HEAT TRANSFER AND HYDRODYNAMIC PERFORMANCE ANALYSIS OF A MINIATURE TANGENTIAL HEAT SINK USING AL₂O₃–H₂O AND TiO₂–H₂O NANOFLUIDS

Seyed Ziaeddin Miry, Majid Roshani, Pedram Hanafizadeh, Mehdi Ashjaee, and Faezeh Amini

In this article, thermal and hydrodynamic performances of a miniature tangential heat sink are investigated experimentally by using Al₂O₃–H₂O and TiO₂–H₂O nanofluids. The effects of flow rate and volume concentration on the thermal performance have been investigated for the Reynolds number range of 210 to 1,100. Experimental results show that the average convective heat transfer coefficient increases 14 and 11% and the bottom temperature of the heat sink decreases 2.2°C and 1.6°C by using Al₂O₃–H₂O and TiO₂–H₂O nanofluid instead of pure distilled water, respectively.

Keywords miniature tangential heat sink, nanofluid, heat transfer, pumping power

INTRODUCTION

Rapid developments in the electronic industry have led to the advent of high-capacity electronic chips, which generate a large amount of heat flux, and damping this heat flux has been presented as a new challenge for researchers during the last decade. Considering the low efficiency of the usual compact heat dampers, researchers turned to methods through which they could enlarge the convective heat transfer capacity by disturbing thermal and hydrodynamic patterns of fluid flow, since the boundary layer flow does not have high heat transfer coefficients.

Miniaturization of the heat sink has been presented as a technique to increase cooling system efficiency, because miniature heat sinks can damp a large amount of heat flux from electronic chips due to their high surface area-to-volume ratio [1, 2]. To achieve damping a large amount of heat generated in the electronic chips, using different geometries, such as multilayer micro channel heat sinks [3], pin fin geometries [4], and stripe fins [5], has been...
suggested. Among the suggested plans, the miniature tangential heat sink is an innovative plan to remove heat from electronic components.

Using an impinging liquid jet in the heat sink is the method to achieve an efficient cooling system because of its very high flux removal. The impinging liquid jet method provides a high local heat transfer coefficient between the impinging liquid and the surface of the heat sink. The advantages of using jet impingement in a heat sink are higher cooling effectiveness, greater temperature uniformity, and more controllability. Using the impinging jet method in the heat sink has been investigated by many researchers.

Naphon and Wongwises [6] experimentally investigated the jet liquid impingement heat transfer characteristic in a mini-rectangular fin heat sink of a CPU based on the real operating condition of a PC. The results of the jet liquid impingement cooling system were compared with a conventional liquid cooling system. The CPU temperature of the jet liquid impingement cooling system was lower than conventional liquid system.

Sung and Mudawar [7, 8] studied a hybrid cooling scheme of multiple impinging jets in micro-channels. Their plan consisted of the injection of HFE 7100 fluid from a series of circular micro-jets to each micro-channel in a modified micro-channel heat sink. They applied numerical analysis for determination of the main parameters in this geometry. The proposed geometry was experimentally investigated, and finally, the superposition technique was used to expand a correlation for determining the characteristic parameters of this type of cooling method. They obtained a cooling capacity of 304.9 W/cm² in their experiments with no phase change. The results of [7] were used in [8] to obtain a high jet Reynolds number and two-phase cooling.
Lelea et al. [9] numerically considered a micro heat sink with an impinging jet positioned tangentially to the tube in the middle of the micro-tube. By using water as the basic fluid in the laminar flow regime (Reynolds number of less than 1,000), they concluded that the temperature of the bottom of the sink remains uniform at 304 K.

In [10], the micro heat sink was optimized to achieve efficient thermal performance. In this research, four different inlet rectangular cross-sections were analyzed. In [11], the number and position of the inlet jet were analyzed and the results of manifold jets and a single jet at the same pumping power compared.

The jet impingement method needs enough jets to ensure temperature uniformity of the heat sink bottom surface. Temperature non-uniformity affects the performance of electronic systems and creates mechanical stresses due to temperature distribution differences from one side of the heat sink to the other. The heat sink used in the present study has five channels in which fluid flows into the channel as an impinging jet through the three inlets, which are tangent to each channel surface to achieve temperature uniformity of the bottom surface of the heat sink.

The idea of using fluids with higher thermal performance as a cooling fluid in the damping of high heat flux was created by Choi et al. [12] by adding metallic and nonmetallic particles to the water-based fluid for the first time.

Investigating the properties of the suggested nanofluids reveals that adding nanoparticles to the basic fluid not only increases the thermal conductivity of the basic fluid but also causes the direct dependence of thermal conductivity with the fluid temperature. This fact enhances convective heat transfer and temperature uniformity at the bottom of the heat sink. Therefore, using such nanofluids as the cooling fluid has opened new horizons to researchers.

In recent decades, many researchers have worked on properties of nanofluids and mentioned different methods to increase heat transfer in nanofluids, the most important of which is conductive heat transfer enhancement.

Yiamsawasd et al. [13] performed research in which conductivity of two nanofluids, TiO₂ and Al₂O₃, were experimentally investigated. They used water and water–ethylene–glycol as the basic fluid. The experimental results showed that nanofluid conductive heat transfer increases with increase in temperature and volume concentration. They suggested a model for predicting nanofluid conductive heat transfer. The obtained results of the presented model were compared with other researchers’ experimental results and showed agreement.

Masuda et al. [14] investigated conductivity of three nanofluids, SiO₂–H₂O, Al₂O₃–H₂O, and TiO₂–H₂O, using the hot-wire method. They found out that conductivity of nanofluids is considerably higher than conductivity of basic fluid, so conductivity of the Al₂O₃–H₂O nanofluid was 32% higher than water in a volume concentration of 4.3%. They concluded that temperature has no effect on the conductivity.

