On the reliable estimation of heat transfer coefficients for nanofluids in a microchannel

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Abstract. Nanofluids (base fluid and nanoparticles) can enhance the heat transfer coefficient \( h \) in comparison to the base fluid. This open the door for the design of efficient cooling system for microelectronics component for instance. Since theoretical Nusselt number correlations for microchannels are not available, the direct method using an energy balance has to be applied to determine \( h \). However, for low nanoparticle concentrations the absolute numbers are small and hard to measure. Therefore, the study examines the laminar convective heat transfer of \( \text{Al}_2\text{O}_3 \)-water nanofluids in a square microchannel with a cross section of \( 0.5 \times 0.5 \text{ mm}^2 \) and a length of 30 mm under constant wall temperature. The \( \text{Al}_2\text{O}_3 \) nanoparticles have a diameter size distribution of 30-60 nm. A sensitivity analysis with error propagation was done to reduce the error for a reliable heat transfer coefficient estimation. An enhancement of heat transfer coefficient with increasing nanoparticles volume concentration was confirmed. A maximum enhancement of 6.9 % and 21 % were realized for 0.6% \( \text{Al}_2\text{O}_3 \)-water and 1% \( \text{Al}_2\text{O}_3 \)-water nanofluids.

1. Introduction

Due to the miniaturization of microelectronic components, the installed power per area is increasing tremendously. Since this power has to be released to the surrounding, thermal management becomes the bottleneck on the rapid development of microelectronic components. The big amount of heat generated by microelectronic components might lead to a decrease in performance and reliability, or operational failure in the worst case. The current thermal management techniques such as pin fin heat sinks and air cooling will not be feasible solutions for future systems, since air is a poor heat transfer medium and heat sinks are limited in size. One of the promising solutions is to combine liquid cooling with microchannels because it offers a higher heat transfer ability especially for a small contact areas. Tuckerman and Pease [1] were the first who introduced the idea to use microchannels as microelectronics cooling system. They found the heat transfer coefficient of water to be in the range of 100 kW/m²K.

Efforts have been made in recent years to enhance the performance of liquid cooling systems. The addition of nano-sized particles (less than 100 nm in diameter) dispersed in commonly used cooling liquids (water, ethylene glycol, lubricant, etc) results in a change of the thermophysical properties of the suspension, the so called nanofluids. Since its introduction by Choi [2] nanofluids have gained the attention of many researchers. The early investigation has been focused on the nanofluids’ thermophysical properties, such as the dependency of the thermal conductivity on particle size and concentration. Various attempts have been made to gain a definitive physical reason on the nanofluids’ thermophysical properties enhancement.
Buongiorno [3] together with a group of researchers from over 30 different institutions worldwide compared the thermal conductivity of identical nanofluids samples through different measurement methods. They found that the thermal conductivity of water based nanofluids from most institutions deviated from the sample by an average of ±5%. Özerinc et al. [4] published the state of the art review on the enhanced thermal conductivity of nanofluids, they summarize the effect of particle size, particle shape, temperature and the clustering of nanoparticles that lead to discrepancies among researchers.

From the heat transfer application point of view nanofluids have drawn attention for a wide range of applications. Nanofluids have been used as working fluid for a direct absorption solar collector [5, 6]. Otanicar et al. [5] found 5% of efficiency improvement for graphite nanofluids. Leong et al. [7] dispersed 2% (volume concentration) of copper nanoparticles in ethylene glycol for car radiator and observed heat transfer enhancement of 3.8%. Nanofluids have also been employed as working fluid in electronic cooling systems [8, 9]. Nguyen [8] found 40% of heat transfer enhancement compared to the base fluid when 6.8% volume concentration of Al$_2$O$_3$ nanoparticles were dispersed in water. Jung et al. [10] investigated the laminar convective heat transfer of Al$_2$O$_3$-water nanofluids in rectangular microchannels for microelectronics cooling. They observed an enhancement up to 32% at a volume fraction of 1.8%.

Several established correlations were often used to predict the theoretical heat transfer coefficient and later compared to the experimental results. For some cases, these correlations were unable to predict the heat transfer coefficient correctly. Prabhat et al. [11] reviewed eight publications on laminar heat transfer of nanofluids and observed a significant deviation between the experimental result and the predictions from the Shah’s correlation. Heris et al. [12] also found the Sieder-Tate’s correlation was unable to predict the heat transfer coefficient of Al$_2$O$_3$-water nanofluids. Since these correlations were not developed for microchannels, their applicability remains question.

One reason for the strong deviations might be that only small differences for the absolute numbers of the heat transfer coefficients are observed for low nanoparticle concentrations. To ensure the heat transfer enhancement of nanofluids is not due to the measurement uncertainty, it is required to have a reliable value of heat transfer coefficient. In this present study, a sensitivity analysis with error propagation was performed to calculate the heat transfer coefficient of Al$_2$O$_3$-water nanofluids at two different volume concentration of 0.6% and 1%.

