Investigations on heat and momentum transfer in CuO-water nanofluid

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Abstract This paper presents results of investigations on the application of CuO-water nanofluids for intensification of convective heat transfer. Performance of nanofluids with 2.2 and 4.0 vol.% CuO NPs (nanoparticles) content were examined with regard to heat transfer coefficient and pressure losses in case of turbulent flow in a tube. Negligible impact of examined nanofluid on heat transfer improvement was found. Moreover, measured pressure losses significantly exceeded those determined for primary base liquid. The observations showed that application of nanofluid for heat transfer intensification with a relatively high solid load in the examined flow range is rather controversial.

Keywords: Nanofluids; Fluid mechanics; Heat transfer

Nomenclature

c –   heat capacity, J/(kgK)
d –   tube internal diameter, m
f –   Darcy's friction factor
F –   heat transfer surface, m^2

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G –   mass flow rate, kg/s
h –   heat transfer coefficient, W/(m^{2}K)
l –   length of the tube, m
s –   thickness of the tube wall (0.001 m)
u –   linear velocity, m/s
U –   overall heat transfer coefficient, W/(m^{2}K)
Q –   heat transfer rate, W
T –   temperature, K

Greek symbols

ϕ –   volume fraction
ρ –   bulk density, kg/m^3
λ –   heat conductivity coefficient, W/(mK)
ΔP –   pressure loss, Pa

Subscripts

Cu –   copper
CuO –   copper oxide
m –   logarithmic mean
s –   refers to a shell section of heat exchanger
w –   pure water
nf –   nanofluid
1 –   refers to the outer diameter of annuli (0.02 m)
2 –   refers to the inner diameter of annuli (0.008 m)

1   Introduction

A concept of nanofluid was coined by Stephen Choi and refers to the suspension of nanoparticles (NPs) in the base liquid, e.g., water, ethylene glycol, oil, etc. [1]. A development of nanotechnology made possible the preparation of highly stable suspensions of solids characterized by small size, typically below 100 nm and relatively high heat conductivity coefficient [2]. This makes nanofluids desirable media for intensification of heat transfer. As the solid phase metals, nonmetals or their oxides are used [3,21]. Due to a very high unit surface area the former may undergo fast oxidation, so application of oxides seems to be more convenient, safe and economical in industrial applications. Much of research on preparation, characterization and thermal performance of nanofluids can be found in the open literature [4,5]. Most reported data refer to convective heat transfer in laminar or turbulent flow of Al_2O_3 [6], TiO_2 [7] or CNT (carbon nanotubes) [8]. There is a relatively small number of papers dealing with the issue of thermal performance of CuO-based nanofluids. Heat transfer enhancement was observed in CuO-water nanofluid stabilized with carboxymethyl cellulose addition

(CMC) [9]. Results of measurements of the apparent viscosity of examined nanofluid suggested high pressure losses in the flow. Unfortunately data on related pressure drop were not presented. Promising results with regard to heat transfer enhancement were also obtained when the nanofluid CuO-water + ethylene glycol was examined, [10]. No information on applied stabilizer was provided. Due to high viscosity and density the examined nanofluid demonstrated again high pressure losses what may be a limiting factor for wider application in practice.

The objective of this study was to investigate CuO-water nanofluid stabilized by citrate triammonium (CTA) with regard to heat transfer enhancement effect, and pressure losses in turbulent flow regime. Measurements of pressure losses allowed to hypothesize about the possibility of reducing the flow resistance by the addition of CuO NPs.