Lee et al. [15] also applied the hot-wire method for measuring conductivity of nanofluids Al₂O₃–H₂O and CuO–H₂O. In their experiments, the scale of Al₂O₃ and CuO particles were 23.6 and 38.4 nm, respectively, and the volume concentration varied from 1 to 4%. Results showed that conductivity increases with increase in nanoparticles. Comparing the achieved results with the Hamilton–Crosser prediction [16], they inferred that this model is capable of predicting conductivity of Al₂O₃ at an acceptable level but cannot accurately predict conductivity of CuO.
Das et al. [17] considered the effect of temperature on conductivity of CuO–H₂O and Al₂O₃–H₂O nanofluids in a volume concentration range of 1–4% via a temperature oscillation technique. Their results showed that nanofluid conductivity increases with increase in temperature; their experimental results also showed higher values than a classical model.

Also, some other mechanisms for enhancing heat transfer of nanofluids are mentioned in the references, such as Brownian motion [18], thermophorsis [19], and so on. Thus, combining the liquid jet impingement and nanofluid technologies is thought to achieve the advantages of both, consequently enhancing the transfer coefficient significantly.

Nguyen et al. [20] carried out experimental investigations to study the heat transfer performance of a nanofluid (Al₂O₃) for confined and submerged impinging jets. From the experimental data of a particular nanofluid with 6.8% particle volume concentration, the heat transfer coefficient was found to increase as much as 40% compared to that of the base fluid.

Tabrizi and Seyf [21] numerically investigated effects of using Al₂O₃–H₂O nanofluid on thermal performance and hydrodynamic performance of a tangential micro heat sink. They optimized the heat sink using the EGM method, which is a novel method in optimizing heat transfer systems. Their results showed that Al₂O₃–H₂O nanofluid considerably increases overall thermal performance of tangential micro heat sink with little increase in pumping power. Creating uniform temperature in the bottom of a heat sink is another benefit of using nanofluid. Moreover, they investigated effects of Reynolds number, volume concentration, and size of nanoparticles on heat transfer and entropy generation.

Kalteh et al. [22] investigated convective heat transfer of nanofluid flows in a rectangular micro-channel heat sink, experimentally and numerically. A Eulerian–Eulerian two-phase method was used in their numerical simulation. The heat sink had been built from silicon, and both experimental and numerical results were analyzed. Finally, the effects of different parameters, such as nanoparticles diameter, volume concentration, and Reynolds number, on heat transfer enhancement were analyzed.

Ho et al. [23] experimentally analyzed convective heat transfer of Al₂O₃–H₂O nanofluid rather than pure distilled water in a micro-channel heat sink. They reported 70% enhancement in average heat transfer coefficient in a nanofluid volume concentration of 1% in comparison with pure distilled water. They also pointed to a 0.029-K/W decrease in heat sink thermal resistance and 25% decrease in the maximum wall temperature of the heat sink.

Fazeli et al. [24] studied heat transfer characteristics of a miniature heat sink cooled by an SiO₂–water nanofluid both experimentally and numerically. Their results showed that dispersing SiO₂ nanoparticles in water significantly increases the overall heat transfer coefficient while the thermal resistance of the heat sink decreases. Moreover, they concluded that channel diameter, heat sink height, and number of channels in the heat sink have significant effects on the maximum temperature of the heat sink.

Zirakzadeh et al. [25] experimentally investigated heat transfer of a miniature heat sink cooled by Al₂O₃–H₂O nanofluids. They developed a new plate pin finned heat sink and observed that the new heat sink increased the heat transfer coefficient up to 20% in comparison with the conventional plate fin heat sink.
In Ho and Chen [26], efficiency of using a nanofluid as the cooling fluid in a mini-channel heat sink was investigated. They estimated enhancement of the average heat transfer coefficients based on the bulk and inlet temperature difference by more than 72 and 35% for Al₂O₃–H₂O nanofluid with volume concentration of 10% compared with pure distilled water. In addition, they mentioned that Al₂O₃–H₂O can serve as an effective coolant for suppressing the hot spot in the mini-channel heat sink.

Nitiapiruk et al. [27] experimentally investigated heat transfer and pressure drop of a TiO₂–H₂O nanofluid flowed in a micro-channel heat sink. Furthermore, the effects of uncertainty of thermophysical properties, such as the conductive heat transfer coefficient and viscosity, in the prediction of Nusselt number and friction factor in a micro-channel were investigated. They used three kinds of models for prediction of viscosity and conductive heat transfer. The size of the applied micro-channel and amount of applied heat flux were completely in accordance with a real situation observed in the personal computers.

Sohel et al. [28] experimentally considered a mini-channel heat sink using A₂O₃–H₂O nanofluid. They reported that the heat transfer coefficient increases 18% and temperature decreases 2.7°C at the bottom of the heat sink by using the Al₂O₃–H₂O nanofluid instead of pure distilled water. A 17.8% increase in Nusselt number and 15.7% decrease in convective thermal resistance using 2.5% Al₂O₃ nanofluid compared with pure distilled water were also results of their investigation.

Escher et al. [29] extended the study of convective heat transfer characteristics of SiO₂–H₂O nanofluids to highly concentrated fluids and presented an experimental and theoretical evaluation of their potential for electronics cooling. They experimentally measured thermophysical properties of the fluid, such as density, specific heat, thermal conductivity, and dynamic viscosity, then studied the heat transfer characteristic of the nanofluids and their potential to enhance the performance of a micro-channel heat sink.

