An experimental investigation of heat transfer enhancement of a minichannel heat sink using Al$_2$O$_3$–H$_2$O nanofluid

M.R. Sohel $^a$, S.S. Khaleduzzaman $^a$, R. Saidur $^{a,*}$, A. Hepbasli $^b$, M.F.M. Sabri $^a$, I.M. Mahbubul $^a$

$^a$ Department of Mechanical Engineering, Faculty of Engineering, University of Malaya, 50603 Kuala Lumpur, Malaysia
$^b$ Department of Energy Systems Engineering, Faculty of Engineering, Yasar University, Bornova, Izmir, Turkey

**Abstract**

The thermal performances of a minichannel heat sink are experimentally investigated for cooling of electronics using nanofluid coolant instead of pure water. The Al$_2$O$_3$–H$_2$O nanofluid including the volume fraction ranging from 0.10 to 0.25 vol.% was used as a coolant. The effects of different flow rates of the coolant on the overall thermal performances are also investigated. The flow rate was ranged from 0.50 to 1.25 L/min as well as the Reynolds number from 395 to 989. The coolant was passed through a custom made copper minichannel heat sink consisting of the channel height of 0.8 mm and the channel width of 0.5 mm. The experimental results showed the higher improvement of the thermal performances using nanofluid instead of pure distilled water. The heat transfer coefficient was found to be enhanced up to 18% successfully. The nanofluid significantly lowered the heat sink base temperature (about 2.7°C) while it also showed 15.72% less thermal resistance at 0.25 vol.% and higher Reynolds number compared to the distilled water.

**1. Introduction**

Due to the higher generation of heat in the electronic chips, there have been widely using liquid cooling systems for their own safety considerations. But the developing technology requires more effective coolant for these systems. The invention of nanofluid [1] has promised to enhance the effectiveness of the new liquid coolant. The suspension of nano-scaled particles (up to 100 nm) to the base fluid is generally defined as a nanofluid. The mixture of solid particles to the liquid generally increases the thermal conductivity of the liquid because of its higher thermal conductivity itself. Miniaturization of the heat sink is another technique to increase the cooling efficiency of the cooling system [2–5]. Since last two decades, there have been committed a lot of numerical, analytical and a few experimental investigation of nanofluids performances applying to the miniaturized heat sink for cooling of electronics [6–11].

Keblinski et al. [12] studied the thermal transportation of nanofluids. They reported that a layer of liquid cluster around the nanoparticle had significant effects to increase the thermal conductivity of the nanofluids. It is also mentioned that the Brownian motion did not play a vital role to improve the thermal transportation of the nanofluids. But Tokit et al. [13] explained that the Brownian motion has an important role to enhance the thermal conductivity by helping the cluster formation of the nanoparticle at a lower motion itself. Putra et al. [14] studied alumina–water and titania–water nanofluids for cooling of electronic chips. The necessity to develop a new cooling technique for microchip cooling instead of conventional technique was discussed in the study. A significant improvement in heat rejection from the electronic chip using nanofluid was obtained. Das et al. [15] inspected the effects of temperature on the heat transfer performances of nanofluids. At 4 vol.% concentration of CuO–H$_2$O, increasing the temperature from 21 to 51°C they observed, the thermal conductivity went up from 14% to 36%, respectively. Nguyen et al. [16] experimentally examined the enhancement of heat transfer coefficient of Al$_2$O$_3$–H$_2$O nanofluid compared to the pure water. About 40% increment in the heat transfer coefficient using 6.8 vol.% of nanoparticle was obtained. The lower particle size of nanoparticle indicated better thermal performances compared to the large particle size. The heat transfer enhancement using Al$_2$O$_3$–H$_2$O nanofluid in a commercial water block was studied by Selvakumar and Suresh [17]. They reported about 20% improvement in the conductance using 20–30 nm alumina nanofluid compared to the deionized water. A little penalty on pumping power with the nanofluids was also mentioned. Whelan et al. [18] designed and investigated a tube array remote heat exchanger for CPU cooling. They successfully obtained a suitable base temperature and a lower thermal resistance.
Recently, the minichannel heat sink has become a very good topic to attract the researcher's attention. The application of nanofluids to minichannel heat sink has offered significant cooling performances for cooling of electronics. Very recently, Ho and Chen [19] performed an experiment on minichannel heat sink using Al2O3–H2O nanofluid. They obtained a very high improvement in heat transfer coefficients using nanofluid compared to the pure water. Tullius and Bayazitoglu [20] also examined the influence of Al2O3–H2O nanofluid on enhancing the heat transfer performance of the circular fin structured minichannel heat sink. The nanofluid showed a great enhancement in thermal transportation. In the same time they also got a little surface imperfection problem because of the nanoparticle sedimentation, which was responsible for reducing the heat transfer performances. The heat transfer characteristics using TiO2–H2O nanofluid in a rectangular minichannel were also experimentally investigated by Naphon and Nakharintr [21]. They perceived a significant enhancement in thermal performance without increasing the pumping power considerably. Keshavarz et al. [22] numerically investigated the nanofluids cooling effects and the pressure drop across the minichannel heat sink. After getting satisfactory results compared to the other available analysis, they suggested a contextual connection for Nusselt number and friction factor.