2 Experimental

2.1 Preparation and tests of nanofluids

For experimental purposes CuO-water nanofluid with 2.2 and 4.0 vol.% load of solid was prepared by a two-step method. A prescribed amount of CuO 30–50 nm NPs was mixed with 0.15 wt.% water solution of CTA and then stirred vigorously with a high-shear stress homogenizer Micra D9 for 1 h at a rotating speed of 15 000 1/min. Then the suspension was processed with an ultrasonic horn Sonics VCX 750 for 5 h at 60% of maximal amplitude (114 μm). Applied CTA adsorbed onto CuO NPs surface impacted double electrostatic layer, reduced aggregation creating static repulsive effect, and lowered pH of CuO-water system to the optimal range 5–6 where zeta potential exceeded 30 mV. Finally, colloid with good stability was fabricated. Thus obtained nanofluid was stable for at least one day without sedimentation. Heat conductivity coefficient of thermostated sample was measured using commercial instrument Decagon KD2 equipped with 6 cm probe KS-1. This instrument employed THW (transient hot wire) method and provided an accuracy of $\pm5\%$. The dynamic coefficient of viscosity was determined by the Brookfield LV II Pro viscometer at mean measurement temperature. Density of the nanofluids was determined with the aid of a pycnometer method.

In the literature two approaches with regard to effective heat capacity calculation can be found. In the approach presented in [11] the heat capacity

is obtained from the relation

$$c_{nf} = \frac{\phi \rho_{CuO} c_{CuO} + (1 - \phi) \rho_w c_w}{\rho_{nf}} \qquad (1a)$$

whereas from [12] it reads:

$$c_{nf} = \phi\, c_{CuO} + (1 - \phi)\, c_w, \qquad (1b)$$

where c_{CuO} – 535.6 J/kgK [13], ρ_{CuO} – 6 300 kg/m^3.

Equation (1b) is recognized as giving overestimated results [14], thus in this study Eq. (1a) was used for calculation of the effective heat capacity as more accurate. Properties of examined nanofluids and water at average of inlet and outlet temperature were gathered in Tab. 1.

Table 1: Properties of examined nanofluids and water.

CuO load [vol.%]	Heat capacity [J/kg K]	Density [kg/m^3]	Dynamic viscosity $\times 10^3$ [Pa s]	Thermal conductivity [W/m K]
0.0	4190	995	0.863	0.610
2.2	3856	1074	1.65	0.620
4.0	3415	1214	2.19	0.682

3   Experimental set-up

Figure 1 shows the experimental loop used for determination of overall heat transfer coefficient and pressure drop. Nanofluid from the container (1) was delivered by the pump (2) through a cooling system (3,4) to the shell-and -tube heat exchanger (6). The shell of the exchanger was heated by water from a thermostat (5) at known constant flow rate, G_s. Inlet and outlet temperatures were measured. Then, the nanofluid was supplied to the container through the second cooling system (7). The inlet and outlet temperatures were measured by means of four K-type thermocouples, calibrated with an accuracy of ± 0.1 K, connected to an A/D Advantech converter. The flow rate of the nanofluid, G_{nf}, was determined by measurement of the time required to fill a 1 dm^3 vessel. Pressure loss in 6 mm ID (inner diameter) tube was measured using a pressure transducer Peltron NPDX over the distance

of 1.080 m with an accuracy of $\pm 0.25\%$. Readings were performed after 45 min time, which was necessary to approach a steady state condition for heat transfer. Experiments were conducted in the Reynolds number range from 4 000 to 12 000.

4 Data reduction

Global heat transfer coefficient for examined nanofluids was determined the on basis of fundamental heat transfer equation

$$U = \frac{Q}{F \Delta T_m}, \tag{1}$$

where Q was calculated as arithmetic mean of Q_s and Q_{nf} according to equations

$$Q_s = G_s \overline{c}_w (T_4 - T_3), \tag{2}$$

$$Q_{nf} = G_{nf} \overline{c}_{nf} (T_2 - T_1). \tag{3}$$

For known U value, heat transfer coefficient of nanofluid, h_{nf}, was calculated as follows

$$h_{nf} = \frac{1}{\frac{1}{U}\frac{F_s}{F_m} - \frac{s}{\lambda_{Cu}}\frac{F_s}{F_m} - \frac{1}{h_s}\frac{F_s}{F_{nf}}}, \tag{4}$$

where $\lambda_{cu} = 400$ W/(mK).