An investigation of the literature shows that there are a number of numerical studies on heat sinks using nanofluids, but experimental studies in this field for cooling of electronic chips are very limited. The lack of research in this field was thus the main purpose behind the investigation of the overall thermal performance of a miniature tangential heat sink using nanofluids. Therefore, in this research, a miniature tangential heat sink is designed and investigated experimentally by using two nanofluids—Al₂O₃–H₂O and TiO₂–H₂O—as the cooling fluids. Likewise, the effects of volume concentration and Reynolds number on thermal and hydrodynamic performance can be considered as other achievements of this research.

**METHODOLOGY**

**Experimental Set-up**

The experiments of this study were conducted in a test facility that consists of three main elements, namely a closed loop for fluid circulation, heat sink test section data acquisition, and pressure drop measuring system. The experimental set-up with a corresponding apparatus is shown schematically in Figure 1.

**Closed loop.** The closed loop of the set-up includes a storage tank, pump, ball valve, and calibrated flow meter for measuring mass flow rate. In the upstream of the test
section, an isothermal bath (F30-Julabo) has been used to maintain the temperature of the inlet fluid of the heat sink constant. The flow rates were regulated by a ball and were measured by the calibrated flow meter with an accuracy of \(\pm 50\) mL. As shown in Figure 1, nanofluid is pumped from the storage tank and, after passing through the flow meter, enters into the Plexiglas tank and passes through the heat sink. Thereafter, nanofluid flows through the copper coil, which is in an isothermal bath, reaches the desired temperature, and returns into the storage tank; this cycle will be repeated.

**Test section.** The test section of the experimental set-up consists of a miniature tangential heat sink, insulated Plexiglas storage, and a heater block that has been placed in an insulated box made of bone fiber; its surroundings have been filled with slag wool (more details will be given later).

**Heat sink.** As shown in Figure 2, the miniature tangential heat sink consists of a block with length, width, and height of 40, 40, and 8.5 mm, respectively, which is made of aluminum 7000 with a CNC system. Five circular channels with diameters of 4 mm and an equal distance (8 mm) from each other have been created in this heat sink.

The distance of the first channel from the side and bottom surface of the heat sink is 4 and 4.5 mm, respectively. Each channel has three tangential inlets with diameters of 2 mm, which provide the impingement liquid jet. The distance between the tangential inlets in the direction of channel axes is 10.5 mm, and their distances with adjacent channels are 10 and 6 mm, respectively. To create minimal thermal resistance between the bottom surface of the heat sink and the heater block, high conductivity silicon pulp was used between the two surfaces.
The inlet and outlet chambers. The heat sink was placed in a Plexiglas case and insulated from the surrounding environment. The surfaces of the Plexiglas case have been covered with Styrofoam, so heat transfer from the Plexiglas surface has been minimized and minimum error achieved. The Plexiglas case has one inlet flow and two outlet flows. Two holes are placed in the Plexiglas in inlet and outlet flows of the heat sink to measure differential pressure of nanofluid flow (see Figure 3). The range of inlet and outlet temperatures, applied flow rates, heat fluxes to the base, and air temperature in the laboratory during the test procedures are summarized in Table 1.

Heater block. The heater block was used for simulation of producing heat flux from an electronic chip. The heater block is made of aluminum, and its geometric dimensions are shown in Figure 4. A heat cartridge was placed in the heater block, and the produced heat flux will be measured using six K-type thermocouples.

Data acquisition and pressure measuring unit. Two K-type thermocouples with an accuracy of ± 0.1°C have been placed at the inlet and outlet of the manifold to calculate the bulk temperature of fluid. The bottom surface temperature of the heat sink will be measured using four K-type thermocouples with an accuracy of ± 0.1°C. These thermocouples were placed in the holes at a distance of 1 mm with the bottom of the heat sink and with diameter and depth of 1 and 20 mm, respectively. The produced heat flux from the heater block will be measured using six K-type thermocouples placed in the heater block. Connecting the thermocouples to the data logger (prova 800 and testo 777 data logger), temperatures will be measured continuously during the experiments.
Considering the unchanged temperature in the steady state, a constant temperature condition in the heat sink can be concluded. The data logger has been connected to a computer using a USB cable. Thermocouples have been calibrated in the range of 0°C–100°C. The pressure drop of fluid flowing through the heat sink has been measured using a high-precision differential pressure transmitter with an accuracy up to ±1.0 Pa.

**Nanofluid Preparation**

In this research, a technique with two stages is used for nanofluid preparation. In this technique, the nanoparticle is first prepared and subsequently dispersed into the basic fluid. Considering previous research experienced in this technique for oxide nanoparticles shows that a uniform and stable nanofluid will be achieved.

Two kinds of particles namely, Al₂O₃ and TiO₂, with diameters of 40 and 20 nm have been utilized in the experiments. Pure distilled water is also used as the basic fluid. The thermophysical properties of water and nanoparticles are summarized in Table 2.

![Image of Plexiglas case.](image)

**Table 1.** Range of inlet and outlet temperatures, applied flow rates, heat fluxes to the base, and air temperature in the laboratory during tests

<table>
<thead>
<tr>
<th>Nanofluid</th>
<th>Inlet temperature (°C)</th>
<th>Outlet temperature (°C)</th>
<th>Volume flow rate (m³/s) × 10⁻⁶</th>
<th>Heat flux to the base (W/m²) × 10¹⁶</th>
<th>Air Temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Al₂O₃–water</td>
<td>40.0</td>
<td>41.4–47.6</td>
<td>6.33–34.22</td>
<td>0.125</td>
<td>20.0</td>
</tr>
<tr>
<td>TiO₂–water</td>
<td>40.0</td>
<td>41.4–47.7</td>
<td>6.22–34.06</td>
<td>0.125</td>
<td>20.0</td>
</tr>
</tbody>
</table>
Applied volume concentrations of nanoparticles are 0.5, 1, 1.5, and 2%. To produce volume concentration, nanoparticles have been weighed and added to the basic fluid of water in three stages and sonicated for 30 min in each stage. For this purpose, an ultrasonic homogenizer (UP 400S) with a power of 400 W and frequency of 24 kHz is used. By this

