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EVALUATION OF NANOFLUIDS PERFORMANCE FOR SIMULATED MICROPROCESSOR

by

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In this investigation, deionized water was used as base fluid. Two different types of nanoparticles, namely Al_2O_3 and Cu were used with 0.251% and 0.11% volumetric concentrations in the base fluid, respectively. Nanofluids cooling rate for flat heat sink used to cool a microprocessor was observed and compared with the cooling rate of pure water. An equivalent microprocessor heat generator i. e. a heated Cu cylinder was used for controlled experimentation. Two surface heaters, each of 130 W power, were responsible for heat generation. The experiment was performed at the flow rates of 0.45, 0.55, 0.65, 0.75, and 0.85 liter per minute. The main focus of this research was to minimize the base temperature and to increase the overall heat transfer coefficient. The lowest base temperature achieved was 79.45 ºC by Al_2O_3 nanofluid at Reynolds number of 751. Although, Al_2O_3-water nanofluid showed superior performance in overall heat transfer coefficient enhancement and thermal resistance reduction as compared to other tested fluids. However, with the increase of Reynolds number, Cu-water nanofluid showed better trends of thermal enhancement than Al_2O_3-water nanofluid, particularly at high Reynolds number ranges.

Key words: *nanofluid, specific heat, thermal capacity, enhancement*

Introduction

In computer industry, as processor's performance is being improved, cooling system's performance becomes a big technical challenge. On the other hand, size of computer is being reduced rapidly. So, there is a need to improve the cooling techniques to maintain the temperature in safe threshold range because conventional air cooling technique has reached its maximum limit in heat removing capacity. Nowadays, research in the field of heat transfer by liquid has attracted increasing interest due to higher thermal capacity of liquid. Research on this subject can be divided into two parts. The first one is modification of heat sink geometry with ordinary fluid and second one is modification of thermophysical properties of fluids with simple geometry in order to maintain proper functioning of electronic products.

About thirty years ago, Tuckerman and Pease [1] reported a study on the micro-channel heat sinks. They showed that by decreasing channel's hydraulic diameter, higher heat transfer

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could be achieved. This perspective gave new idea for research. Subsequently, the concept of heat transfer improvement by decreasing channel's hydraulic diameter up to micro level has been adopted for other application [2-4].

Xie *et al.* [5] performed a numerical study on mini-channel with water as a coolant and concluded that the heat transfer increased by decreasing channel width, bottom thickness of channel and by increasing channel depth. They found optimized configuration of heat sink, which was capable to remove heat up to 256 W/cm^2 by only 0.205 W pumping power, which was much higher than the maximum heat removed by air *i. e.* 100 W/cm^2.

Steinke and Kandlikar [6] studied different enhancement techniques that can be implemented on micro-channels. Those included different techniques like entrance effects, flow obstruction, secondary flows, curved path of flow, surface roughness increment, and addition of solid particles in coolants. The dynamic techniques included coolant or surface vibration, varying flow rate, exposed flow to electric field.

Nowadays, novel cooling techniques are required to improve the cooling efficiency of computer processor. Thus, researchers focused on cooling by special kind of fluid having high conductive *nano* particles. Water based nanofluids with various nanoparticles offer better cooling mechanism [7-9]. These fluids have better thermal properties. Convective heat transfer performance investigated experimentally by many researchers using different concentrations of Al_2O_3 nanofluids [10].

Sohel *et al.* [11] examined Al_2O-water nanofluid with four different volumetric concentrations (0.1%, 0.15%, 0.20%, and 0.25%) and compared those results with distilled water. They found that thermal effectiveness was increased up to a certain limit of flow rate, after that a decrease in thermal effectiveness was occurred. Higher concentration of nanofluids always showed better thermal effectiveness for all flow rates. They found 18% heat transfer coefficient enhancement compared to water by 0.25% vol. concentration.

