Experimental analysis of energy and friction factor for titanium dioxide nanofluid in a water block heat sink

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A B S T R A C T

Heat dissipation is a critical issue in modern electronic components, due to the technological advances that have reduced their size and caused their heat flux to rise. Different types of heat sinks are promising for cooling of such electronics and nanofluid can enhance the cooling performances. In this present work, a titanium dioxide (TiO2/water) nanofluid (with a volume fraction of 0.1%) is prepared by dispersing nanoparticles in distilled water. The nanofluid is then passed through the heat sink at various flow rates (1.00, 1.25, and 1.50 L/min). The interface temperature of the water block was reduced up to 6.40°C by using the nanofluid, as compared to water. Due to the decline of interface temperature the heat transfer coefficient was improved by 20.82% compared to water. The maximum energy efficiency found 77.56% for nanofluid. Therefore, the titanium dioxide nanofluid is a superior coolant than pure water. Moreover, the heat transfer effectiveness and energy effectiveness were found highest at the minimum flow rate of 1.00 L/min.

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1. Introduction

Advances in technology and consumer demands are driving electronics industries to create new designs with a higher level of integration, compactness, and performance. As a result, the need to dissipate the heat generated by these electronic devices and circuitry, to prevent premature failure and improve reliability, has also been elevated. Researchers have introduced the liquid cooling system for electronics items, and nanofluids are promising to increase the heat dissipation rate of these systems.

A pioneering analytical study by Ijam and Saidur [1] reported the effect of using TiO2-water and SiC-water nanofluids on heat flux and pumping power of a minichannel heat sink. Selvakumar and Suresh [2] conducted experiments with a thin channeled copper water block having the overall dimension of $55 \times 55 \times 19 \text{ mm}$. A 29.63% higher convective heat transfer coefficient (HTC) was observed by using the nanofluid composed of 0.2 vol.% of copper oxide mixed with water, compared to deionized water alone. Similarly, Jung et al. [3] observed up to 32% higher convective HTC compared to distilled water (in a laminar flow regime) without significant friction loss for the nanofluid composed of 1.8% volume fraction of an aluminum oxide ($\text{Al}_2\text{O}_3$) mixed in water. Pak and Cho [4] measured the convective HTC of aluminum oxide ($\gamma$-$\text{Al}_2\text{O}_3$) and titanium dioxide (TiO2) nanoparticles dispersed in water. Their experimental results showed that the HTC increased in direct proportion to the volume fraction of nanoparticles and the Reynolds number of the nanofluid. Abbasi et al. [5] studied the influence of Cu-water nanofluid in a microchannel heat sink (MCHS). The thermal dispersion coefficient and Reynolds number on thermal fields were investigated. Moreover, the impact of flow turbulence on the heat transfer rate was considered. They found that the overall Nusselt number was increased with increasing nanoparticle concentration, thermal dispersion coefficient, and Reynolds number.

Nguyen et al. [6] experimentally studied the heat transfer enhancement behavior of a nanofluid (aluminum oxide mixed with water) for cooling the microprocessors. A maximum of 40% HTC was found to be increased compared to water. Mohammed et al.
using the dual method of helical coils and nanofluids, which caused an enhancement of energy efficiency. They applied water with Al2O3 (35 nm) and TiO2 (50 nm) nanoparticles (0.25–1.0% volume fractions). Hamut et al. [15] studied the performance of a coolant circuit in a vehicle. Second thermodynamic law analysis was used to examine areas on the system with low exergy efficiencies. Zaki et al. [16] used second law analysis in a water-chiller cooler that was used to cool the intake air in gas turbines. An average of 8.5% exergetic power gain ratio was dropped. Khaleduzzaman et al. [17] studied the exergy and entropy generation in a water block operated with 0.10 vol.% of TiO2–water nanofluid and changed the flow rate from 1.0 to 1.5 L/min. They observed a maximum of 39.63% exergy efficiency enhancement.

In this research, the thermal performance of a titanium dioxide nanofluid with a thin-channeled water block was experimentally measured and analyzed. The base temperature difference, thermal resistance, heat transfer coefficient, heat transfer effectiveness, Nusselt number and Reynolds number variation, energy efficiency, energy effectiveness, friction factor, pressure drop, pumping power, and performance index are calculated and compared to water. These comparative results were used to define the cooling performance of the titanium dioxide nanofluid.