Although many analytical and numerical analyses have been done on cooling systems for electronics using nanofluids in the open literature, the number of experimental studies conducted is very limited to the best of the author's knowledge. This was the main motivation behind this experimental study, which will be very helpful in recovering some research gap on minichannel heat sink for cooling of electronics using nanofluids.

2. Methodology

2.1. Experimental setup

A simple flow diagram of a close loop for the cooling system is presented in Fig. 1. The close loop comprises of four main components, namely the heat sink, the pump, the radiator and the liquid storage tank. A pump (model: XSPC X20 750) was used to force the coolant to pass through the heat sink where the coolant became hot absorbing heat from the heat source. To recirculate the coolant, it was needed to become cool before entering the heat sink again. This was realized using a radiator type cooler (model: XSPC RS360). So, the hot coolant became cool at the radiator cooler and then went back to the storage tank to recirculate the same process through the system. A rectangular minichannel was used as a heat sink. The copper minichannel with the dimensions of 50 mm × 50 mm × 10 mm was fabricated using a wire electrical discharge machine (WEDM). The magnifying view of the channel is presented in Fig. 2.

Each channel was equally spaced (Wch = Wfin = 0.5 mm) with a height (Hch) of 0.8 mm. Two cartridge heaters (each of 200 W) with the uncertainty range of +5% to –10% in wattage were positioned just below the heat sink separated by a bottom plate to simulate the heat that generates by the electronic equipment. Different temperatures were measured using RTD type thermocouples.
The nanoparticles of different volume fraction were prepared by two step method. First, the appropriate amount of Al₂O₃ nanoparticles was mixed with pure distilled water carefully in a beaker and then ultrasonicated using an ultrasonic homogenizer for 1 h without using any surfactant. The temperature was kept constant using a thermal bath during the homogenization process. In the end of preparation, to make sure the effective size of the particle cluster, the TEM image of nanofluid and the image taken by Zetasizer are shown in Fig. 4(a) and (b), respectively. From figure it is seen that the maximum cluster size is about 142.1 nm.

2.3. Thermophysical properties measurement

Various effective thermophysical properties like the thermal conductivity, the viscosity, the density and the specific heat were experimentally measured using different measurement apparatus. The thermal conductivity was estimated using KD2 Pro Thermal Conductivity meter (Decagon devices, Inc.). Before taking the data, this meter was calibrated and each of the data was repeated in several times. The viscosity was measured using Brookfield DV-III Ultra-Viscosity meter. Before taking the data, the meter was calibrated using glycerin. The density was amounted using density meter (DA-130N, Kyoto electronics). A Digital Scanning Calorimeter (DSC 4000) was used to estimate the specific heat of the coolant. The measuring method and the accuracy of this equipment were followed from the previous study [23]. Also there are different correlations are used to compute the thermophysical properties as well.