Ho *et al.* [12] assessed forced convection heat transfer of Al_2O_3-water nanofluid with two volumetric concentrations (1% and 2%) and compared results with water. They observed 2% vol. nanofluid was less efficient than that of 1% vol. nanofluid due to less variation occurrence in dynamic viscosity with temperature. They found 70% enhancement in convective heat transfer coefficient compared to water using 1% vol. concentrated alumina nanaofluid.

Generally heat transfer rate increases with the increase of flow rate. However, this is not always true especially for the higher flow rates. Anoop *et al.* [13] used three concentrations of water based SiO_2 nanofluids to find heat transfer rate flowing through micro-channel fabricated by poly di-methyl siloxane in Reynolds number range of 4-22. All three weight concentrations (0.2%, 0.5%, 1%) of SiO_2-water nanofluids showed better heat transfer enhancement at lower Reynolds number as compared to heat transfer enhancement at higher Reynolds number with respect to water.

Rafati *et al.* [14] used ethylene glycol and deionized water based nanofluids having three different volumetric concentrations of alumina, silica, and titania. They performed test at three different flow rates of 0.5, 0.75, and 1.0 litre per minute and showed prominent reduction of base temperature by increasing flow rate. A 1.0% concentrated Al_2O_3 nanofluid showed better performance and reduced base temperature by 5.5 °C compared to base fluid.

Dixit and Ghosh [15] experimentation involved straight, diamond, and offset mini-channels. They found that Nusselt number varied linearly with Reynolds number, and remained invariant to heat flux. They also observed that thermal resistance was inversely proportional to fluid flow rate. A higher pressure drop was observed in diamond mini-channel as compared to offset mini-channel.

Peyghambarzadeh *et al.* [16] reported experimental investigation on water based CuO and Al_2O_3 nanofluids with 0.2% and 1% volumetric concentration, respectively. The CuO nanofluid showed better performance as compared to Al_2O_3 nanofluid. Heat transfer coefficient enhancement was up to 27% and 49% with CuO and Al_2O_3 nanofluid, respectively.

Corcione [17] gave empirical correlation for the dynamic viscosity and thermal conductivity of nanofluids based on available experimental data. He found that the ratio of nanofluid and base fluid thermal conductivity increased by decreasing size and by increasing volumetric concentration. Further, the ratio of nanofluid and base fluid dynamic viscosity increased by decreasing size, and by increasing volumetric concentration of nanofluid.

Jajja *et al.* [18] performed experiment on different integral fin heat sinks by varying spacing with water as a coolant at power of 325 W. They concluded that thermal resistance and base temperature decreased with the increase of flow rate and decrease of fin spacing. They found maximum enhancement ratio of 1.39 in overall heat transfer coefficient against 3.9 in area enhancement ratio.

Recent research shows that the thermal conductivity of nanofluids is inversely proportional to grain size. Keblinski *et al.* [19] explained this by considering particles Brownian motion, liquid layering at liquid/particle interface, heat transport nature, and nanoparticles clustering effects. They found that the Brownian motion role was not important as compared to other studied factors in heat transport properties. They also showed that decrease in thermal conductivity occurred with increase of particle diameter.

Naphon and Nakharintr [20] compared cooling rate achieved by TiO_2-water nanofluid with cooling rate achieved by deionized water using three different rectangular heat sinks by varying height. They found that heat transfer rate increased and pressure drop decreased with increase of fin height. They observed 11%, 27%, and 42.3% enhancement in average heat transfer rate compared to water with 1.0 mm, 1.5 mm, and 2.0 mm height heat sinks, respectively.

Shenoy *et al.* [21] performed experiments on multi-walled carbon nanotubes grown as integral part of silicon mini-channel with water as a coolant. Two different multi-walled carbon nanotubes mini-channel were tested, one mini-channel had 6×12 (rows, columns) and other was fully covered with multi-walled carbon nanotubes. Experiment showed that 6×12 bundle device and fully covered multi-walled nanotubes were capable of removing base heat flux 2.3 times and 1.6 times, respectively, while keeping same base temperature.