2. Methodology

2.1. Experimental setup

The schematic diagram of the experimental set-up is shown in Fig. 1. This Fig. 1 shows a closed-loop electronics cooling system, consisting of a water block heat sink, cooling fluid loop and data-acquisition system. The connections within the piping system and test section are designed so that the parts can be changed or repaired easily. The closed loop consists of the storage tank, pump, volumetric flow meter, and an air cooled radiator.

The process begins by filling the storage tank with the working fluid (at ambient conditions). A pump (model: XSPC X20 750) is
used to transfer the fluid out of the storage tank, through a flow meter, and into the water block. The coolant absorbs heat from the heat sink, increasing its temperature. An air-cooled radiator type cooler (model: SSPC RS360) is used to decrease the temperature of the coolant before it enters a storage tank for recirculation. An adjustable volumetric flow meter was used to maintain the coolant flow rates. A pressure transducer is also installed on the heat sink to measure the pressure drop.

The schematic and images of the water block heat sink are shown in Fig. 2. The XSPC RASA CPU water block was used (details listed in Table 1). The size of copper base water block is \(94 \times 94 \times 20\) mm. The total number of the thin channel is 55 (each side of the cross section). The length of thin channels is 66 mm and channel width 0.3 mm and channel height 2 mm. The cross-sectional area of the channel is \(33 \times 33\) mm. The inlet flow is directed through the thin channels of the water block. The inner diameter of inlet and outlet tube in water block is 6.35 mm.

Two controllable, cartridge heaters (200 W each) were positioned just below the heat sink, separated by a bottom plate, to provide the heat flux. The heaters of the system have the control valve to fix the desired input level of heat in the system. In this system, heaters maximum capacity was 80 °C. Coolant temperatures and pressures were measured at different positions using resistance temperature detector (RTD) thermocouples and a pressure transducer connected to a data logger, as shown in Fig. 1. One thermocouple was used to record the heater temperature \(T_h\) (to control the heat generation), and the other five were used to measure the average base temperature \(T_{w,avg}\) of the heat sink. It is considered that the system does not lose heat as the heat sink was covered by an insulating box. The box was fabricated from Teflon, and glass wool was used inside the box for insulation. Insulating material covered the top and four sides of the water block heat sink.

### 2.2. Preparation of nanofluid

Pure 99.90% titanium dioxide \((\text{TiO}_2)\) nanoparticles (21 nm diameter, manufactured by Sigma Aldrich, USA) were mixed with
pure distilled water to form a nanofluid. The nanoparticles were measured out to a precise, pre-calculated amount (using Eq. (1)) with an analytical balance (HR-250 AZ, AND, Japan), in order to give a 0.1% volume fraction mixture.

\[
\phi = \frac{m_n}{\rho_n} = \frac{m_n}{\rho_n} + \frac{m_f}{\rho_f}
\]  

where \( \phi, m_n, m_f, \rho_n, \rho_f \) refers to volume fraction, the mass of nanoparticles, the weight of the base fluid, the density of nanoparticle, and density of the base fluid, respectively.

The nanoparticles were mixed with the distilled water into a beaker and ultrasonicated using a sonic dismembrator system (FB505, Fisher Scientific, USA). The power and frequency of the machine were 500 W and 20 kHz, respectively. For better colloidal dispersion, the ½-in. standard tip, 50% amplitude and 2 s ON and 2 s OFF pulses were used, and the process was continued for two hours [18]. A refrigerated circulating water bath (C-DRC 8, CPT Inc., South Korea) was used to provide a fixed temperature (15°C–176°C) during the ultrasonication process to avoid vaporization. The nanofluid was found stable more than one week by this method without adding any surfactant.

### 2.3. Thermophysical properties of nanofluid

Several thermophysical properties of pure distilled water and the titanium dioxide nanofluid were experimentally measured (without using any surfactant) at 30°C. The results are shown in Table 2. Thermal conductivity was measured using the KD2 pro thermal properties analyzer (Decagon, USA). Density was determined by using a DA130 density meter (Kyoto Electronics, Japan). An LDV-III ultra-programmable rheometer (Brookfield, USA) was used to measure the viscosity. A DSC 4000 differential scanning calorimeter (Perkin Elmer, USA) was used to determine the specific heat of the nanofluid.