Figure 4. Geometric configuration of heater block.
action, an agglomeration of nanoparticles in the base fluid decreases, and a homogenous mixture is produced. Finally, two nanofluids of Al$_2$O$_3$ –H$_2$O and TiO$_2$ –H$_2$O are produced with four different volume concentrations. Al$_2$O$_3$ and TiO$_2$ are the most common nanoparticles used by researchers in their experimental studies. Investigation into the thermophysical properties of Al$_2$O$_3$ –H$_2$O and TiO$_2$ –H$_2$O nanofluids, such as heat transfer characteristic [20–29], thermal conductivity, and viscosity [30, 31], has been conducted by many researchers. Al$_2$O$_3$ –H$_2$O and TiO$_2$ –H$_2$O nanofluids are important because they can be used in numerous applications, such as cooling automobile engines and welding equipment, and to cool high heat flux devices, such as high-power microwave tubes and high-power laser diode arrays.

**Thermophysical properties measurement.** Since thermophysical properties of nanofluids are used to calculate results of this research, the applied model for determining thermophysical properties is explained in this section. The nanofluids used in this research are Al$_2$O$_3$ –H$_2$O and TiO$_2$ –H$_2$O, which are produced at volume concentrations of 0.5, 1, 1.5, and 2%. A classical model was used for calculation of density [32], special heat capacity, and viscosity as below:

\[
\rho_{nf} = (1 - \varphi)\rho_{bf} + \varphi\rho_{p},
\]

where subscripts $nf$, $bf$, and $p$ denote nanofluid, basic fluid, and particle properties. $\varphi$ is the volume fraction of the particle in the base fluid. Density, viscosity, and special heat capacity of the nanofluid depend on temperature, because temperature-based models are used for the calculation of density and effective viscosity of the basic fluid. The variations of density and viscosity of nanofluid with temperature are given by the follow formulas:

\[
\rho_{nf} = -13.89\sqrt{T} + 1,237,
\]

\[
\mu_{bf}(T) = \mu_B \exp \left( \frac{T_\mu}{T} \right),
\]

where $T_\mu = 1,713$ K and $\mu_B = 2.761 \times 10^{-6}$ Ns/m$^2$. It is assumed that basic fluid and nanoparticles are in thermal equilibrium, and therefore, nanofluid special heat capacity is calculated by [33]

\[
(\rho C_p)_{nf} = (1 - \varphi)(\rho C_p)_{bf} + \varphi(\rho C_p)_p.
\]

The Brinkman model [34] is used for calculation of viscosity of nanofluid in which the mixture has been considered homogeneous; this model can estimate viscosity of a
nanofluid correctly for volume concentrations of less than 4%, defined as

$$\mu_{nf} = \frac{\mu_{bf}}{(1 - \varphi)^{2.5}},$$ \hspace{1cm} (5)

where $\mu_{bf}$ and $\mu_{nf}$ are viscosity of the basic fluid and nanofluid, respectively, and $\varphi$ shows the volume concentration.

To calculate effective conductivity of Al$_2$O$_3$–H$_2$O, the model suggested by Koo and Kleinstreuer [35] was used. This model consists of two static and dynamic parts. The static part of this model is the famous model of Hamilton–Crosser [16], which is merely a function of conductivity of the basic fluid and volume concentration. Its dynamic part is used for modeling the Brownian motion of nanoparticles, which is influenced by volume concentration, size of nanoparticles, temperature, type of nanoparticles (density), and basic fluid combination. Therefore, the conductivity of nanofluid can be calculated by the following correlation:

$$k_{nf} = k_{static} + k_{Brownian},$$ \hspace{1cm} (6)

$$k_{static} = \left[\frac{k_p + (n-1)k_{bf} - (n-1)(k_{bf} - k_p) \times \varphi}{k_p + 2k_{bf} + (k_{bf} - k_p) \times \varphi}\right]k_{bf},$$ \hspace{1cm} (7)

where $n$ is considered to be equal to 3 for spherical particles in the Hamilton–Crosser model; hence, by substitution, this equation becomes

$$k_{static} = \left[\frac{k_p + 2k_{bf} - 2(k_{bf} - k_p) \times \varphi}{k_p + 2k_{bf} + (k_{bf} - k_p) \times \varphi}\right]k_{bf},$$ \hspace{1cm} (8)

where $k_{bf}$ will be calculated as follows:

$$k_{bf}(T) = k_B(1 + \alpha_k T),$$ \hspace{1cm} (9)

where $k_B = 0.6$ W/mK and $\alpha_k = 4.167 \times 10^{-5}$/K. The dynamic part of the conductivity is defined as

$$k_{Brownian} = 5 \times 10^4 \varphi \rho_{bf}(C_p)_{bf} \beta \sqrt[3]{\frac{\kappa T}{d_p \rho_p}} f(T, \varphi).$$ \hspace{1cm} (10)

In which $f(T, \varphi)$ considers the effect of volume coefficient and temperature and corresponds to the results of the model of experimental measurement proposed by Moghadami and Riasi [36] for the Al$_2$O$_3$–H$_2$O nanofluid:

$$f(T, \varphi) = (-0.8467 \varphi + 0.0753) \times T + (237.67 \varphi - 21.998).$$ \hspace{1cm} (11)

Also in Eq. (10), $\beta$ is a semi empirical parameter, which is evaluated as

$$\beta = \begin{cases} 0.0137 \times (100\varphi)^{-0.8229} & \text{for } \varphi < 1\% \text{ for } \text{Al}_2\text{O}_3-\text{H}_2\text{O}, \\ 0.0017 \times (100\varphi)^{-0.0841} & \text{for } \varphi > 1\% \text{ for } \text{Al}_2\text{O}_3-\text{H}_2\text{O}, \end{cases}$$ \hspace{1cm} (12)

since suggested models for calculation of the conductivity of TiO$_2$–H$_2$O nanofluid do not have an acceptable accuracy in predicting conductivity.
As the size of nanoparticle, volume concentration, and operating temperature of the nanofluid of Yiamsawasd et al.'s [13] work are in accordance with the nanofluid used in the present study, their experimental results are selected for calculation of the conductivity of the TiO₂–H₂O nanofluid.