Hung *et al.* [22] computationally investigated hydraulic and thermal performance of 3-D porous micro-channel having rectangular, trapezoidal, outlet enlargement, thin rectangular, sandwich, and block distributions for Reynolds number range of 45-1350. They found, porous configuration of heat sink exhibited better thermal performance and became prominent at high Reynolds number, while pressure drop also increased with addition of porous material.

Yang *et. al.* [23] analyzed air cooling rate on restricted geometries heat sinks. They compared the result of plate, slit and louver fin heat sinks. They found that louver fin heat sink had better heat transfer rate than plate and slit fin heat sinks. However, it also showed more pressure drop. Experimental result showed that best thermal design of louver heat sink was at 1.65 mm fin spacing, which reduced 25% requisite heat dissipation area.

Literature study shows that most of the researcher focused on heat sink geometry in order to maintain temperature in a safe threshold range. The aim of the present study is to minimize the manufacturing cost and complexity involved in manufacturing of mini-channel/micro-channel heat sinks. In this investigation a simple flat heat sink is used: Al_2O_3 and Cu based nanofluids with 0.251% and 0.11% volumetric concentration were tested at different Reynolds number.

Experimental set-up

The pictorial and schematic view of test rig is shown in figs. 1(a) and 1(b), respectively. The rig was consisted of fluid reservoir, two brushless DC pumps (DC30A-1230, China), radiator (R121225BH, Gigabyte, Taiwan), needle valve, micro-flow meter (FTB333D, Omega, USA), heat sink, Cu block, two surface heater, K-type thermocouples (5TC-TT-KI-30-1M, Omega, USA), data acquisition system (34972A, Agilent, USA), and DC power supply (8102, Lodestar, USA).

One litter of fluid from liquid reservoir was pumped by two DC brushless pumps, connected in parallel to each other, to maintain the constant flow rate. Each pump consumed 4 W with a maximum flow rate of 4 LPM. Pumps were able to manage pressure drop produced by experimental loop.
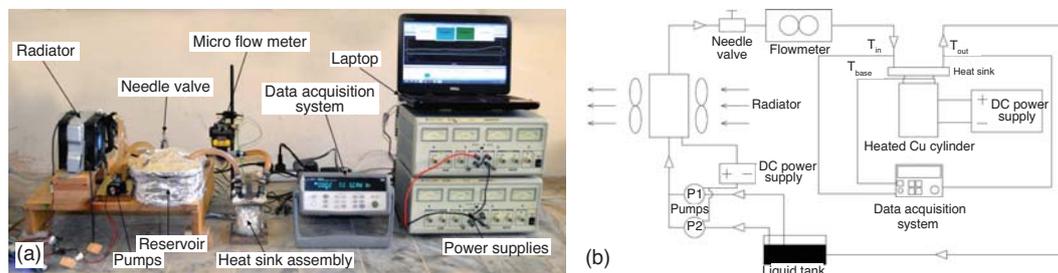
Figure 1. (a) Experimental set-up and (b) schematic diagram of experimental set-up

The energy absorbed by fluid from heat sink was dissipated by radiator, which maintained the fluid temperature of 38 °C at the inlet of heat sink. The radiator of a commercial CPU liquid cooling system (GALAXY) was used. Next to the radiator, needle valve and micro-flow meter was installed to control and measure the required flow rate, respectively. The full scale accuracy of flow meter was ±7%.