### 2.4. Governing equations

The following Eq. (2) was used to calculate the base temperature of the heat sink. In this equation, the conduction heat transfer through the base height (\( H_b \)) is considered.

\[
T_b = T_{b(as,x)} - \left( \frac{QH_b}{k_{hs}A_b} \right)
\]

The area of the base of the heat sink (\( A_b \)) is obtained by Eq. (3).

\[
A_b = \ln(W_{ch} + W_{fm})
\]

The thermal resistance (\( R_{th} \)) is defined by Eq. (4).

\[
R_{th} = \frac{T_b - T_{nf,in}}{Q}
\]

From the collected data of temperatures and mass flow rates, the convective HTC of the heat sink (\( h \)), can readily be determined by Eq. (5).

\[
h = \frac{Q}{A_f(T_b - T_{nf})}
\]

In Eq. (5), the mean temperature of nanofluid (\( T_{nf} \)) is determined using Eq. (6).

\[
T_{nf} = \frac{T_{nf,in} + T_{nf,out}}{2}
\]
Likewise, the surface area \( (A_d) \) in Eq. (5) can be determined using Eq. (7).

\[
A_d = nW_{ch}L + 2nH_{fin}W_{ch}L
\]  

(7)

In Eq. (7) the fin efficiency is considered 100\% due to high thermal conductivity of copper. The fin efficiency of water block \( (\eta_{fin}) \) was computed by Eq. (8) and (9).

\[
m \times H_{ch} = \sqrt{\frac{2h}{k_{hs}W_{fin}}} \times H_{ch}
\]  

(8)

where \( k_{hs} \) is thermal conductivity of the copper MCHS.

\[
\eta_{fin} = \frac{\tanh(m \times H_{ch})}{m \times H_{ch}}
\]  

(9)

For the better thermal performance analysis, the heat transfer effectiveness \( (\varepsilon) \) can be quantified using Eq. (10) [19].

\[
\varepsilon = \frac{h_{ref}}{h_{ref}}
\]  

(10)

The Nusselt number is defined by Eq. (11).

\[
Nu = \frac{hD_a}{k_{ref}}
\]  

(11)

The Reynolds number is defined by Eq. (12).

\[
Re = \frac{\rho_u u_{in}D_h}{\mu_{in}}
\]  

(12)

where the mean fluid velocity \( (u_{in}) \) can be determined from the following Eq. (13).

\[
u_{in} = \frac{m}{\rho_{in}A_c}
\]  

(13)

The energy efficiency of the heat sink \( (\eta_{1st}) \) is determined by Eq. (14) [20].

\[
\eta_{1st} = \frac{mc_p(T_{out} - T_{in})}{Q}
\]  

(14)

The energy effectiveness \( (\varepsilon) \) of the water block heat sink is determined from Eq. (15) [19].

\[
\varepsilon = 1 - \frac{\left(\frac{T_{br}-T_{in}}{T_{br}-T_{max}}\right)}{\left(\frac{T_{br}-T_{in}}{T_{br}-T_{max}}\right)}
\]  

(15)

The friction factor \( (f) \) of the water block for laminar flow \( (Re < 2000) \) is computed by Eq. (16) [21].

\[
f = \frac{96(1 - 1.3533z + 1.94672z^2 - 1.7012z^3 + 0.9564z^4)}{Re}
\]  

(16)

where the new dimensionless variable \( (z) \) is defined as the ratio of channel height to width and is calculated by Eq. (17) [22].

\[
\alpha = \frac{H_{ch}}{W_{ch}}
\]  

(17)

The pressure reduction \( (\Delta P) \) through the microchannel heat sink is determined by Eq. (18).

\[
\Delta P = f \frac{L}{D_h} \frac{\rho u^2}{2}
\]  

(18)

The required pumping power \( (P_p) \) for the nanofluid through the channel is calculated using Eq. (19).

\[
P_p = \frac{m}{\rho_{in}} \Delta P
\]  

(19)

2.5. Uncertainty analysis

The uncertainties of measurements are summarized in Table 3. The uncertainties of the analysis were obtained by Eq. (20), in terms of uncertainties associated with the related independent variables.

