Investigation of forced convection ZrO_2 nanofluid in a channel with artificial roughness

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Abstract. The paper presents the results of experimental study of turbulent forced convection of water-based nanofluid with ZrO_2 nanoparticles. Volumetric concentration of the nanoparticles in the experiments was equal to 4%. The average particle size was 105nm. The study was carried out for a smooth round tube and round tube with knurling ring. The effect of additives of ZrO_2 nanoparticles on the average heat transfer coefficient and the pressure drop in the channel was investigated. Has been shown that the intensification of heat transfer by using nanofluid weakly dependent on the shape of the channel, and determined only by the properties nanofluid.

1. Introduction

The enhancement of convective heat transfer and the related experimental and theoretical research become at present an independent, important and rapidly developing field of heat transfer theory. High heat flux removal is a major consideration in the design of many machines, equipment and technologies, and can be accomplished using various kinds of heat transfer equipment. The urgency of this problem is determined by driving to enhance the performance of heat transfer devices, reduce energy costs and achieve maximum compactness with minimum material consumption. One of the solutions to the problem of heat transfer performance enhancement could be the use of so called nanofluids, which are fluids containing nanoparticles of various composition. The first experiments have shown that even very small additions of nanoparticles to the fluid (a fraction of volume concentration) may lead to increase in thermal conductivity and heat transfer of the nanofluids by tens of percent, whereas the critical heat flux may be increased several times [1-5]. Over the last two decades, a great number of works have been emerged in this area. Most of them were focused on the study of thermal conductivity and viscosity of nanofluids. Actually, the study of heat transfer was performed in a relatively few research works, and the results presented are extremely controversial [5]. Most works reveal an increase in heat transfer when using nanoparticles. The increase in heat transfer coefficient in nanofluids as compared to base fluids can range from a few percent to 350% for carbon nanotubes. However, there are publications showing the deterioration of heat transfer when adding nanoparticles. Most of these works are dealt with the study of convection of nanofluids in laminar flow regime [6-9]. Turbulent convection is explored to a substantially lesser degree.

The first work, in which the turbulent heat transfer of nanofluids was studied, is apparently the work of Pak and Cho [10]. They experimentally studied the turbulent heat transfer in water-based nanofluids, containing Al_2O_3 and TiO_2 nanoparticles, and flowing in a horizontal circular pipe with a

constant wall heat flux. The results have shown that the Nusselt number in nanofluids increases with increasing both bulk concentration of the particles and the Reynolds number. However, in the same paper [10] it has been revealed also that at high concentrations of nanoparticles the heat transfer coefficient may be lower than that in pure water (by 12% in a nanofluid with particles concentration of 3%).

Convective heat transfer in turbulent flow regime of nanofluids containing copper nanoparticles was experimentally investigated in the subsequent work [11]. The experimental results have shown that adding nanoparticles to the base fluid significantly improved the heat transfer efficiency of base fluid (by 60% in a nanofluid with particles concentration of 2%), while the friction factor remained almost the same as for water.

In [12] the efficiency of heat transfer in water-based nanofluids containing titanium oxide nanoparticles was investigated in laminar and turbulent flow regimes in vertical tube under constant wall heat flux boundary condition. The results have shown that the heat transfer coefficient definitely increases with increase of nanoparticles concentration in both laminar and turbulent regimes at a fixed Reynolds number. The maximum intensification of heat transfer coefficient, recorded in the experiment, was 40% for 1.1% nanofluid. In this case, the pressure drop in nanofluids, when flowing in the channel, was very close to that in pure fluid. In addition, the authors investigated the effect of particles size, though no effect of size on the heat transfer coefficient was detected, possibly, because the particles were large enough (95-210 nm).

Duangthongsuk and Wongwises [13] experimentally studied turbulent heat transfer and pressure drop in water-based nanofluids with TiO_2 particles. And again the results were extremely controversial. They revealed a 32% enhancement in thermal performance at particles concentration of 1%, and 14% reduction in the heat transfer coefficient at particles concentration of 2% as compared with the pure fluid.

Fotukian and Nasr Esfahany [14] studied turbulent convective heat transfer in nanofluids with a very low concentration of Al_2O_3 nanoparticles in water flowing in a circular pipe. They recorded a clear increase in heat transfer coefficient and pressure drop with increasing concentration of particles. The maximum increase of heat transfer coefficient equal to 48% was reached at negligibly small volume concentration of the nanoparticles (0.054%). In a subsequent paper [15] the same authors studied turbulent heat transfer of water-based nanofluids containing CuO nanoparticles in a circular pipe. They achieved a 25% increase in the heat transfer coefficient and a 20% increase in pressure drop at a concentration of nanoparticles equal to 0.24%, as compared to pure water. Besides, they revealed that the heat transfer coefficient was almost independent of the concentration of nanoparticles within the observed range of volume percent change from 0.039 to 0.24.