Heat transfer coefficient in the shell section h_s, was calculated according to [15]

$$\text{Nu} = \frac{h_s (d_1 - d_2)}{\lambda_w} = 0.020 \, \text{Re}^{0.8} \text{Pr}^{0.33} \left(\frac{d_1}{d_2}\right)^{0.53}. \tag{5}$$

5 Results

Firstly, the accuracy of the method for determining of h_{nf} was examined. Figure 2 shows the comparison of experimental data for water with data calculated according to Gnielinski's equation [16]

$$\text{Nu} = \frac{h_w d}{\lambda_w} = \frac{\left(\frac{f}{8}\right)\text{RePr}}{1 + 12.7\left(\frac{f}{8}\right)^{0.5}\left(\text{Pr}^{2/3} - 1\right)}\left[1 + \left(\frac{d}{l}\right)^{2/3}\right], \tag{6}$$
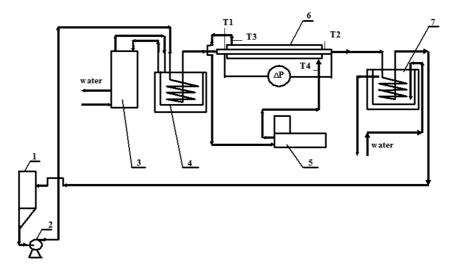
Figure 1: Experimental set-up. 1 – nanofluid container, 2 – pump, 3 – chiller, 4 – secondary cooler, 5 – thermostat, 6 – shell-tube heat exchanger, 7 – primary cooler, T_1, T_2 T_3, T_4 – K-type thermocouples, ΔP – pressure transducer.

where Darcy's friction factor, f, for turbulent flow in the smooth tube was originally calculated as [16]

$$f = \frac{1}{(1.8 \log \mathrm{Re} - 1.5)^2}. \tag{7}$$

Experimentally determined values of the Nusselt number are slightly higher than theoretical, and maximal discrepancy does not exceed 13%.

In Fig. 3 a comparison of the Nusselt number value for water and examined nanofluids vs. Reynolds number is shown. The resulting Nusselt number is almost the same as determined for water or slightly lower. Expected heat enhancement in this case is rather controversial but in agreement with findings of other works [17]. This is clearly visible in case of turbulent flow regime where heat transfer coefficient is a complex function of nanofluid properties as heat conductivity, heat capacity, viscosity, and density. Presence of NPs impacts the value of the last as shown in Tab. 1, and a resultant trend of changes may lead, in general, to moderate or even minute heat transfer improvement though thermal conductivity of nanofluids is significantly larger than for the base liquid.

In the paper, the pressure losses in the flow through a straight, circular tube were also investigated. In order to validate accuracy of pressure
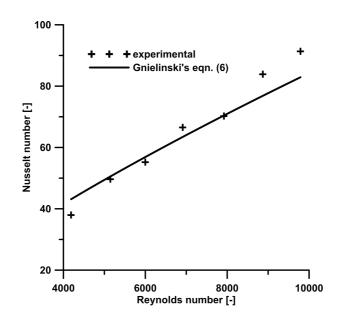
Figure 2: Comparison of experimental and calculated Nusselt number.

losses measurements experimental data for pure water were compared with calculations obtained from the relation:

$$\Delta P = f \frac{l}{d} \frac{u^2}{2} \rho, \qquad (8)$$

where f was calculated using Eq. (7).