\[
W_k = \left( \frac{\delta R}{\Delta X_1} W_1 \right)^2 + \left( \frac{\delta R}{\Delta X_2} W_2 \right)^2 + \cdots + \left( \frac{\delta R}{\Delta X_n} W_n \right)^2 \right)^{1/2}
\]  

(20)

In Eq. (20), \( R \) is a function of the independent variables \( X_1, X_2, \ldots, X_n \) and \( W_1, W_2, \ldots, W_n \) are uncertainties of the independent variables. In this equation, \( W_k \) is the uncertainty of dependent variables, such as HTC, energy efficiency, effectiveness, etc. The maximum errors attained are ±0.92\% for the HTC \( (h) \), ±2.21\% for the energy efficiency \( (\eta_{1st}) \), and ±6.35\% for the effectiveness \( (\varepsilon) \).

3. Results and discussion

3.1. Temperature and thermal resistance

The measured temperatures at the base of the heat sink for varying flow rates of the titanium dioxide nanofluid and pure distilled water are shown in Fig. 3. Fig. 3 shows a lower base temperature when the nanofluid is used. The highest temperature difference was observed at 1.0 L/min flow rate where the base temperature were 77.32 °C and 70.92 °C respectively for water and the titanium dioxide nanofluid. The lowest temperature difference was observed at 1.5 L/min flow rate when base temperature were 66.12 °C and 63.95 °C respectively for water and the titanium dioxide nanofluid. This is because the nanofluids eliminate more heat from the base of the heat sink in comparison to water (base fluid). As a result, the base temperature is kept at a minimum level [2]. At higher flow rates, base temperatures were found to be decreased for both fluids. This is because fluid cannot get enough time or resistance to absorb heat at higher flow rates.

The effect of fluid flow rate of nanofluid on the thermal resistance of the heat sink is shown in Fig. 4. It can be seen in Fig. 4 that the highest thermal resistance was observed for the water at 1.0 L/min flow. The highest thermal resistance found for water at all flow rates because of the lower thermal conductivity values of water. The thermal resistance was found to be decreased with the increase of flow rates. The minimum thermal resistance was found at 1.5 L/min flow rate for the nanofluid. The highest thermal resistance reduction difference was 13.48\% at 1.0 L/min flow rate of the nanofluid. At 1.5 L/min flow rate for the nanofluid, the highest thermal resistance was found to be decreased with the increase of flow rates. The minimum thermal resistance was found at 1.5 L/min flow rate for the nanofluid. The highest thermal resistance reduction difference was 13.48\% at 1.0 L/min flow rate of the nanofluid. This substantial decrease in thermal resistance gives a strong inspiration to apply nanofluid as a coolant for electronics devices.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Uncertainty (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density ( (p) )</td>
<td>±1.0</td>
</tr>
<tr>
<td>Thermal conductivity ( (k) )</td>
<td>±1.0</td>
</tr>
<tr>
<td>Viscosity ( (\mu) )</td>
<td>±1.0</td>
</tr>
<tr>
<td>Specific heat ( (c_p) )</td>
<td>±1.0</td>
</tr>
<tr>
<td>Pressure drop ( (\Delta P) )</td>
<td>±2.5</td>
</tr>
<tr>
<td>Ambient temperature ( (T_a) )</td>
<td>±0.1</td>
</tr>
<tr>
<td>Electric power ( (Q) )</td>
<td>±3.5</td>
</tr>
<tr>
<td>Mass flow rate ( (m) )</td>
<td>±0.3 to ±2.2</td>
</tr>
<tr>
<td>Inlet temperature difference ( (T_{in}-T_{out}) )</td>
<td>±1.5 to ±5.0</td>
</tr>
<tr>
<td>Outlet temperature difference ( (T_{out}-T_{in}) )</td>
<td>±1.5 to ±5.0</td>
</tr>
<tr>
<td>Overall temperature difference ( (T_{out}-T_{in}) )</td>
<td>±0.4 to ±2.2</td>
</tr>
</tbody>
</table>
3.2. Thermal analysis

The HTC of the mini channel heat sink operated with TiO$_2$–water nanofluids were analyzed at different flow rates. Fig. 5 shows that HTC increases with the increase of flow rates. It can be seen in Fig. 5 that HTCs are found to be higher for nanofluids because of their higher thermal conductivity. The maximum HTC was found for the nanofluid at 1.5 L/min flow rate. The enhancement percentages of the HTC for the nanofluid (as compared to water) were: 20.82%, 13.63%, and 13.85% for 1.00, 1.25, and 1.50 L/min flow rates, respectively.