Thus, the analysis of the literature shows that in the available works there is neither quantitative nor qualitative consensus in terms of turbulent heat transfer performance of nanofluids. In the meantime, almost all scientists note that the effect of nanoparticles on the heat transfer in the turbulent flow regime is much more complicated than that in laminar flow regime. This circumstance requires additional systematic experimental study of turbulent heat transfer of nanofluids.

The purpose of this paper - the study of the influence of artificial roughness on the turbulent forced convection and pressure drop for nanofluid.

2. Description of the experimental setup

The diagram of the installation to study the heat transfer coefficient is shown in figure 1. The installation is a closed loop with a circulating coolant [16]. The working fluid is pumped through the heated test section to the heat exchanger, where heat is removed by the thermostat. The flow rate of the working fluid in the loop is controlled by controlling the pump power by means of laboratory transformer. Power input to the pump is measured through Omix meter.
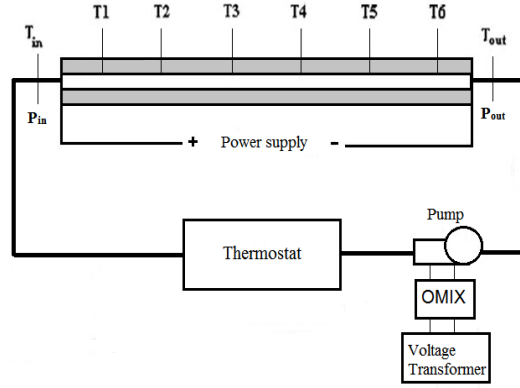
Figure 1. A schematic diagram of the measuring system.

The heated test section is a stainless steel tube 6 mm in diameter, 1m long with wall thickness of 0.5 mm. The tube was heated by supplying electric current directly to its wall. This heating technique allows obtaining a constant wall heat flux boundary condition. In addition, this heating technique is universal and easily applicable to tubes of any cross section. The tube is insulated by multi-layer insulation. The heating power is controlled by a transformer. Six chromel-copel thermocouples fixed on the tube wall at the equal distance from each other were used to measure the local temperature of the tube. Temperature measurements were carried out by TPM-200 meters. In addition, the temperatures at the inlet and outlet of the heated test section were measured using thermocouples. At that, the thermocouple designed to measure the fluid temperature at the outlet of the loop was located at a considerable distance from the end of the heated test section to ensure uniformity of the fluid temperature in the metering point. The loop section between the heater and the fluid temperature metering point was also insulated. The pressure drop measurements were carried out using a differential manometer OWEN PD200. The designed experimental setup was tested based on known empirical data for heat transfer of pure water. The water flow rate ranged from 0.65 to 2 l/min that corresponds to the range of Reynolds number from 2300 to 8000. Figure 2 shows the comparison of the experimental data on the average Nusselt number depending on the Reynolds number with the known empirical correlation [17]:

$$Nu = 0.021 \text{Re}^{0.8} \text{Pr}^{0.43}.$$

Experimental Nusselt number – $Nu = \dfrac{\alpha d}{\lambda}$, where $\alpha = GC_P(T_{out} - T_{in})S^{-1}(T_w - \bar{T})^{-1}$ – is the average heat-transfer coefficient; G – mass flow rate measured by flowmeter, C_p – is the specific heat of the fluid; S – is the area of the lateral surface of the channel; d – is the diameter; T_{out}, T_{in} – are fluid temperatures at the channel inlet and outlet; $\bar{T} = \dfrac{(T_{in} + T_{out})}{2}$ – is the average fluid temperature; T_w – is the arithmetic mean of the channel wall temperature, obtained by averaging of the values of six thermocouples; Pr – is the Prandtl number; $\text{Re} = \dfrac{4G}{\pi d \mu}$ – is the experimental Reynolds number, λ and μ – are fluid thermal conductivity and viscosity coefficients, respectively. The viscosity and thermal conductivity coefficients of this nanofluid also been measured and are equal to $\mu = 0.001047$ Pa×s and $\lambda = 0.657$ W/(m×K).

The viscosity of the nanofluid was measured using a rotational viscometer DV2T. Viscosity measurements were carried out within the range of shear velocity from 10 to 200 1/s at a temperature

of 25°C. Before the measurements, the viscometer was calibrated with distilled water and ethylene glycol. The results obtained were in a good agreement with the reference data.

The coefficient of thermal conductivity of nanofluid was measured using a non-stationary hot wire method. The detailed description of the installation and its testing is given in [18].