Comparison presented in Fig. 4 indicates good agreement between experimental and calculated data in case of water for the examined Reynolds number range. The maximal error was lower than 5%. For the nanofluids under consideration an increase of pressure loss can be observed with accompanied increase of Re number and NPs content. All experimental data for nanofluids significantly exceed those measured for water. This can be attributed to the impact of NPs on the increase of density and viscosity of investigated colloids. Approximation of the experimental data by curve of $\Delta P = A\mathrm{Re}^B$ type gives values of exponential factor B equal to 1.73, 1.70, 1.65 for water, nanofluid 2.2, and 4.0 vol.%, respectively. These values are close to a value of 1.75 originated from the Blasius formula for friction factor. It suggests the presence of turbulent flow regime in the investigated system.

Experimental pressure losses for nanofluids were also compared with

data calculated using Eqs. (7) and (8) and properties of fluids from Tab. 1. As shown in Fig. 4 experimental data for nanofluids are lower than those calculated at the same Reynolds number. The difference increases almost proportionally with the increase of CuO NPs contents, and is by 18% and 35% lower (in terms of measured pressure loss) for nanofluis 2.2 and 4.0 vol.%, respectively.
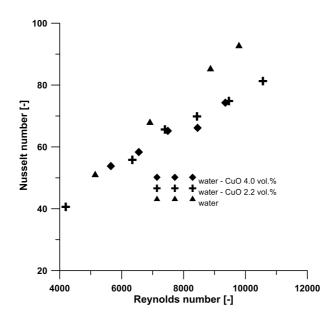

Figure 3: Nusselt number vs. Reynolds number for nanofluids and water.

In the literature there is a well described influence of solid particles on drag force for suspensions owing to Toms' effect, [18]. However, limited data on the influence of nanomaterials on drag reduction can be found. For nanofluid water-carbon nanotubes (CNT) lowering of the friction factor, f, below the value determined for the base fluid was observed for certain flow rate range [19]. This effect was attributed to extension of laminar flow regime due to turbulence suppression by CNT. An impact of CuO NPs on pressure drop was also observed for nanofluid at low NPs content [20]. For the Reynolds number range from 9 000 to 30 000 a moderate decline of the ratio of pressure loss in flow of nanofluid to pressure drop of base fluid, accompanied with increase of NPs content, was observed. It should also be noted that in case of CuO-water nanofluid with CuO load of 0.2 vol.% the reduction of apparent viscosity with regard to host liquid (water+CMC)
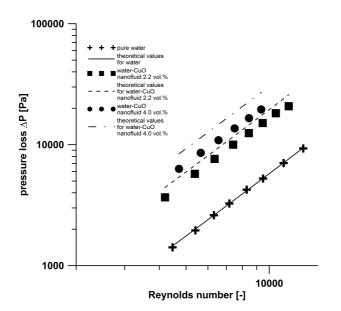
Figure 4: Pressure loss in flow of investigated nanofluids and water. Theoretical values calculated using Egs. (7) and (8).

viscosity was observed, [9].

At this moment there is no clear evidence that aforesaid differences in observed and predicted pressure losses can be attributed to the CuO NPs presence, and detailed explanation of this effect needs further research.

6   Conclusions

The present work dealt with investigations on the application of CuO-water nanofluids for intensification of convective heat transfer. Performance of nanofluids with 2.2 and 4.0 vol.% CuO contents was examined with regard to heat transfer and pressure losses for turbulent flow in the straight tube. The main results can be summarized as follows:

- Addition of NPs to the base liquid affects significantly physical properties, i.e., increases heat conductivity, viscosity and density of the resultant nanofluid, and decreases heat capacity.

- For the range of Reynolds number, $0.4 \times 10^4 \leq \text{Re} \leq 1.2 \times 10^4$, there was found a negligible impact of NPs presence on heat transfer improvement. The experimentally determined Nusselt's number for

nanofluids was the same or slightly lower than that determined for host liquid.

- Pronounced pressure losses in flow of 2.2 and 4.0 vol.% CuO-water nanofluids in comparison to base liquid were observed, that increased with NPs contents.

- In case of CuO-water nanofluids reduction of pressure loss below theoretical prediction is observed, that can be attributed to a NPs presence but explanation of this effect needs further investigations.

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