The heat sink was manufactured by CNC machine. The specification of heat sink is shown in tab. 1, see fig. 2(a). At the center of the base, there was a hole of 1 mm with depth of 2.5 mm, which was 1 mm below the upper surface of heat sink as show in fig. 2(b).
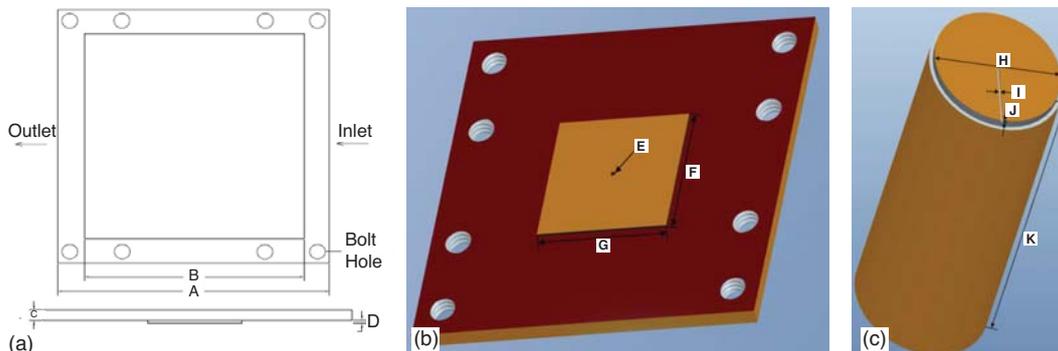
Figure 2. (a) Drawing of heat sink, (b) base of heat sink, and (c) the Cu cylinder

One thermocouple was inserted in the hole to measure the base temperature of heat sink. The protruded base of heat sink was mounted on the heating block.

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The heating block was mainly consisted of Cu circular cylinder. A slot of 1 × 1 mm along the radius was made on the top of the cylinder to insert a thermocouple to the base of heat sink as shown in fig. 2(c).

Two surface heaters, each of 386 Ω resistance, were mounted parallel on the Cu cylinder. Two DC power supplies connected in series were responsible to generate 260 W through the heaters. The voltage and current was set to 224 V and 1.16 A, respectively. To minimize thermal resistance between Cu cylinder and heat sink, high heat conductive silver thermal paste was used. Before application of thermal paste, surface of both of heat sink base and Cu cylinder upper surface was finished by 1500 micron fine sand paper. Four nuts and bolts arrangement was used for fit assembling of heat sink with Cu cylinder as shown in fig. 3.

To measure the inlet, outlet, base, and ambient temperature, Agilent data acquisition system was used. The employed nanofluid, Al_2O_3 and Cu were prepared at National Center of Physics, Pakistan. These fluids were water based and stable for one month at ambient temperature. The concentration of each nanofluid was 1 wt.%.

Table 1. Specifications of heat sink geometry

Geometry	Dimensions [mm]
Heat sink	Total length (A) = 76 Characteristic length (B) = 60 Thickness (C) = 3 Extruded part thickness (D) = 0.5 Hole diameter (E) = 1 Extruded part area (F×G) = 28.7 × 28.7
Cu cylinder	Diameter (H) = 28 Slot width (I) = 1 Slot height (J) = 1 Cylinder height (K) = 80
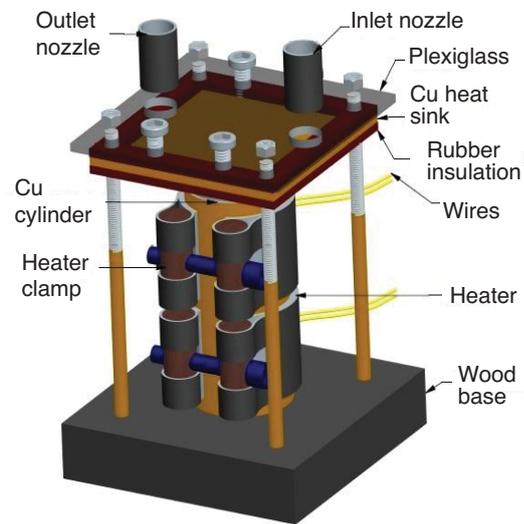
Figure 3. Heat sink assembly