The heat transfer effectiveness of the nanofluid is shown in Fig. 6. Fig. 6 demonstrates that nanofluids are capable of improving heat transfer effectiveness in the water block heat sink, compared with water. The effectiveness of using the nanofluid was decreased as the flow rate increases from 1.00 L/min until about 1.25 L/min and then slightly increases. At the initial flow rate (1.00 L/min), nanofluid effectiveness was high because of greater HTC than base fluid at the same flow rate. After that, with the increase of flow rate, the ratio of the HTC was found to be decreased at 1.25 L/min flow rate. The reason is that, with the rise of the flow rate, the nanofluid cannot get enough time to take more heat from the base of the water block, because fluid flows faster and absorbing capacity of nanofluid was reduced. However, in the case of water, the effect of flow rate on absorbing heat was not highly influenced and was almost kept the same up to 1.25 L/min. Again, after the 1.25 L/min, the higher flow rate affected the heat-absorbing capacity of water and it was going down. As a result, again effectiveness was found to be increased as it is the ratio of nanofluid by water.

Fig. 7 shows that the Nusselt number increases accordingly with the Reynolds number and flow rate. The maximum Nusselt number for the nanofluid occurred at the flow rate of 1.5 L/min. The minimum Nusselt number value occurred for water at the flow rate of 1.0 L/min. The observation found that the nanofluid produces higher Nusselt numbers rather than pure water, thus shows the greater efficiency as a better coolant to replace pure water. The maximum Reynolds number was found for water at the 1.5 L/min flow rate.

Fig. 8 shows the relationship between reducing rate of interface temperature and enhancement of HTC. It is found from the analysis that enhancement of HTC was increased with the rise of reducing the rate of interface temperature of water block (WB). The highest enhancement found at a low fluid flow rate (1.0 L/min) where maximum temperature reduces up to 6.40 °C.

3.3. Energy analysis

Energy efficiencies for the change of flow rates of the nanofluid and water were calculated by using Eq. (14) and shown in Fig. 9. The first law (energy) efficiency declines for both coolants with the increase of the flow rates. The energy efficiency was augmented by the addition of nanoparticles. The maximum energy efficiency was found 77.56% for the 0.10 vol.% of TiO$_2$–water nano-
fluid at the lowest 1.0 L/min flow rate. The fluid factors as mass flow rate, temperature difference, and specific heat are responsible for the energy efficiency. Also for a constant volume flow rate, the mass flow rate will vary, depending upon the density of the base fluid and volume fraction of the nanofluid. As the volume flow rate rises, the temperature variance falls amongst the inlet and outlet. The highest temperature variance occurred at a minimum volume flow rate. The energy efficiency declines as the volume flow rate rise since temperature variance drop with an increase in volume flow rate.

The energy effectiveness ($\varepsilon$) of the water block heat sink was defined in Eq. (15). Fig. 10 shows that the energy effectiveness increases as the flow rate increases. The maximum energy effectiveness was found 0.134 at 1.0 L/min flow rate of the nanofluid. The effectiveness declines as the flow rate increases until the flow rate reached at 1.25 L/min, after that, the effectiveness gradually increases. The energy effectiveness decreases at 1.25 L/min flow rate than 1.00 L/min due to lower nanofluid temperature. On the other side, the energy effectiveness increases at 1.50 L/min flow rate than 1.25 L/min due to lower temperature of water (base fluid).

