

Experimental investigation of forced convection of water/EG- Al_2O_3 nanofluids inside horizontal tube

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Abstract. In this paper, forced convection of water/EG- Al_2O_3 nanofluids inside horizontal stainless steel tube is studied experimentally. As base fluid served water/EG mixture of two ratios (60:50 and 50:50). Nanoparticle mass concentrations was 0.1% or and 1%. Transition and turbulent flow regimes were tested. Average heat transfer coefficient and pressure drop values were determined for nanofluids and compared to base fluids. Deterioration of heat transfer for all tested nanofluids has been recorded compared to base fluid. Negligible increase of pressure drop for nanofluids compared to base fluids has been observed.

1 Introduction

Simultaneously with research aimed at enhancing heat transfer by modifying the construction of a thermal apparatus (change of surface roughness, surface development [1], use of flow turbulence elements) or using additional elements in the installation causing the liquid to pulsate or vibrate, numerous works are carried out to change thermophysical properties of conventional working fluids or proposing new products that can replace traditional liquids. In last two decades strong interest in fluids with suspended nanosize solid particles (called nanofluids) results in large number of experimental and numerical studies. So far carried out experiments dealing with forced convection of nanofluids in both laminar and turbulent conditions and conducted under the same thermal-hydraulic conditions, i.e., for the same Reynolds number and temperature of the process showed that the degree of intensification of heat transfer is higher than can be expected from an increase in thermal conductivity [2]. Some of the researchers shown that adding Al_2O_3 or TiO_2 nanoparticles to distilled water have considerable impact on thermal conductivity of base fluid [3]. Azmi et al. [4] show that for given operating temperatures heat transfer coefficients can be almost similar for nanofluids with different thermal conductivity. Changes in fluid internal shear stress and weakening of the inter-particle and inter-molecular adhesion forces with decreasing of the nanofluid viscosity can be the main reason of this behavior. Higher thermal conductivity of nanofluids compared to the base fluid can be caused by several mechanisms proposed by Khurana et al. [5]. These are: Brownian motion (random motion of the nanoparticles), interfacial layer (liquid layer close to nanoparticle), aggregation (formation of a linear assembly of nanoparticle chain), thermophoresis (nanoparticle motion induced by thermal gradients) and

intensification of turbulence (small solid particles in nanofluid develop turbulent eddies). All of this facts lead to the conclusion that the thermal conductivity is not the only reason for the increase of heat transfer coefficient in forced convection conditions. Liquid properties, such as viscosity, density and specific heat, also have impact on heat transfer coefficient during forced convection. From the other hand, geometry of the system, i.e. the diameter and the length of a tube, also change conditions of the heat transfer process.

Many groups of investigators conducted experiments to determine heat transfer coefficients for different types of nanofluids. Noghrehbadi and Pourrajab [6] studied heat transfer inside circular tubes under uniform constant heat flux boundary condition. Three volume concentrations, i.e. 0.1%, 0.3% and 0.9% of water/ γ - Al_2O_3 were tested under laminar flow regime. Comparison with distilled water shows that average heat transfer coefficient increases of about 16.8% at 0.9% volume concentration and $\text{Re}=2070$. Enhancement of heat exchange was particularly significant in the entrance region of the test section and decreases with axial distance. Sahin et al. [7] collate experimental data for water/ CuO nanofluid tested inside horizontal aluminium pipe. They concluded that the particle volume concentrations higher than 1% vol. were not appropriate with respect to the heat transfer performance of the water/ CuO nanofluid. The highest heat transfer enhancement was achieved for 0.5% volume concentration and $\text{Re}=16.000$. At higher than 1% particle volume fraction, the Nusselt number decreases with increasing particles loading. Another, widely tested type of nanofluid is liquid based on water/EG mixture, as an often used working fluid for car engine cooling, solar collector systems etc. Azmi et al. [8] investigated forced convection of Al_2O_3 nanofluids in 60:40, 50:50, 40:60 water and ethylene glycol mixture ratio. Reynolds number was varied between 3000 to 25000. Maximum

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heat transfer enhancement was noticed at a working temperature of 70°C for 60:40 water/EG base fluid and 1% volume concentration of nanoparticles, and reached 24,6%. Authors also mentioned that for all mixtures friction factor increases slightly as particle concentration increased. Kulkarni et al. [9] showed that not only particle loading, but also size of these particles have substantial influence on heat transfer process. They tested nanofluids consisted of SiO₂ nanoparticles and water/ethylene mixture (40:60 % by weight). Nanoparticle of 20 nm, 50 nm and 100 nm diameters were used. Published results shown that the heat transfer increases with the particle size increase. According to Kulkarni et al. larger diameter particles results in lower viscosity of nanofluids, what cause that the Reynolds number increases. What is more, for larger particles also particle Peclet number increases. Combination of these two effects is a reason for the increasing of the heat transfer coefficient with increasing of the particle size.