The correlations are given bellow:

Pak and Cho [24] density equation is,

$$\rho_{nf} = (1 - \varphi)\rho_f + \varphi\rho_p$$  \hspace{1cm} (1)

Specific heat model proposed by Xuan and Roetzel [25],

$$C_{p,nf} = \frac{(1 - \varphi)C_{p,f} + \varphi C_{p,p}C_{p,f}}{\rho_{nf}}$$  \hspace{1cm} (2)

Thermal conductivity can be calculated by using Hamilton and Crosser model [26] as follows,

$$k_{nf} = k_f + \frac{(SH - 1)k_f - (SH - 1)\varphi(k_f - k_p)}{k_f + (SH - 1)k_f + \varphi(k_f - k_f)}$$  \hspace{1cm} (3)

where, $SH$ is the shape factor. For spherical shape of the nanoparticle, $SH = 3$.

Another thermal conductivity model considering the nano-layer thickness around the nanoparticle is proposed by Yu and Choi [27],

$$K_{nf} = k_f \left( \frac{k_f + 2k_f - 2(k_f - k_f)(1 + \beta)^2 \varphi}{k_f + 2k_f - 2(k_f - k_f)(1 + \beta)^2 \varphi} \right)$$  \hspace{1cm} (4)

where, $\beta = t/r$. $t$ = nano-layer thickness and $r$ = radius of the nanoparticle.

The viscosity model proposed by Nguyen et al. [28],

$$\mu_t = \frac{\mu_{nf}}{\mu_f} = (1 + 0.025\varphi + 0.015\varphi^2)$$  \hspace{1cm} (5)

Recently developed viscosity model by Corcione [29] considering the effective diameter of the nanoparticle,

$$\frac{\mu_{nf}}{\mu_f} = \frac{1}{1 - 34.87(d_p/d_f)^{0.3} \varphi^{0.3}}$$  \hspace{1cm} (6)

where, $d_f$ is the molecular equivalent diameter of water. It can be calculated by,

$$d_f = 0.1 \left( \frac{6M}{\pi\rho_f} \right)^{1/3}$$  \hspace{1cm} (7)
where, $M = 18$ g/mol is the molecular weight of water, the Avogadro number, $N = 6.02214129 \times 10^{23}$ mol$^{-1}$ and $\rho_f$ is the density of the base fluid at room temperature.

### 2.4. Experimental data calculation

The heat received by the coolant is calculated by the following Eq. (8) [19],

$$ q_f = \rho QC_p (T_{out} - T_{in}) $$

The base temperature of the heat sink is evaluated using Eq. (9a) where the base height effect is considered [21].

$$ T_b = T_{av,lc} - \left( \frac{q_f H_b}{k_{hs} A_b} \right) $$

With the heat sink base area,

$$ A_b = L_b N (W_{ch} + W_{fin}) $$

The heat transfer performances are directly related to the inlet–outlet temperature of the coolant and the base temperature of the heat sink. The log mean temperature difference method is very common method used to calculate the thermal performances of the heat exchanger. In this experiment, the log mean temperature difference ($\Delta T_{LMTD}$) is estimated using Eq. (10) [19] based on the base temperature of the heat sink.

$$ \Delta T_{LMTD} = \frac{(T_b - T_{in}) - (T_b - T_{out})}{\ln \left( \frac{T_b - T_{out}}{T_b - T_{in}} \right)} $$