2. Experimental setup

The Al$_2$O$_3$-water nanofluids were purchased from Sigma Aldrich (642991) and have a particle size between 30-60 nm as reported by the vendor. The nanofluids were placed in an ultrasonic bath for one hour to maintain the stability of the nanofluids. The schematic diagram of the experimental setup and a photograph of the copper block including the microchannel are shown in Fig. 1a and Fig. 1b, respectively. The setup consisted of a test section, a pressure regulator, a fluid reservoir of 1 L, a waste fluid reservoir, a thermostatic bath and a precision scale. The test section was machined from a solid copper block of dimensions 70 × 70 × 50 mm$^3$. The microchannel was milled out on the bottom surface of the copper block with a cross section of 0.5 × 0.5 mm$^2$ and a length of 30 mm. The stability and uniformity of microchannel temperature was examined. For further details the reader is referred to our previous paper [13]. A total of eight PT-100 temperature sensors were connected to DEWE-50-PCI-16 data acquisition from Dewetron GmbH. The temperature sensors were mounted at different locations: one sensor was placed at the fluid reservoir to monitor the working fluid temperature, two sensors were inserted to the flow stream close to inlet and outlet of the microchannel, a set of four sensors were placed at different locations at the copper block to monitor the uniformity and stability of the wall temperature and another one was used to monitor the temperature around the microchannel. The temperature sensors were calibrated and the uncertainty of each sensor was
better than ± 0.06 °C. The temperature data was recorded every second for 30 minutes once the setup reached the steady state condition. A thermostatic bath D-7633 from Julabo which has a temperature resolution of 0.1 °C was used to fed the copper block and maintain a constant wall temperature. A pressure driven flow was utilized and regulated by a SPAB-P10R-G18-PB-L1 pressure regulator from FESTO. The AX623 precision scale with ± 0.1 mg accuracy from Sartorius was used to measured the mass of the discharged fluid. The working fluid was flowed through the microchannel with inlet temperature of 20 °C and the thermostatic bath temperature was set in accordance to the sensitivity analysis to 50 °C.

3. Thermophysical properties of nanofluids
The nanofluids’ thermophysical properties were calculated from well known correlations which are often used in a number of publications. The properties were calculated at average bulk temperature of the working fluid.

The density $\rho$ of the nanofluids was calculated by the correlation from Pak and Cho [14]:

$$\rho_{nf} = (1 - \alpha) \rho_f + \alpha \rho_{np}$$  \hspace{1cm} (1)

The specific heat $C_p$ of the nanofluids was calculated by the correlation from Xuan and Roetzel [15]:

$$\left(\rho C_p\right)_{nf} = (1 - \alpha) \left(\rho C_p\right)_f + \alpha \left(\rho C_p\right)_{np}$$  \hspace{1cm} (2)

Drew and Passman [16] introduced the correlation to calculate the viscosity $\mu$ of the nanofluids with volume concentration less than 5%:

$$\mu_{nf} = \mu_{bf} (1 + 2.5\alpha)$$  \hspace{1cm} (3)

Where the $nf$, $np$, $f$ and $\alpha$ are refer to nanofluids, nanoparticles, base fluid and volume concentration, respectively.

4. Data analysis
The $\rho_{nf}$, $C_{pnf}$ and $\mu_{nf}$ from equations 1, 2 and 3 were used to calculate the heat transfer coefficient and Reynolds number of nanofluids. The laminar heat transfer coefficient under a constant wall temperature was calculated as follows [17]:

$$q = mC_p \left[ (T_w - T_i) - (T_w - T_o) \right]$$  \hspace{1cm} (4)
\[ q = hA \left[ \frac{(T_w - T_1) - (T_w - T_o)}{\ln \frac{T_w - T_i}{(T_w - T_o)}} \right] \]

combining equation 4 and equation 5 :

\[ h = \frac{mC_p}{A} \ln \left( \frac{T_w - T_i}{T_w - T_o} \right) \]

where:

- \( h \): convective heat transfer coefficient in W/m²K
- \( m \): mass flow rate of the working fluid in kg/s
- \( C_p \): specific heat of the working fluid in kJ/kg K
- \( A \): contact area between microchannel wall and working fluid in m²
- \( T_i \): inlet temperature in °C
- \( T_o \): outlet temperature in °C
- \( T_w \): wall temperature in °C

5. Uncertainty analysis

To examine the uncertainty of the experimental results, an uncertainty analysis was done by using the error propagation method. Equation 7 can be used to calculate the uncertainty of a parameter that consist of several variables [18]. Where \( \Delta z \) represents the standard deviation of function \( f \), \( \Delta x_1 \) represents the standard deviation of variable \( x_1 \). The 95% confidence interval was taken into account for the uncertainty analysis.