Also more complicated geometries of tubes and inserts were tested with the different types of nanofluids. Mohhamadiun et al. [10] combined in their study three heat transfer enhancing methods, i.e. corrugation, inserts and nanofluids. They investigated fluid flow of the EG/Al₂O₃ nanofluid under turbulent flow conditions. Nanofluids of 0.5, 1 and 1.5% volume concentrations of Al₂O₃ were prepared and tested with three twist ratios of twisted tape. Maximum intensification of heat transfer was found with the use of 0.5% particle loading and lowest twisted tape ratio and was defined as maximum thermal performance factor equal to 4,2. Some of researchers go further and conducted experiments to identify application possibilities of nanofluids. Cieśliński et al. [11] used water-Al₂O₃ nanofluids instead of traditional working fluid in flat solar collector. The tests were conducted for three mass concentration of nanoparticles: 0.1%, 0.2% and 0.4%, for two values of radiation: 1050 W/m² and 400 W/m², and volume flow rates respectively 60 l/h, 120 l/h and 170 l/h. The efficiency was calculated according to the recommendations of the Polish Standard. The results obtained show that the application of nanofluids instead of pure water improves efficiency of the tested flat plate solar collector for given range of thermal radiation. Independent of the volume flow rate of the nanofluid, the maximum efficiency was recorded for nanoparticle mass concentration equal to 0.2%. Kaya et al. [12] also carried out the experiment for solar collector. In this case, evacuated U-tube solar collector was used. Water/EG base fluid in 50:50 proportion was used to prepare nanofluids with volume concentrations of ZnO nanoparticles of 1, 2, 3, 4%. Maximum growth of solar collector efficiency was noticed for 3,0% nanoparticle loading and working fluid mass flow rate of 0,045 kg/s and it was calculated as almost 27% increase comparing to pure base fluid. Sidik et al. [13] presented comprehensive review on the application of nanofluids for cooling system of the traditional car engines. From rich database they come to the conclusion that heat transfer coefficient in coolant radiator can be improved up to 50% compared to the base coolant. From the other hand, problem is that with adding nanoparticles pressure

drop is also increasing, so the optimum performance of cooling system can be reached for lower concentration of nanoparticles (<1% by vol).

This paper shows an experimental study on heat transfer potential of water-ethylene glycol/Al₂O₃ nanofluids during forced convection inside horizontal stainless steel tube.

2 Experimental setup

Scheme of the experimental test loop used in this experiment is shown in Fig. 1.

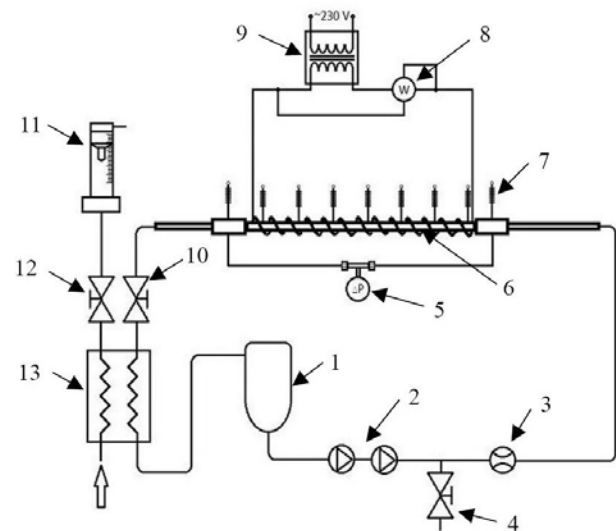


Fig. 1. Scheme of the test rig: 1 – reservoir tank, 2 – rotary pumps, 3 – flow meter, 4 – drain valve, 5 – differential pressure transducer, 6 – test tube, 7 – temperature sensors, 8 – wattmeter, 9 – autotransformer, 10 – valve, 11 – rotameter, 12 – cooling water control valve, 13 – plate heat exchanger.