**Experimental Data Calculation**

Heat flux applied to the bottom surface of the heat sink has been calculated from the following equation:

\[
q = \rho_n \mathcal{C}_p n_f (T_{out} - T_{in}),
\]

where \(q''\) is amount of heat flux and is given by

\[
q'' = \frac{q}{wL}.
\]

The temperature on the bottom side of the heat sink is measured using four thermocouples placed in the heat sink. The bulk temperature of the fluid is calculated by the following equation:

\[
T_m = \frac{T_{in} + T_{out}}{2}.
\]

The local convective heat transfer coefficient \(h_i\), which is based on absorbed heat by the nanofluid at four points of the heat sink, is correlated with

\[
h_i = \frac{q''wL/N_C (\pi DL + 3 \pi dl - 3(\frac{1}{2} + \sqrt{2}) \pi d^2)}{(T_{w_i} - T_m)}.
\]

The average convective heat transfer coefficient \((h_{avg})\) is calculated by

\[
h_{avg} = \frac{\sum_{i=1}^{3} \left(\frac{h_i + h_{i+1}}{2}\right)(x_{i+1} - x_i)}{\sum_{i=1}^{3}(x_{i+1} - x_i)}.
\]

The dimensionless Nusselt number, which is a parameter for evaluating the thermal performance of the heat sink, is the ratio of convective heat transfer to conductive heat transfer and is determined by

\[
Nu = \frac{h_{avg} D}{K_{nf}}.
\]

Another parameter that plays an important role for evaluation of thermal performance is thermal resistance, which is expressed by Eq. (19):

\[
R_{th} = \frac{T_{w_{max}} - T_{in}}{q};
\]

\(T_{in}\) is the inlet fluid temperature, and \(T_{w_{max}}\) is the maximum temperature on the bottom surface of the heat sink.
A MINIATURE TANGENTIAL HEAT SINK

A dimensionless Reynolds number that expresses flow characteristics is determined according to channel diameter and thermophysical properties of the nanofluid, as below:

$$Re = \frac{\rho_{nf} V D}{\mu_{nf}}.$$  \hspace{1cm} (20)

The consumption pumping power is used to express hydrodynamic performance of the heat sink. Pressure loss caused by the passing of cooling fluid through the heat sink is multiplied to the amount of volume flow rate to calculate the consumption pumping power as

$$P.P = Q(P_{out} - P_{in}).$$  \hspace{1cm} (21)

Uncertainty Analysis

It is essential to calculate uncertainties of measurement because measuring physical quantities, such as temperature, pressure differences, and volume flow rate, always comes with an accuracy of measuring instruments. So the amount of uncertainty in the presented results is caused by accuracy in the measuring of physical quantities.

Since thermophysical properties of nanofluid is one the most effective parameters on the presented results and there is no experimental measurement in this field for this research, the amount of this parameter has been considered as 2% for thermal conductivity [37] and 1% for other thermophysical parameters [28]. The amount of uncertainties has been presented in Table 3.

Using the following equations and data of Table 3, the uncertainties of experimental results, including convective heat transfer coefficient, thermal resistance, pumping power, and Nusselt number, are calculated. The achieved results are the uncertainties of the mentioned quantities expressed in Table 4:

$$\frac{\delta q}{q} = \left[ \frac{\delta \rho_{nf}}{\rho_{nf}} \right]^2 + \left( \frac{\delta Q}{Q} \right)^2 + \left( \frac{\delta C_{p,nf}}{C_{p,nf}} \right)^2 + \left( \frac{\delta(T_{out} - T_{in})}{(T_{out} - T_{in})} \right)^2 \right]^{1/2},$$  \hspace{1cm} (22)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Uncertainty (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density, $\rho_{nf}$ ($Kg/m^3$)</td>
<td>± 1.0</td>
</tr>
<tr>
<td>Thermal conductivity, $k_{nf}$ ($W/mK$)</td>
<td>± 2.0</td>
</tr>
<tr>
<td>Viscosity, $\mu_{nf}$ (Pa.s)</td>
<td>± 1.0</td>
</tr>
<tr>
<td>Specific heat, $(C_p)_{nf}$ ($j/kg K$)</td>
<td>± 1.0</td>
</tr>
<tr>
<td>Temperature, $T$ (°C)</td>
<td>± 0.35</td>
</tr>
<tr>
<td>Differential pressure, $P$ (Pa)</td>
<td>± 5.0</td>
</tr>
<tr>
<td>Flow rate, $Q$ ($m^3/s$)</td>
<td>± 7.2</td>
</tr>
<tr>
<td>Hydraulic diameter, $D$ (m)</td>
<td>± 1.0</td>
</tr>
<tr>
<td>Width of heat sink, $w$ (m)</td>
<td>± 1.0</td>
</tr>
<tr>
<td>Length of heat sink, $L$ (m)</td>
<td>± 1.0</td>
</tr>
<tr>
<td>Diameter of entrance channel, $d$ (m)</td>
<td>± 1.0</td>
</tr>
<tr>
<td>Length of entrance channel, $l$ (m)</td>
<td>± 1.0</td>
</tr>
</tbody>
</table>
\[ \delta h_{avg} / h_{avg} = \left[ \left( \frac{\delta q}{q''} \right)^2 + \left( \frac{\delta W}{W} \right)^2 + \left( \frac{\delta L}{L} \right)^2 + \left( \frac{\delta D}{D} \right)^2 + \left( \frac{\delta d}{d} \right)^2 + \left( \frac{\delta l}{l} \right)^2 \right]^{1/2} + \left( \frac{\delta(T_w - T_m)}{(T_w - T_m)} \right)^2 \]  