Data reduction

When fluid passes from heat sink, it extracts heat. The heat transfer rate between the heat sink and the liquid is calculated:

$$Q = \dot{m}c_p\left(T_{out} - T_{in}\right) \tag{1}$$

Density, specific heat, and viscosity are calculated at mean temperature of fluid given:

$$T_m = \frac{T_{in} + T_{out}}{2} \tag{2}$$

The heat transfer rate in term of overall heat transfer coefficient is calculated:

$$Q = UA_r\left(LMTD\right) \tag{3}$$

where

$$A_r = B^2 \tag{4}$$

The LMTD is log mean temperature difference. This can be calculated:

$$LMTD = \frac{(T_b - T_{in}) - (T_b - T_{out})}{\ln \dfrac{(T_b - T_{in})}{(T_b - T_{out})}} \tag{5}$$

By comparing eqs. (1) and (3), we get eq. (6) for overall heat transfer coefficient:

$$U = \frac{\dot{m} c_p (T_{out} - T_{in})}{A_r (LMTD)} \tag{6}$$

Thermal resistance is another important evaluating parameter of heat sink performance, defined:

$$R = \frac{LMTD}{Q} \tag{7}$$

Following eq. (8) is used to calculate the volumetric percentage, ϕ, of nanofluids from the given wt.% [24]:

$$\phi = \frac{w_{np} \rho_{bf}}{\rho_{np} (1 - w_{np}) + w_{np} \rho_{bf}} \tag{8}$$

Density of nanofluids depends upon density of base fluid, which is dependent of mean fluid temperature. Following eq. (9) is used to calculate the density of nanofluids [7]:

$$\rho_{nf} = \phi \rho_{np} + (1 - \phi) \rho_{bf} \tag{9}$$

Specific heat capacity of nanofluids is also dependent on base fluid density and specific heat capacity. Following eq. (10) is used to calculate the specific heat capacity of nanofluids [7]:

$$c_{nf} = \frac{\phi \rho_{np} c_{np} + (1 - \phi) \rho_{bf} c_{bf}}{\rho_{nf}} \tag{10}$$

To calculate the viscosity of nanofluids, a well-known Batchelor [25] relation is used:

$$\mu_{nf} = \mu_{bf} (1 + 2.5\phi + 6.2\phi^2) \tag{11}$$

Reynolds number is calculated:

$$Re = \frac{\rho v d_h}{\mu} \tag{12}$$

Whereas hydraulic diameter can be calculated as:

$$d_h = 4 \frac{A_c}{P} \tag{13}$$

Uncertainty analysis

To incorporate the effect of experimental uncertainty of flow rate, inlet and outlet temperature of coolant and heat sink base temperature on the final calculated parameters of interests, Kline and McClintock [26] method was used. The maximum uncertainties in heat transfer rate, overall heat transfer coefficient and Reynolds number were never found greater than 7.90%, 7.92%, and 0.9%, respectively, in any case.

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Results and discussion

Comparison of fluid temperature difference with Reynolds number

The test is performed at low Reynolds number range of 400-800 due to the flow available by two DC brushless pumps. Moreover, for specific heat input, inlet and outlet temperature difference gradually decreases by increasing Reynolds number as shown in fig. 4.