\[ \Delta z = \pm \sqrt{\sum_{i=1}^{n} \left( \frac{\partial f}{\partial x_1} \Delta x_1 \right)^2} \]  

(7)

By applying the error propagation method in equation 7 to equation 6, the uncertainty of heat transfer coefficient can be calculated with the following equation.

\[ \Delta h = \pm \sqrt{\left( \frac{\partial h}{\partial m} \Delta m \right)^2 + \left( \frac{\partial h}{\partial C_p} \Delta C_p \right)^2 + \left( \frac{\partial h}{\partial T_w} \Delta T_w \right)^2 + \left( \frac{\partial h}{\partial T_i} \Delta T_i \right)^2 + \left( \frac{\partial h}{\partial T_o} \Delta T_o \right)^2} \]  

(8)

The uncertainty of every measured variable is listed in table 1. The uncertainty of specific heat \( \Delta C_p \) is relatively small therefore it can be neglected in the calculation.

| Variable            | Uncertainty | Uncertainty value |
|---------------------|-------------|-------------------|
| Mass                | \( \Delta m \) | ±0.1mg            |
| Wall temperature    | \( \Delta T_w \) | ±0.06°C           |
| Inlet temperature   | \( \Delta T_i \) | ±0.04°C           |
| Outlet temperature  | \( \Delta T_o \) | ±0.04°C           |

Table 1: The uncertainty of different measured variables
6. Results and discussions

The result of the sensitivity analysis is depicted in Fig 2. The analysis was done to check the effect of the change in wall temperature and Reynolds number to the uncertainty of the heat transfer coefficient which was calculated using equation 8. The wall temperature was set from 30 °C to 90 °C with an increment of 5 °C and the Reynolds number was between 76 until 260. A maximum of 13.7% uncertainty in heat transfer coefficient was found. The uncertainty was at its highest value when the wall temperature was set to 30 °C and Reynolds number of 76. This high uncertainty was due to a relatively small different between wall and outlet temperature. The sensitivity analysis shows that increasing the wall temperature and Reynolds number will minimized the uncertainty of the heat transfer coefficient, since the Reynolds number is fixed for a certain pressure difference for a microchannel, the wall temperature should be maximized for a reliable heat transfer measurement. Therefore, for the further experiments the wall temperature was set to 50 °C and the Reynolds number will start from at least 100.

Fig. 3 shows the heat transfer coefficients of water and Al₂O₃-water nanofluids as a function of Reynolds number. The experiments were done when the thermostatic bath temperature was set into 50 °C (except for the Al₂O₃-water 1%, where the wall temperature was 34 °C in a preliminary experiment,- therefore the uncertainty in Fig. 3 shows larger values, represented by larger error bars) and due to the heat loss, the recorded wall temperature was 48 °C. To validate the accuracy and reliability of the experimental setup, three identical experiments were done with water and it can be seen the water heat transfer coefficient of each experiment shows only small differences. It can be clearly seen, the heat transfer coefficient of nanofluids are higher than those of the base fluid including the uncertainty. An enhancement of 5.9 %, 6.9 % and 3.6 % were observed when the Reynolds number were 143, 215 and 268 for 0.6% Al₂O₃-water nanofluids. A higher enhancement was observed for 1% Al₂O₃-water nanofluids. An enhancement of 14 % and 21 % were found for the Reynolds number of 104 and 268, respectively. The experiment with Al₂O₃-water 1% nanofluids was done when the thermostatic bath was 35 °C and the recorded wall temperature was 34 °C. Therefore, a higher uncertainty (as expected from the
Figure 3: Heat transfer coefficient comparison of water and Al$_2$O$_3$-water nanofluids

sensitivity analysis) was experienced in this experiment. As can be seen from the experimental results, the enhancement of convective heat transfer coefficient is higher than the enhancement of nanofluids’ thermal conductivity. According to Maxwell’s correlation [19] an addition of 0.6% and 1% Al$_2$O$_3$ nanoparticles to water will only increase the thermal conductivity to 1.6% and 2.6%, respectively. For this reason, the thermal conductivity enhancement cannot be the only factor for the heat transfer enhancement.

7. Conclusion
A sensitivity analysis of the laminar convective heat transfer in a square microchannel under a constant wall temperature was performed. The analysis was done to ensure a reliable method to estimate the heat transfer coefficient by using an error propagation method. The effect of wall temperature and Reynolds number to the uncertainty of the heat transfer coefficient was quantified for the direct calculation of the heat transfer coefficient. The convective heat transfer with a constant wall temperature was evaluated for two different Al$_2$O$_3$-water nanofluids volume concentrations of 0.6% and 1%. A notable convective heat transfer coefficient enhancement of 21% was observed for Al$_2$O$_3$-water 1% nanofluids. It has also been found that the enhancement increased with increasing nanoparticle concentration within the solution.

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