Nanofluid circulates in the test loop by use of the group of rotary pumps (2). Reservoir tank (1) is made from glass in order to observe the potential settlement of the nanoparticles during the test. To measure the mass flow rate of the nanofluid magnetic flow meter (3) is used. Control valve (10) and different pump operation sequences are used to change mass flow rate. Test tube (6) with 8 mm internal diameter, 1 mm wall thickness and 2000 mm length is made from stainless steel. Tube is connected to the autotransformer (9) and works as a direct heater. Heat flux is estimated due to the measurement of electrical power by wattmeter (8) assuming no heat loss to the environment. The hydrodynamic entry section is long enough to accomplish fully developed region at the entrance of the heat transfer test section. Plastic (PTFE) fittings are attached at both ends of the test section to thermally isolate the connection. Two types of temperature sensors (7) are used: two thermocouples of J-type to measure the nanofluid temperature at the inlet and the outlet of the test section and six Pt100 sensors (mounted to the surface of the tube) are used to measure the axial wall temperature distribution of the test tube. The inside wall temperature is calculated assuming radial heat conduction within the tube wall. Cooling section is responsible for keeping temperature of tested fluid on

required value. Tap water flowing through the plate heat exchanger (11) cools hot nanofluid. Flow rate of cooling water is controlled by valve (12) and measured by rotameter (13).

3 Preparation and characterization of the tested nanofluids

In the present study Al_2O_3 nanoparticles were used while mixture of distilled, deionized water and ethylene glycol was applied as a base fluid. Two volume proportions of this mixture (W:EG) were used, i.e. 60:40 and 50:50. Nanofluids with 0,1 and 1% nanoparticle mass concentrations were tested during the experiment. Alumina (Al_2O_3) nanoparticles have mean diameter estimated to be lower than 50 nm according to the manufacturer (Sigma-Aldrich Co.). The tested nanofluids were prepared by use of the two-step method. Nanoparticles of the required amount and base liquid were mixed together in ultrasonic bath. There wasn't used any dispersant to stabilize the suspension.

4 Data reduction and uncertainty estimation

The Nusselt number is calculated as:

$$Nu = \bar{\alpha}d_i/\lambda_{nf} \quad (1)$$

Thermal conductivity of the tested nanofluids λ_{nf} is calculated from the Maxwell model [14]:

$$\lambda_{nf} = \lambda_f + \frac{3(\lambda_n - \lambda_f)\Phi}{\left(\frac{\lambda_n}{\lambda_f} + 2\right) - \left(\frac{\lambda_n}{\lambda_f} - 1\right)\Phi} \quad (2)$$

where λ_n is taken from the producer of nanoparticles (Sigma Aldrich Ltd.) and λ_f is measured by KD2 (Decagon Devices Inc.) thermal properties analyzer. Average heat transfer coefficient is calculated as:

$$\bar{\alpha} = \dot{Q}/A_i(\bar{T}_w - \bar{T}_f) = P/A_i(\bar{T}_w - \bar{T}_f) \quad (3)$$

where heat transfer area is calculated as:

$$A_i = \pi d_i L \quad (4)$$

and average nanofluid temperature \bar{T}_f reads:

$$\bar{T}_f = (T_{in} + T_{out})/2 \quad (5)$$

Average inside wall temperature of the tube \bar{T}_w is calculated as [15]:

$$\bar{T}_w = \bar{T}_o - \frac{P}{2\pi\lambda_i L} \ln(d_o/d_i) \quad (6)$$

where \bar{T}_o is calculated as an arithmetic mean of measured outside wall temperatures in six positions along the tube - Fig.1:

$$\bar{T}_o = \sum_{j=1}^{j=6} T_{oj}/6 \quad (7)$$

Reynolds number is calculated as:

$$Re = wd_i/v_{nf} \quad (8)$$

where fluid velocity w is calculated as:

$$w = \dot{V}/A_{cs} \quad (9)$$

Kinematic viscosity of the tested nanofluids is calculated as:

$$v_{nf} = \mu_{nf}/\rho_{nf} \quad (10)$$

Dynamic viscosity of the tested nanofluids is calculated from Brinkman formula [16]:

$$\mu_{nf} = \frac{\mu_f}{(1-\Phi)^{2,5}} \quad (11)$$

where μ_f is measured by Rheotest LK 2.2 (Medingen GmbH) viscometer.