\[ \delta R_{th} / R_{th} = \left[ \left( \frac{\delta(T_{w_{max}} - T_{in})}{(T_{w_{max}} - T_{in})} \right)^2 + \left( \frac{\delta q}{q} \right)^2 \right]^{1/2} \] 

\[ \delta P.P / P.P = \left[ \left( \frac{\delta Q}{Q} \right)^2 + \left( \frac{\delta(P_{out} - P_{in})}{(P_{out} - P_{in})} \right)^2 \right]^{1/2} \] 

\[ \delta Nu / Nu = \left[ \left( \frac{\delta h_{avg}}{h_{avg}} \right)^2 + \left( \frac{\delta D}{D} \right)^2 + \left( \frac{\delta k_{nf}}{k_{nf}} \right)^2 \right]^{1/2} \]  

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\[ \frac{\delta R_{th}}{R_{th}} = \left[ \left( \frac{\delta(T_{w_{max}} - T_{in})}{(T_{w_{max}} - T_{in})} \right)^2 + \left( \frac{\delta q}{q} \right)^2 \right]^{1/2} \]  

\[ \frac{\delta P.P}{P.P} = \left[ \left( \frac{\delta Q}{Q} \right)^2 + \left( \frac{\delta(P_{out} - P_{in})}{(P_{out} - P_{in})} \right)^2 \right]^{1/2} \]  

\[ \frac{\delta Nu}{Nu} = \left[ \left( \frac{\delta h_{avg}}{h_{avg}} \right)^2 + \left( \frac{\delta D}{D} \right)^2 + \left( \frac{\delta k_{nf}}{k_{nf}} \right)^2 \right]^{1/2} \]  

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\[ \delta h_{avg} / h_{avg} = \left[ \left( \frac{\delta q}{q''} \right)^2 + \left( \frac{\delta W}{W} \right)^2 + \left( \frac{\delta L}{L} \right)^2 + \left( \frac{\delta D}{D} \right)^2 + \left( \frac{\delta d}{d} \right)^2 + \left( \frac{\delta l}{l} \right)^2 \right]^{1/2} + \left( \frac{\delta(T_w - T_m)}{(T_w - T_m)} \right)^2 \]  

\[ \delta R_{th} / R_{th} = \left[ \left( \frac{\delta(T_{w_{max}} - T_{in})}{(T_{w_{max}} - T_{in})} \right)^2 + \left( \frac{\delta q}{q} \right)^2 \right]^{1/2} \]  

\[ \delta P.P / P.P = \left[ \left( \frac{\delta Q}{Q} \right)^2 + \left( \frac{\delta(P_{out} - P_{in})}{(P_{out} - P_{in})} \right)^2 \right]^{1/2} \]  

\[ \delta Nu / Nu = \left[ \left( \frac{\delta h_{avg}}{h_{avg}} \right)^2 + \left( \frac{\delta D}{D} \right)^2 + \left( \frac{\delta k_{nf}}{k_{nf}} \right)^2 \right]^{1/2} \]  

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**RESULTS AND DISCUSSION**

**Heat Sink Base Temperature**

Minimizing temperature and achieving thermal uniformity are the basic principles in electronic device cooling. Figure 5 shows the effect of using an impinging liquid jet and Al$_2$O$_3$–H$_2$O, TiO$_2$–H$_2$O nanofluids on the decrease in the heat sink base temperature. Using nanofluid as a cooling fluid not only causes a decrease in the heat sink base temperature but also causes thermal uniformity in the base of the heat sink, because, by an increase in nanofluid temperature along the heat sink, thermal conductivity of nanofluid increases and causes increase in heat absorptivity. Previous researchers [20] showed that an increase in Reynolds number causes a decrease in temperature of the base of the heat sink. The measured decrease in the base temperature of the heat sink is 2.2°C for Al$_2$O$_3$–H$_2$O and 1.6°C for TiO$_2$–H$_2$O in comparison with pure distilled water in this research.
An increase in absorptivity by using nanofluid instead of using pure distilled water is the reason for the decrease in heat sink base temperature. Nanofluid conductivity increases with increase in volume concentration, so heat absorptivity increases in the higher nanofluid concentration.

Convection Heat Transfer Coefficient of Nanofluids

The effects of Reynolds number and volume concentration of nanofluids on average convective heat transfer coefficient are shown in Figure 6. This figure shows the heat transfer coefficient increases by an increase in volume concentration from 0.5 to 2% of a Reynolds number range of 210 to 1,100. The results revealed an 11 and 14% increase in convective heat transfer coefficient compared with pure distilled water for Al2O3–water and TiO2–water, respectively, in the highest volume concentration. The main reason of increase in convective performance can be contributed to the thermophysical properties of nanofluid over the pure distilled water. Thermal conductivity and Brownian motion increase with increase in volume concentration, causing an increase in the convective heat transfer coefficient of nanofluid compared with pure distilled water.

Figure 7 gives the comparison of heat transfer coefficients between Al2O3–H2O and TiO2–H2O nanofluids. The results shows that the maximum convective heat transfer coefficient
Figure 6. Influence of Reynolds number and volumetric concentration of nanofluids on convective heat transfer coefficient of the heat sink: (a) Al₂O₃–H₂O and (b) TiO₂–H₂O.