The convective heat transfer coefficient is estimated based on the heat absorbed by the nanofluid and it is calculated by the Eq. (11a) [21],

$$ h = \frac{q_f}{A_{eff} \Delta T_{LMTD}} $$

where, the effective surface area of the minichannel,

$$ A_{eff} = N L_{ch} (W_{ch} + 2H_{ch}) $$

The heat sink is made of high thermal conductivity copper material that is why the fin efficiency is almost 100%. Also, the fin efficiency can be calculated by an iterative method using Eqs. (12) and (13) [30],

$$ \eta_{fin} = \tanh(mH_{ch}) $$

$$ m = \sqrt{2h/(\rho_f C_p W_{fin})} $$

A dimensionless Nusselt number is the proportion of the convective heat transfer to the conductive heat transfer. The Nusselt number can be calculated from,

$$ Nu = \frac{hD_b}{k_{eff}} $$

The dimensionless Reynolds number can be calculated by,

$$ Re = \frac{\rho_{nf} u_m D_h}{\mu_{nf}} $$

With,

$$ u_m = \frac{m}{\rho_{nf} (N A_c)} $$

The convective thermal resistance has substantial effects to resist the heat removal performance of the coolant. The convective thermal resistance for the coolant flow can be calculated using Eq. (16) [21],

$$ R_{th} = \frac{1}{h A_{eff}} = \frac{\Delta T_{LMTD}}{q_f} $$

The thermal effectiveness of the heat sink, based on the maximum wall temperature can be calculated using following Eq. (17) [31],

$$ \varepsilon_{th} = 1 - \frac{(T_{w,\text{max}} - T_{in})_{\text{eff}}}{(T_{w,\text{max}} - T_{in})_{\text{full}}} $$

The pumping power is estimated using the Eq. (18) [17],

$$ P_P = Q \times \Delta P $$

### 2.5. Uncertainty analysis

The uncertainties associated with the experimentally measured properties are demonstrated in Table 1. The uncertainties of the calculated results are determined using propagation analysis [32].
Table 1
Uncertainty analysis of the experimentally measured properties.

<table>
<thead>
<tr>
<th>Parameters (unit)</th>
<th>Uncertainty (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density, ρ (kg/m³)</td>
<td>±1.0</td>
</tr>
<tr>
<td>Thermal conductivity, k (W/m K)</td>
<td>±1.0</td>
</tr>
<tr>
<td>Viscosity, μ (Ns/m²)</td>
<td>±1.0</td>
</tr>
<tr>
<td>Specific heat, Cₚ (J/kg K)</td>
<td>±1.0</td>
</tr>
<tr>
<td>Effective heat transfer area, Aₑ (cm²)</td>
<td>±0.2</td>
</tr>
<tr>
<td>Hydraulic diameter, Dₚ (mm)</td>
<td>±0.2</td>
</tr>
<tr>
<td>Flow rate, Q (L/min)</td>
<td>±0.3 to ±2.2</td>
</tr>
<tr>
<td>Temperature difference, (T₃ − T₄) (°C)</td>
<td>±1.5 to ±5.0</td>
</tr>
<tr>
<td>Temperature difference, (T₃ − T₄) (°C)</td>
<td>±1.5 to ±5.0</td>
</tr>
<tr>
<td>Temperature difference, (T₅ − T₆) (°C)</td>
<td>±0.4 to ±2.2</td>
</tr>
</tbody>
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\[
\delta q_f = \left[ \left( \frac{\delta q_f}{q_f} \right)^2 + \left( \frac{\delta \rho}{\rho} \right)^2 + \left( \frac{\delta C_p}{C_p} \right)^2 + \left( \frac{\delta Q}{Q} \right)^2 + \left( \frac{\delta (T_{out} - T_{in})}{(T_{out} - T_{in})} \right)^2 \right]^{1/2} \tag{19}
\]

\[
\delta h = \left[ \left( \frac{\delta h}{h} \right)^2 + \left( \frac{\delta \rho}{\rho} \right)^2 + \left( \frac{\delta C_p}{C_p} \right)^2 + \left( \frac{\delta Q}{Q} \right)^2 + \left( \frac{\delta (T_{out} - T_{in})}{(T_{out} - T_{in})} \right)^2 \right]^{1/2} \tag{20}
\]

\[
\delta Nₚ = \left[ \left( \frac{\delta Nₚ}{Nₚ} \right)^2 + \left( \frac{\delta D_{n₁}}{D_{n₁}} \right)^2 + \left( \frac{\delta h}{h} \right)^2 \right]^{1/2} \tag{21}
\]

\[
\delta Rₚ = \left[ \left( \frac{\delta Rₚ}{Rₚ} \right)^2 + \left( \frac{\delta (T_{out} - T_{in})}{(T_{out} - T_{in})} \right)^2 + \left( \frac{\delta (T_{out} - T_{in})}{(T_{out} - T_{in})} \right)^2 + \left( \frac{\delta (T_{out} - T_{in})}{(T_{out} - T_{in})} \right)^2 \right]^{1/2} \tag{22}
\]