Comparison of base temperature with Reynolds number

The heat sink's base temperature and the processor's working temperature are analogous to each other. The systematic effects of Al_2O_3-water and Cu-water nanofluids is studied on base temperature of heat sink and compared with water at five different Reynolds number. The base temperature decreases with increase of Reynolds number for all tested fluids as shown in fig. 5. Moreover, high concentrations of nanoparticles in fluid may lead to high heat transfer rate by that fluid. Although, Al_2O_3 nanoparticles have lower thermal conductivity than Cu nanoparticles, but due to high volumetric concentration of Al_2O_3 nanoparticles, Al_2O_3-water nanofluid shows lower base temperature than that of Cu-water nanofluid at same Reynolds number. However, at high Reynolds number, Cu-water nanofluid shows more base temperature reduction and becomes nearly equal to Al_2O_3 nanofluid base temperature. Water shows less reduction in base temperature with respect to both Al_2O_3-water and Cu-water nanofluids. As an overall, Al_2O_3-water and Cu-water nanofluids show 5.39% and 3.89% less base temperature compared to water, respectively.
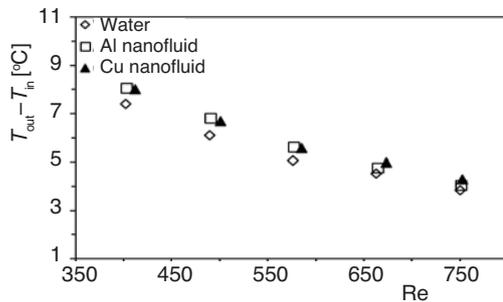
Figure 4. Comparison of fluid temperature difference with Reynolds number
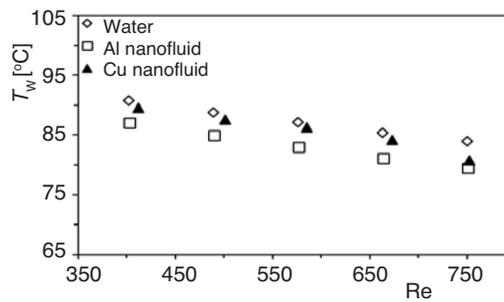
Figure 5. Comparison of base temperature with Reynolds number

Comparison of heat transfer rate with Reynolds number

Heat transfer rate as a function of Reynolds number is shown in fig. 6. The trends are different for different kind of fluids. For Al_2O_3-water nanofluid, at low Reynolds number, heat flow rate increases with increase of Reynolds number first. After that reduction in heat rate occurs for further increase of Reynolds number. The Al_2O_3-water nanofluid shows maximum enhancement 8.34% compared to water at Reynolds number of 577.
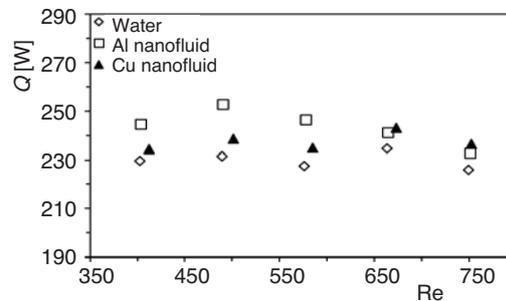
Figure 6. Comparison of heat transfer rate with Reynolds number

Now consider Cu-water nanofluid and water case, which are following nearly same pattern. Heat transfer rate is not following specific trends for both cases and makes a zigzag graph. At lower Reynolds number range, Al_2O_3-water nanofluid shows better performance as compared to both Cu-water nanofluid and water. However, at higher Reynolds number, Cu-water nanofluid shows better performance even more than Al_2O_3-water nanofluid.

A maximum enhancement of 4.66% with respect to water is found at Reynolds number of 752. Cu-water nanofluid containing lower concentration of nanoparticle is more effective than Al_2O_3-water nanofluid at higher Reynolds number.

Comparison of overall heat transfer coefficient with Reynolds number

The overall heat transfer coefficient is the best criteria to evaluate heat transfer characteristic because overall heat transfer coefficient includes both base temperature and heat transfer rate. Figure 7 shows the comparison of Reynolds number and overall heat transfer coefficient for all tested fluids. The overall heat transfer coefficient increases with increase of Reynolds number for all tested fluids. For Al_2O_3-water nanofluid, overall heat transfer coefficient increases gradually with the increase of Reynolds number. Overall heat transfer coefficient increases with a high rate at lower Reynolds number as compared to its enhancement at high Reynolds number. The Al_2O_3 nanofluid shows a maximum overall heat transfer coefficient value of 1678 W/m^2 °C. For Cu-water nanofluid, at lower Reynolds number, its performance with respect to Al_2O_3-water nanofluid is less efficient, but with the increase of Reynolds number, its performance improves and becomes almost equal to Al_2O_3 nanofluid at 721 Reynolds number. The Cu nanofluid shows a maximum overall heat transfer coefficient 1644.46 W/m^2 °C. Overall, Al_2O_3-water and Cu-water nanofluids show 12.56% and 9.80% enhancement compared to water, respectively.

