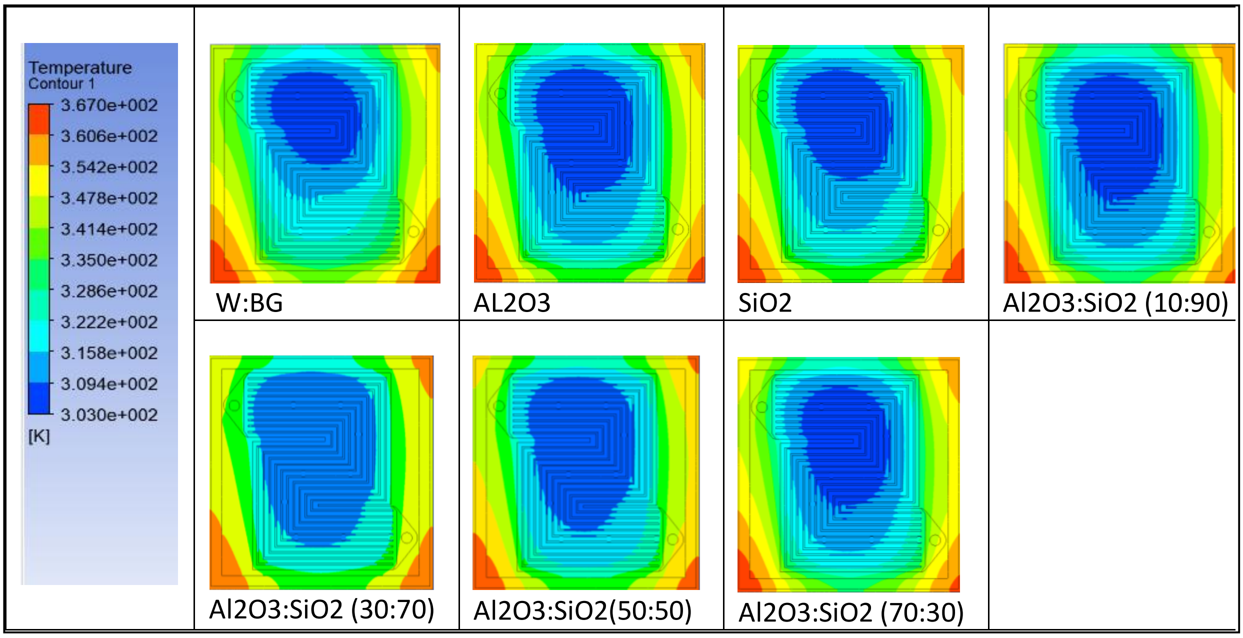


Fig. 10. Temperature Contour at Re 600 for Serpentine Cooling Plate

4. Conclusions

As a conclusion, this numerical simulation has studied the heat transfer enhancement and pressure drop impact on water:BG (60:40) hybrid nanofluids for both the distributor and serpentine cooling plates. Based on the results, it can be concluded that the heat transfer performance of hybrid nanofluids is superior to that of the base fluid. However, there was some penalties on increase of pumping power required with the adoption. The advantage ratio shows that the adoption in serpentine cooling plate is more favourable than distributor type of cooling plate. The adoption of Al_2O_3 : SiO_2 (30:70) hybrid nanofluids and Al_2O_3 : SiO_2 (50:50) hybrid nanofluids were favourable at Re 300 while the single SiO_2 nanofluids adoption in PEMFC can be further increased to Re 1200. However, the findings need to be further validated with actual experiment for confirmation.

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