Eqs. (19)–(22) are used to analyze the uncertainty of the heat received by the coolant, the heat transfer coefficient the Nusselt number and the thermal resistance, respectively. The maximum uncertainties are calculated, \( q_f = ±4.42\% \), \( h = ±4.01\% \), \( Nₚ = ±1.58\% \), and \( Rₚ = ±0.78\% \). From the calculation, it is seen that the variation in volume fraction has no substantial effects on the variation of the uncertainties, but changing the flow rates has a great effect on changing the uncertainties. Normally at lower volume flow rates, the uncertainties are high.

3. Results and discussion

3.1. Thermophysical properties

Fig. 5 represents the changing in different thermophysical properties with the variation of the volume fraction of the nanoparticle. Also the experimentally measured data is compared with the analytical data calculated using different correlations. Fig. 5(a) shows that the relative density gradually rises with the enhancement of volume portion. The analytical data also shows the same inclination. So, from figure it can be said that there is a good covenant (maximum deviation is about 0.15%) with the experimental and analytical data.

Relative specific heat is schemed against the volume fraction in Fig. 5(b). The figure represents a covenant between experimental and analytical measurement. The specific heat does not increase following the other thermophysical properties. Lower specific heat of the nanoparticle likened to the base fluid causes to reduce the specific heat of the nanofluid. The maximum deviation of the experimentally measured data with the analytically calculated data was found about 3%.

The thermal conductivity is the most dominating properties to improve the heat transfer performance of the nanofluid. The relative thermal conductivity is presented in the Fig. 5(c). From figure it can be explained that the thermal conductivity of the nanofluid increases significantly with the increasing of the volume fraction of the nanoparticle. Besides the experimental results there are also calculated using two different correlations. The experimental result shows a little deviacyon with the Hamilton and Crosser model whereas a good covenant is seen with the recently developed Yu and Choi model. A nano-layer is not considered in the Hamilton and Crosser model but is considered in the Yu and Choi model that is also responsible to maximize the thermal conductivity of the nanofluid significantly. The experimentally measured data shows maximum 0.27% higher value compared to the Hamilton crosser model and maximum 0.0057% higher value compared to the Yu and Choi model.

The viscosity is another important thermophysical properties also measured experimentally and compared with different correlations. The relative viscosity is plotted in Fig. 5(d). From figure it is observed that the viscosity rises with the volume loading of the nanoparticle. The maximum abnormality of experimental result is seen with the Nguyen et al. model whereas a good agreement is perceived with the Corcione model. Maximum deviation of the experimentally measured data was found about 0.63% with the Nguyen et al. model and about 0.21% with the Corcione model. The viscosity does not depend only on the volume percentage of the nanoparticle anymore but there are some more properties also need to be considered. The surface adsorption and the nano-layer cluster around the nanoparticles are responsible to increase the effective diameter of the nanoparticle that is also responsible to enhance the viscosity of the nanofluid. In the Corcione model, the effective diameter of the nanoparticle is considered and as a consequence the viscosity is little higher compared to the Nguyen et al. model.

3.2. Heat sink base temperature

The main objective of the cooling system is to minimize the base temperature as well as the heat source temperature and to hold at a constant level. The effect of nanofluid with different volume fraction is shown in Fig. 6 at a range of Reynolds number from 395 to 989.