Figure 7. Comparison of heat transfer coefficients of Al₂O₃–H₂O and TiO₂–H₂O.
coefficient of $\text{Al}_2\text{O}_3\text{–H}_2\text{O}$ is 3% higher than those of $\text{TiO}_2\text{–H}_2\text{O}$. The higher conductivity of $\text{Al}_2\text{O}_3\text{–H}_2\text{O}$ compared with $\text{TiO}_2$ in the same operating temperature can be considered the reason of this enhancement.

**The Nusselt Number**

The Nusselt number expresses the convective performance of the cooling fluid at the heat sink. An increase in Reynolds number causes flow regime change from boundary layer flow to turbulent flow; hence, the convective heat transfer coefficient and Nusselt number increase. Increase in volume concentration increases heat absorptivity of the nanofluid more than its conductivity and, therefore, increases the Nusselt number. Figure 8 shows the change in Nusselt number in different volumes of $\text{Al}_2\text{O}_3\text{–H}_2\text{O}$ and $\text{TiO}_2\text{–H}_2\text{O}$ nanofluids. As shown in Figure 8, the Nusselt numbers in a volume concentration of 2% for $\text{Al}_2\text{O}_3\text{–H}_2\text{O}$ and $\text{TiO}_2\text{–H}_2\text{O}$ are 8.3 and 6.8%, respectively, higher than pure distilled water at the same Reynolds number.

Considering fluid entry from the tangential surface of the channel in the heat sink, an increase in Reynolds number causes the flow be turbulent, and therefore, the convective

![Figure 8](image-url)  
*Figure 8. Influence of Reynolds number and volumetric concentration of nanofluids on Nusselt number: (a) $\text{Al}_2\text{O}_3\text{–H}_2\text{O}$ and (b) $\text{TiO}_2\text{–H}_2\text{O}$.***
heat transfer coefficient will be increased. So the increase in convective performance caused by increase in Nusselt number by using Al$_2$O$_3$–H$_2$O and TiO$_2$–H$_2$O justify the use of nanofluids rather than convectional coolants.

**Thermal Resistance**

Figure 9 presents the thermal resistance of Al$_2$O$_3$–H$_2$O and TiO$_2$–H$_2$O in different volume concentrations for a Reynolds number range of 210 to 1,100. This figure shows that the thermal resistance decreases with an increase in Reynolds number; it also reveals that the increase in volume concentration causes a decrease in convective thermal resistance compared with pure distilled water.

The increase in convective heat transfer coefficient and thermal dispersion of the nanofluid can be expressed as the main reasons of decrease in convective thermal resistance with developing the volume concentration.

![Figure 9](image-url)

**Figure 9.** Influence of Reynolds number and volumetric concentration of nanofluids on thermal resistance: (a) Al$_2$O$_3$–H$_2$O and (b) TiO$_2$–H$_2$O.
In this research, 12 and 15% decreases in convective thermal resistance, respectively, for Al$_2$O$_3$–H$_2$O and TiO$_2$–H$_2$O have been achieved in the highest volume concentration and Reynolds number compared with pure distilled water.

Reynolds number enhancement causes an increase in Browning motion of nanoparticles, which causes an increase in thermal transportation of nanofluid. Heat transfer modification causes a decrease in heat sink base temperature and also a decrease in convective thermal resistance. By comparing Al$_2$O$_3$–H$_2$O and TiO$_2$–H$_2$O, it can be seen that the thermal resistance of Al$_2$O$_3$–H$_2$O is less than 3% that of TiO$_2$–H$_2$O.

**Pumping Power**

The injection of fluid from the tangential surface of the heat sink, which passes through the channel, causes pressure drop at the heat sink. Therefore, the flow requires additional pumping power to compensate this pressure drop. Figures 10a and 10b illustrate the variation of required pumping power versus Reynolds number of different volume concentrations of Al$_2$O$_3$–H$_2$O and TiO$_2$–H$_2$O nanofluids. The increase in pumping power to the increase in Reynolds number can be obviously inferred in Figures 10a and 10b.

Figure 10. Influence of Reynolds number and volumetric concentration of nanofluids on pumping power: (a) Al$_2$O$_3$–H$_2$O and (b) TiO$_2$–H$_2$O.
As shown in Figure 10, pumping power increases with increase in volume percentage and Reynolds number. Adding nanoparticles to the water-based fluid increases the density of the fluid, and this is the main reason for the enhancement of pumping power of nanofluid compared with pure distilled water. Also, the increase in nanofluid viscosity due to surface absorption and the nanolayer cluster around the nanoparticle is another reason for the increase in pumping power. The results show that there are 30 and 45% increases in pumping power, respectively, for Al₂O₃–H₂O and TiO₂–H₂O compared with pure distilled water. Enhancement in pumping power can be compensated with enhancement in the thermal performance achieved by using a nanofluid instead of pure distilled water.

**Combined Investigation of Pumping Power and Heat Transfer Coefficient Enhancement**

Adding nanoparticles to the base fluid leads to increasing density and viscosity, and thus, pressure drop and pumping power increase. Parameter \( \psi = \frac{h_{np}}{h_{H_2O}} \) has been introduced to investigate the proportion of heat transfer coefficient to pumping power enhancement caused by adding nanoparticles to the base fluid. Figures 11a and 11b show

![Figure 11](image)

**Figure 11.** Influence of Reynolds number and volumetric concentration of nanofluids on parameter \( \psi \): (a) Al₂O₃–H₂O and (b) TiO₂–H₂O.
the variation of this parameter versus Reynolds number for Al₂O₃–H₂O and TiO₂–H₂O nanofluids, respectively. It is obvious that appropriate Reynolds number range is once the amount of \( \psi \) is more than 1. Figure 11 reveals that the appropriate Reynolds number range is 300 to 700, and an increasing in the volume concentration of nanofluid decreases this range. The comparison of Figures 11a and 11b shows that in the same volume concentration of nanofluid, the appropriate Reynolds number range for Al₂O₃–H₂O is wider than that for the TiO₂–H₂O nanofluid. Comparing the results for Al₂O₃–H₂O and TiO₂–H₂O nanofluids also discloses that \( \psi \) for Al₂O₃–H₂O is higher than that for TiO₂–H₂O nanofluids at the same Reynolds number, because Al₂O₃–H₂O has a higher heat transfer coefficient and lower pumping power.