Density of the nanofluids is calculated as [17]:

$$\rho_{nf} = (1-\Phi) \cdot \rho_f + \Phi \cdot \rho_n \quad (12)$$

The uncertainties of the measured and calculated parameters are estimated by mean-square method. Because heat flux is calculated from the formula

$$q = P / \pi d_i L \quad (13)$$

the experimental uncertainty of the heat flux is estimated as follows:

$$\Delta q = \sqrt{\left(\frac{\partial q}{\partial P} \cdot \Delta P\right)^2 + \left(\frac{\partial q}{\partial d_i} \cdot \Delta d_i\right)^2 + \left(\frac{\partial q}{\partial L} \cdot \Delta L\right)^2} \quad (14)$$

where the absolute measurement errors of the electrical power ΔP , inside tube diameter Δd_i and active length of the tube ΔL are 10 W, 0.02 mm, and 0.2 mm, respectively. So, the maximum overall experimental limits of error for heat flux extended from $\pm 1.4\%$ for maximum heat flux up to $\pm 3\%$ for minimum heat flux.

Because average heat transfer coefficient is calculated from the formula

$$\bar{\alpha} = q/(\bar{T}_w - \bar{T}_f) = q/\Delta T \quad (15)$$

the experimental uncertainty for the average heat transfer coefficient is calculated as:

$$\Delta \bar{\alpha} = \sqrt{\left(\frac{\partial \bar{\alpha}}{\partial q} \Delta q\right)^2 + \left(\frac{\partial \bar{\alpha}}{\partial \Delta T} \delta \Delta T\right)^2} \quad (16)$$

where the absolute measurement error of the temperature difference, $\delta\Delta T$, estimated from the systematic error analysis equals ± 1 K. The maximum error for average heat transfer coefficient was estimated to $\pm 7.0\%$.

5 Results

Figure 2 shows average Nusselt number versus Reynolds number for tested base fluids and prepared nanofluids for selected heat flux equal to 20 kW/m^2 . It is seen that addition of nanoparticles results in Nusselt number decrease for given Reynolds number. Nusselt number (heat transfer coefficient) decreases with the nanoparticle loading increase with maximum decrease of about 25% for water:EG ratio equal to 50:50 and 1% of Al_2O_3 nanoparticle concentration. Pressure drop was presented in Fig. 3. Negligible pressure drop for all tested nanofluids has been recorded.

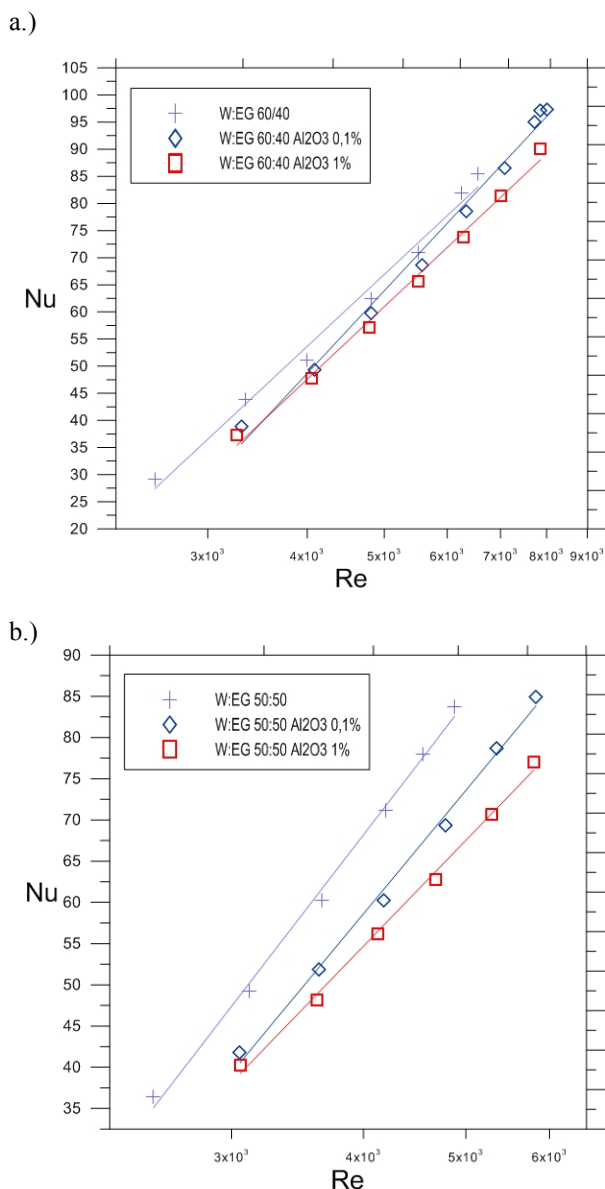


Fig. 2. Average Nusselt number for water/EG- Al_2O_3 nanofluids: a) water:EG (60:40), b) water:EG (50:50).