Comparison of thermal resistance with Reynolds number

Figure 8 shows the thermal resistance variation with respect to Reynolds number for all tested fluids. All fluids show nearly similar trend against Reynolds number and thermal resistance decreases with increase of Reynolds number. The Al_2O_3-water nanofluid shows the lowest thermal resistance as compared to other two fluids. The Al_2O_3 nanofluid shows a minimum thermal resistance of 0.17 K/W at Reynolds number of 751. For Cu-water nanofluid, thermal resistance gradually decreases with the increase of Reynolds number. The Cu nanofluid shows minimum thermal resistance of 0.17 K/W at Reynolds number 752.
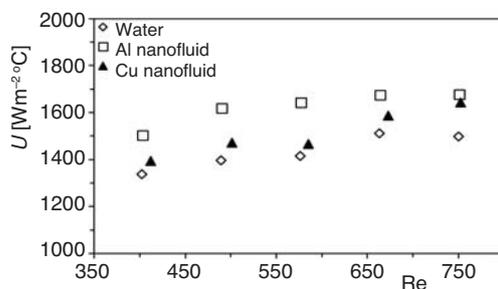
Figure 7. Comparison of overall heat transfer coefficient with Reynolds number
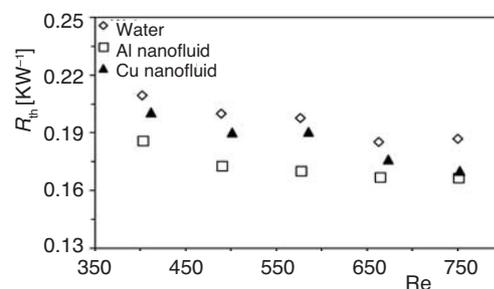
Figure 8. Comparison of thermal resistance with Reynolds number

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Conclusions

The Al_2O_3 and Cu nanofluids with 0.251% and 0.11% volumetric concentrations with water as a base fluid were tested using a flat plate heat sink. Following are the important findings obtained from the experiment:

- Both nanofluids showed higher heat transfer performance in comparison with pure water.
- The Al_2O_3-water nanofluid showed greater heat transfer rate than Cu-water nanofluid at low Reynolds number, but Cu behaved more effectively at high Reynolds number.
- Heat transfer rate was not necessarily increased or decreased with the increase of Reynolds number.
- The lowest base temperature achieved was 79.45 °C by Al_2O_3 nanofluid at Reynolds number of 751.
- The Al_2O_3-water and Cu-water nanofluids showed 8.34% and 4.66% enhancement in heat transfer rate compared to water, respectively.
- The Al_2O_3-water and Cu-water nanofluids exhibited 12.56% and 9.80% augmentation in overall heat transfer coefficient compared to water, respectively.

Nomenclature

A_r – heat transfer area of the
    flat plate heat sink, [m^2]
A_c – cross sectional area, [m^2]
c_p – specific heat, [kJkg^{-1}°C^{-1}]
d_h – hydraulic diameter, [m]
$\dot{m}$ – mass flow rate of fluid circulating through
    the heat sink, [kgs^{-1}]
P – perimeter, [m]
Q – heat removed by the fluid circulating
    through the heat sink, [W]
R – thermal resistance of heat sink [KW^{-1}]
Re – Reynolds number ($= \rho v d h / \mu$)
T – temperature, [°C]
U – overall heat transfer coefficient, [Wm^{2}K^{-1}]
w – weight fraction

Greek symbols

μ_{bf} – viscosity of base fluid, [kgm^{-1}s^{-1}]
ρ – density, [kgm^{-3}]
Φ – volumetric fraction

Subscripts

bf – base fluid
in – inlet
m – mean
nf – nanofluid
np – nanoparticle
out – outlet

Abbreviation

LMTD – log mean temperature difference
LPM – liter per minute

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