**COMPARATIVE PERFORMANCE**

The present experimental study is compared with the previous numerical study [21] in Table 5. The heat sink is used in the previous study is more compact and the number of channels is greater. The previous numerical study has a higher heat flux and volume concentration, and the Reynolds number of the present study is higher. Although the results of the present study show a similar range of Nusselt number, it shows higher pressure drop as a consequence of the higher Reynolds number of nanofluid flow. The difference in results can be attributed to the differences in geometrical dimension, volume concentration, and flow characteristics of nanofluids.

Figure 12 shows the comparison between the results of two experimental studies. The heat transfer coefficient of the water base fluid in the tangential heat sink and conventional heat sink [24] has been compared, and they have exactly the same geometrical dimensions.

Figure 12 reveals that the increase in Reynolds number from 330 to 1,100 increases the heat transfer coefficient of water flow in both tangential and conventional heat sinks by 42.6 and 25.7%, respectively. The heat transfer coefficient of water flow in the tangential heat sink at a Reynolds number of 1,100 is 131% higher than that in the conventional heat sink. It can thus be concluded that the tangential heat sink with the impinging jet method has higher thermal performance compared to the conventional heat sink.

<table>
<thead>
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<th>Parameter</th>
<th>Present study</th>
<th>Present study</th>
<th>Tabrizi and seyf [21]</th>
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<tr>
<td>Nanofluid</td>
<td>Al₂O₃–H₂O</td>
<td>TiO₂–H₂O</td>
<td>Al₂O₃–H₂O</td>
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<td>0.5–2.0%</td>
<td>1.0–4.0%</td>
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<td>Diameter of nanoparticle (nm)</td>
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<td>40 × 40 × 8.5</td>
<td>10 × 10 × 1.5</td>
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<td>4.0</td>
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<tr>
<td>Hydraulic diameter of inlet cross-section (mm)</td>
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<td>5</td>
<td>9</td>
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<td>Reynolds number variation</td>
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<td>210–1,100</td>
<td>50–200</td>
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<td>240</td>
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<td>Nusselt number variation</td>
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<td>25.3–43.9</td>
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<td>Pumping power variation (W)</td>
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<td>0.00017–0.0243</td>
<td>0.000002–0.000076</td>
</tr>
</tbody>
</table>

Table 5. Comparisons between thermal performance of two tangential heat sinks
CONCLUSION

In this research, overall thermal and hydrodynamic performance of a tangential miniature heat sink has been investigated experimentally. Two nanofluids—Al$_2$O$_3$–H$_2$O and TiO$_2$–H$_2$O—have been produced and used at four different volume concentrations of 0.5, 1, 1.5, and 2% as the working fluids. The achieved results show an increase in convective heat transfer by using nanofluid instead of pure distilled water. Also, enhancement in pumping power can be negligible compared with the increase in thermal performance and decrease in thermal resistance. The following conclusions are drawn from the experimental results.

1. By using Al$_2$O$_3$–H$_2$O and TiO$_2$–H$_2$O instead of pure distilled water, the average temperature of the heat sink floor decreases 2.2°C and 1.6°C, respectively. In addition, using nanofluids causes thermal uniformity at the base of the heat sink.

2. Fifteen and 12% increases in convective heat transfer coefficient by using Al$_2$O$_3$–H$_2$O and TiO$_2$–H$_2$O have been seen, respectively, compared with pure distilled water in a Reynolds number range of 210 to 1,100. The Nusselt number increases 8.3 and 6.8% as well, respectively, for the mentioned nanofluids.

3. By using Al$_2$O$_3$–H$_2$O and TiO$_2$–H$_2$O in the maximum volume concentrations, thermal resistance of tangential miniature heat sink decreases 15 and 12%, respectively.

4. The enhancement in consumption pumping power by using Al$_2$O$_3$–H$_2$O and TiO$_2$–H$_2$O in a volume concentration of 2% has been reported as 30 and 45%, respectively. This enhancement can be attributed to the increase in density and viscosity of working nanofluids compared with pure distilled water.

5. A comparison of the heat transfer coefficient of water flow in the tangential heat sink and the conventional heat sink at a Reynolds number of 1,100 shows the heat transfer coefficient of water flow in the tangential heat sink is 131% more than of that of water flow in the conventional heat sink.

![Figure 12. Comparison between heat transfer coefficient of water base fluid in tangential heat sink and conventional heat sink.](image-url)
6. Parameter $\psi$, which is the proportion of $h_{nf}/h_{H2O}$ to $P_{nf}/P_{H2O}$, is a presented scale for simultaneous measurement of enhancement both in the heat transfer coefficient and pumping power. The results show that there is an appropriate Reynolds number range in which $\psi$ has the amount more than 1. This range decreases with an increase in volume concentration of nanofluid and varies from 300 to 700 for 0.5% $Al_2O_3$ nanofluid.

Finally, considering the achieved experimental results, using the jet impinging method and $Al_2O_3$–$H_2O$ and $TiO_2$–$H_2O$ nanofluids instead of pure distilled water in the heat sink is suggested for cooling of electronic elements with high heat flux.

REFERENCES