From the figure, it is observed that the Al₂O₃–H₂O nanofluid with enhancing the volume percentage has a significant effect on the reduction of the heat sink base temperature. In the same time, with increasing the Reynolds number also minimizes the base temperature fruitfully. Selvakumar and Suresh [17] got a maximum 1.5 °C reduction of the interface temperature at 2.45 L/min and 0.2 vol.% using CuO–H₂O nanofluid with the comparison of the pure water. Whereas in this experiment, the maximum base temperature reduction at the lowest Reynolds number (395) for 0.25 vol.% of Al₂O₃–H₂O nanofluid attained about 1.4 °C and at the highest Reynolds number \( (Re = 989) \) that was 2.7 °C for the same volume fraction compared to the pure distilled water. From the figure, it can be explained that the nanofluid absorbs more heat compared to the pure distilled water. Also, the addition of volume percentage of the nanoparticle develops the heat absorptivity dramatically. Because of all this effectiveness, nanofluid shows lower base temperature of the heat sink compared to the water.

3.3. Log mean temperature difference

Eq. (11a) denotes the heat transfer coefficient is inversely related to the log mean temperature difference and that is also related to the other thermal properties significantly. Changing the log mean temperature difference with the changing in Reynolds
Increasing the volume fraction shows the decreasing drift of the log mean temperature difference as well as the lowering of the base temperature of the heat sink. Also the developing of the flow rate causes to raise the Reynolds number has the same trend to decrease the log mean temperature difference. A maximum decreasing rate of 12.50% was obtained for the Reynolds number of 989 and 0.25 vol.% of the Al₂O₃–H₂O nanofluid related to the pure distilled water.

3.4. Heat transfer coefficient

The thermal performances of the nanofluids depend on the convective heat transfer coefficient. The effects of the Reynolds number and the volume percentage of the Al₂O₃ nanoparticle on heat transfer coefficient are shown in Fig. 8. The heat transfer coefficient is calculated based on the average base temperature and the inlet and outlet temperature difference. Fig. 8 emphasizes that there is an effective enhancement of the heat transfer coefficient in the range of Reynolds number from 395 to 989 and also with increasing of the volume fraction from 0.10 to 0.25 vol.%. At the highest volume fraction, Al₂O₃–H₂O nanofluid gave around 18% higher value of convective heat transfer coefficient with the comparison of the distilled water. The higher thermophysical properties of the nanofluid over the distilled water are the main reason for enhancing the convective performance of the nanofluid. The addition of more volume concentration of nanoparticle to the base fluid contributes to improve the thermal conductivity and the Brownian motion, which are directly responsible for enhancing the thermal performances of the nanofluid over the pure distilled water. Besides, the increasing Reynolds number also represents the improvement of the convective heat transportation capacity of the coolant and that maximize the reduction of the base temperature of the heat sink.
3.5. Nusselt number and Reynolds number

For a particular heat sink, the conductive heat transfer is constant because of the particular heat sink material, physical dimensions and also for a heat source. But the convective heat transfer changes with the fluid and flow properties of the cooling system. The Nusselt number represents the convective performance of the coolant. Besides the Reynolds number also represents the convective performance and the flow characteristics (laminar flow or turbulent flow). Fig. 9 illustrates the variation of Nusselt number and Reynolds number at a different volume fraction of Al\textsubscript{2}O\textsubscript{3}-H\textsubscript{2}O nanofluid. From the figure, the nanofluid containing the higher volume fraction has the higher Nusselt number as well as the higher Reynolds number. At 0.25 vol.\%, Al\textsubscript{2}O\textsubscript{3}-H\textsubscript{2}O nanofluid shows 17.80% higher Nusselt number over the pure distilled water. The Reynolds number characterizes the laminar flow of the coolant in this experiment. On the other hand, the Reynolds number also proportionally related to the inlet velocity of the coolant. So, the higher Reynolds number means the higher inlet velocity. With the increasing nanofluid velocity increases the nanoparticles movement or inter-collision that effectively enhances the heat transfer rate. So, the significant increase in Nusselt number and Reynolds numbers motivates to use Al\textsubscript{2}O\textsubscript{3}-H\textsubscript{2}O nanofluid as a better coolant instead of a conventional coolant.