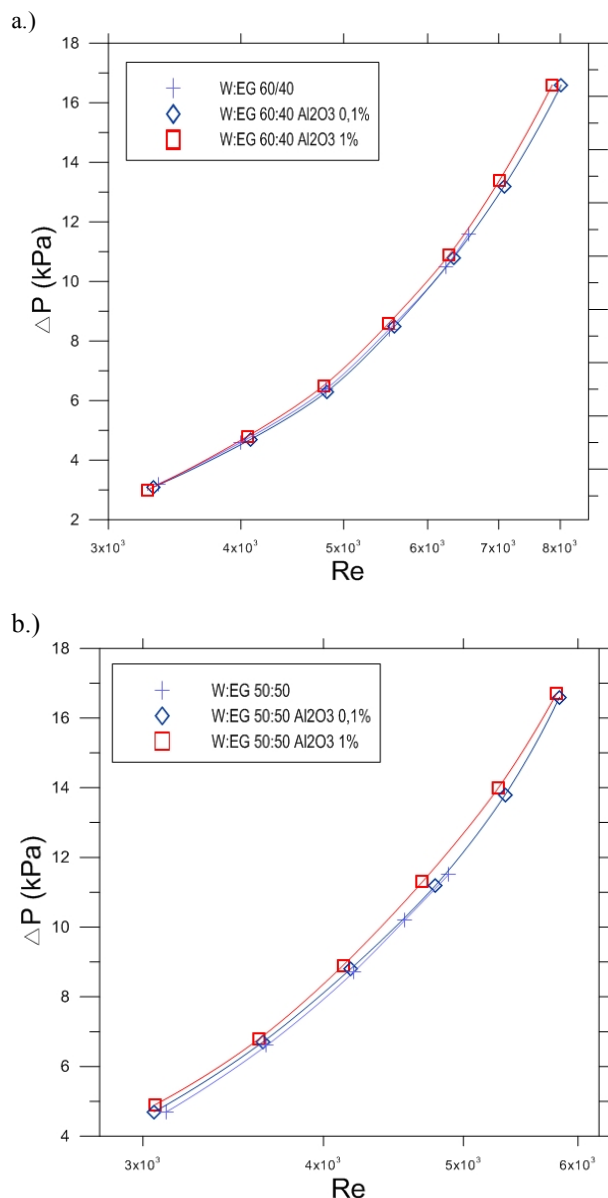


Fig. 3. Pressure drop for water/EG- Al_2O_3 nanofluids: a) water:EG (60:40), b) water:EG (50:50).

6 Conclusions

Obtained results revealed:

- degradation of heat transfer for water/EG- Al_2O_3 nanofluids with 0,1% and 1% nanoparticle mass concentrations within tested range of Reynolds number compared to base fluids,
- negligible increase of pressure drop for all tested nanofluids compared to base fluids.

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Nomenclature

A - area, m²,
 A_i - heat transfer area, m²,
 d - diameter, m,
ED - ethylene glycol,
 L - active tube length, m,
 P - electrical power, W,
 \dot{Q} - heat flow rate, W,
 T - temperature, °C,
 \bar{T} - average temperature, °C,
 \dot{V} - volume flow rate, m³/s,
 w - fluid velocity, m/s,

Greek symbols

$\bar{\alpha}$ - average heat transfer coefficient, W/(m²K),
 λ - thermal conductivity, W/(m·K),
 μ - dynamic viscosity mPa·s
 ν - kinematic viscosity, m²/s,
 ρ - density, kg/m³,
 Φ - nanoparticle volumetric fraction,

Subscripts

cs - cross section
 f - fluid
 i - inside
 in - inlet
 n - nanoparticle
 nf - nanofluid
 o - outside
 out - outlet
 t - tube
 w - inside wall