3.4. Friction factor, pressure drop, and pumping power

Friction factor is contingent on the fluid density and therefore, also the Reynolds number. The Reynolds number has a contrary relation with friction factor. With the growth of the nanoparticle volume fraction raises the fluid mass flow rate and density and then decreases the friction factor. At the highest level of densities, the influence of volume fraction is significant over the friction factor, but the reverse goes for a minimum level of densities. The relationship between friction factors and flow rates of coolant is shown in Fig. 11. The friction factor decreases as the flow rate increases and has an increased value for the nanofluid as compared to pure water. The highest friction factor was found to be 0.22 for the nanofluid at a flow rate of 1.0 L/min. The lowest friction factor value was 0.11 for pure water at a flow rate of 1.5 L/min.
Fig. 12 shows that the amount of pressure drops increased as the volumetric flow rate rises. A high-pressure drop is an unwanted manner, which indicates the reason that parameters like the energy efficiency decreased as the flow rate increases. The maximum pressure drop (7.801 kPa) was found at a flow rate of 1.25 L/min for the nanofluid. The addition of nanoparticles increases the viscosity of nanofluid, which is the reason for higher pressure drop. The minimum pressure drop was found for water (2.887 kPa) at the lowest flow rate of 1.00 L/min. The highest-pressure drops were observed for higher flow rates. This is because when the flow rate increase, it creates more force and resistance to the flow and pressure difference is increased. The nanofluid pressure drop increment was high at lower flow rate compare with water.

The effect of flow rates on pumping power is shown in Fig. 13. The required pumping power ($P_p$) was found to be increased accordingly with the volumetric flow rate of the fluid. The required pumping power is found higher for the nanofluid than water. This difference is entirely due to the pressure drop as shown in Fig. 12. The highest required pumping power was found to be 0.19 W for the titanium dioxide nanofluid at a volumetric flow rate of 1.5 L/min. The highest required pumping power for pure water was 0.13 W at this same flow rate.

It could be noted that energy effectiveness is a key performance parameter of a heat exchanger system, which was found to be higher for the nanofluid. Nevertheless, highest pumping power was also observed for the nanofluid, and it is a negative impact.

However, the above parameters were found to be variable with flow rates. Therefore, to get the maximum benefit from the nanofluid for the heat sink, a performance index is introduced in Fig. 14, which was determined as the ratio of the amount of energy effectiveness to the amount of pumping power augmentations due to the nanofluid. It can be seen in Fig. 14 that highest performance index was observed at 1.50 L/min flow rate and the index value of 1.00 L/min flow rate was found to be almost similar to that of 1.50 L/min flow rate. The trend of performance index with flow rate was found same as heat transfer effectiveness and energy effectiveness due to the same reason of temperature of nanofluid and water phenomena at 1.25 L/min and 1.50 L/min flow rate.

4. Conclusion

In this study, temperature difference, thermal resistance, heat transfer coefficient, heat transfer effectiveness, Nusselt number and Reynolds number variation, energy efficiency, energy effectiveness, friction factor, pressure drop, pumping power, and performance index were analyzed based on experimental data by using 0.1 vol.% of TiO$_2$–water nanofluid in a box type heat sink. The flow rates were varied from 1.0–1.5 L/min. Analysis of the experimental results shows that the titanium dioxide nanofluid is a superior coolant than pure water for an electronic cooling device. The detail results are summarized below:

1. The base temperature was found to be decreased up to 6.40 °C at 1.0 L/min flow rate. Thermal resistance was also found to be decreased for the nanofluid. Both of the above parameters were found to be decreased with increasing flow rates for both fluids.
2. A higher heat transfer coefficient was observed for the nanofluid, and it was increased with increasing flow rates. The highest improvement of heat transfer coefficient was 20.82% for the nanofluid. The maximum heat transfer effectiveness for the nanofluid was observed at 1 L/min flow rate. Nusselt and Reynolds numbers were found to be increased proportionally for both fluids. A corresponding higher Nusselt number but lower Reynolds number was observed for the nanofluid and vice versa for the water.
3. The nanofluid showed higher-energy efficiencies of 78%, 75%, and 75% for the 1.00, 1.25, and 1.50 L/min flow rates, respectively. In the case of water, the efficiencies were 22%, 11%, and 9%, respectively. The highest energy effectiveness for the nanofluid was observed at 1 L/min flow rate.
4. The highest friction factor was observed for using the nanofluid, and it was found to be higher at lower flow rates for the case of both fluids. The highest pressure drop and pumping power were observed for the nanofluid. They were found to be higher at higher flow rates for the both fluids.

5. The highest performance index was observed at 1.50 L/min flow rate, and the index value of 1.00 L/min flow rate was found to be almost similar to that of 1.50 L/min flow rate.

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Conflict of interest

The authors declared that there is no conflict of interest.

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