3.6. Thermal resistance

Fig. 10 represents the diversity of the convective thermal resistance with the disparity of the volume concentration of the Al\textsubscript{2}O\textsubscript{3}-H\textsubscript{2}O nanofluid in the range of 395 ≤ Re ≥ 989. From the figure, it is seen that the higher Reynolds number represents the lower thermal resistance. Also increasing the volume concentration of the nanoparticle shows a declining trend to diminish the convective thermal resistance. Developing the volume concentration of the nanoparticle raises the convective heat transfer coefficient and thermal dispersion, which are the main causes for reducing the convective thermal resistance compared to the base fluid. In this experiment, at the highest volume concentration (0.25 vol.\%) and the Reynolds number, the maximum reduction about 15.72% in the convective thermal resistance was obtained compared to the pure distilled water. At a higher Reynolds number, the particle Brownian motion increases with the increasing of nanofluid’s velocity and in a consequence the thermal transportation of the nanofluid also increases. Improvement in thermal transportation increases the convective heat transfer coefficient that is inversely related to the convective thermal resistance. So, the enhancing in convective heat transfer coefficient ultimately decreases the thermal resistance of the nanofluid.

3.7. Thermal effectiveness

The thermal effectiveness of a cooling system is important to understand the overall cooling performance of the system. It helps to select the most effective operating condition of a system. Fig. 11 shows the change of the thermal effectiveness of the cooling system with the variation of flow rate as well as the volume fraction of the nanoparticle. From figure it can be interpreted that the higher volume fraction of the nanoparticle shows higher thermal effectiveness. But, the increasing flow rate of the coolant has different effects on the cooling performance. Initially, the thermal effectiveness increases with the increasing of the flow rate but after certain value it goes down even the flow rate is still increasing. Normally, the increasing flow rate also raises the velocity of the coolant passing through the minichannel. In practical case, the high velocity coolant gets less time to absorb heat; as a consequence the heat sink base temperature reduction rate also decreases after certain flow rate, which is already investigated. As a result, the effectiveness decreases after increasing the flow rate at a definite level.

3.8. Pumping power

When the coolant passes through the narrow channel of the heat sink, a pressure drop occurs. To overcome this pressure drop, the system needs some extra pumping power. Fig. 12 demonstrates the increasing in pumping power with the increasing in volume flow rate as well as the volume percentage of the nanofluid. Comparatively higher density of the nanofluid is the main region to increase a little amount of the pumping power compared to the pure water. Also the increasing in viscosity is another cause to raise the pumping power of the nanofluid. But this extra pumping power can be compromised considering the performance improvement of the nanofluid compared to the pure water.
1. A great performance was obtained using Al₂O₃–H₂O nanofluid as a coolant compared to the pure distilled water. It was successfully attained to minimize and hold the base temperature of the heat sink at a constant level with the Reynolds number.

2. A significant improvement in the heat transfer coefficient (18%) was achieved using the nanofluid compared to the distilled water. The heat transfer coefficient was evaluated based on the log mean temperature difference. The Nusselt number also increased considerably with the enhancement of the volume percentage as well as the Reynolds number of the nanofluid.

3. The convective thermal resistance was minimized in a significant manner. The experiment showed the minimum convective thermal resistance at a higher volume fraction and Reynolds number of the nanofluid about 15.72% lower compared to the distilled water.

Now it can be concluded that the Al₂O₃–H₂O nanofluid applied to the minichannel heat sink has a significant enhancement in heat transfer performances for electronics cooling compared to the pure distilled water.

### Conflict of interest

I, as the third and corresponding author hereby on behalf of all the authors, announce that, we do not have any actual or potential conflict of interest including any financial, personal or other relationships that is: employment, consultancies, stock ownership, honoraria, paid expert testimony, patent applications/registrations, and grants or other funding with other people or organizations within thirteen (13) years of beginning the work submitted that could inappropriately influence (bias) our work.

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