As is obvious from the plots presented in figure 2, experimental data obtained in the turbulent flow regime, are in good agreement with the empirical correlation [17]. The discrepancies do not exceed 5% that is comparable to the accuracy of the correlation.
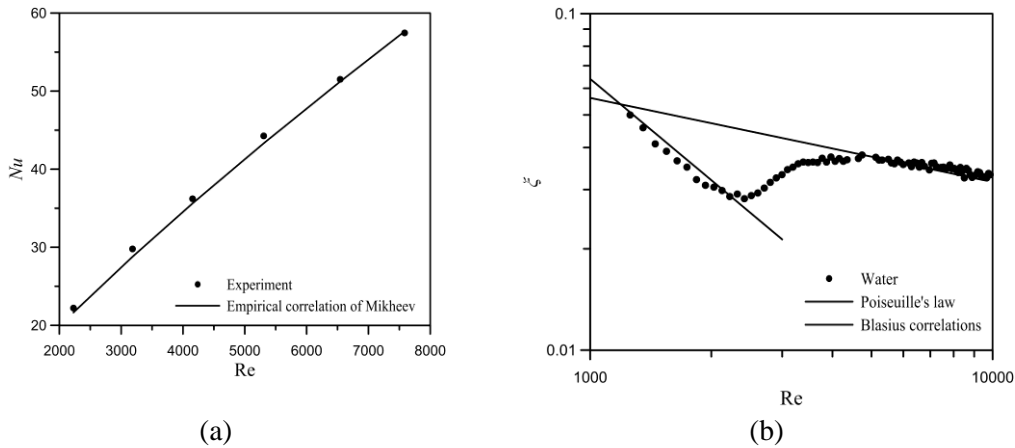

(a) (b)

Figure 2. Heat transfer coefficient (a) and friction factor (b) for pure water versus Reynolds number.

In addition to the heat transfer of pure water, pressure drop was measured as well. Figure 2b shows the measured correlation between friction factor and Reynolds number for pure water. The friction factor was calculated by known formula:

$$\xi = \frac{2d\Delta P}{\rho U^2 L}$$

where U – is the superficial velocity; d – is the diameter; L – is the test section length; and ΔP – is the measured pressure drop.

For comparison, the chart shows also calculations by Poiseuille's theoretical relationship $\xi = \dfrac{64}{\text{Re}}$ for laminar flow and Blasius correlation $\xi = 0.316\text{Re}^{-0.25}$ for turbulent flow. It is obvious that within the range of Reynolds numbers from 2300 to 3000 we observe flow transition from laminar to turbulent regime. The measured pressure drop values are consistent with the theoretical values with an accuracy of 5%.

3. The results of the experiments

The turbulent forced convection of water-based nanofluid with zirconia nanoparticles with the volume concentration of 4% was studied. This study was carried out for a round smooth tube and round tube with knurling ring. The experimental results are shown in figures 3-4. On the smooth round tube the ZrO_2 nanofluid allows to intensify the heat transfer coefficient by 27% compared to pure water at a fixed Reynolds number. The experiment was shown that the using of nanofluid can improve the heat transfer coefficient in the tube with knurling by 23% compared with pure water (figure 3). The difference in the intensification of heat transfer for smooth tube and tube with knurling due to nanofluid does not exceed the experimental error 5%.
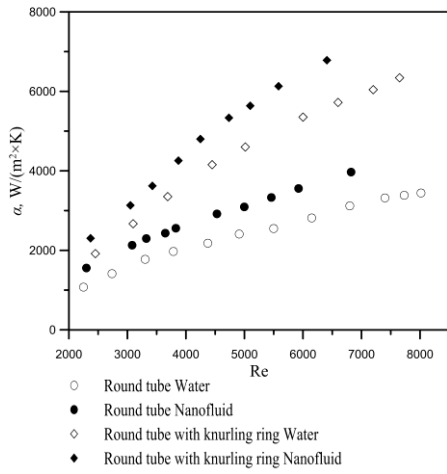
Figure 3. Average heat transfer coefficient versus Reynolds number.
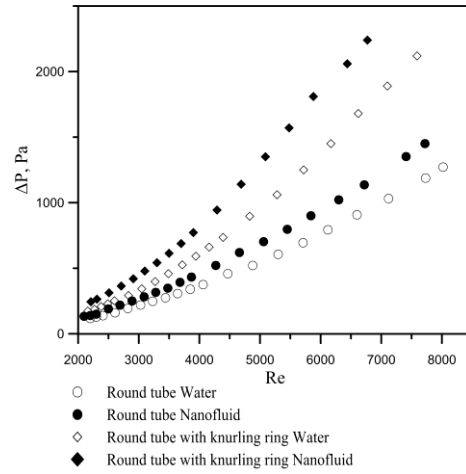

Figure 4. Pressure drop versus Reynolds number.

Figure 4 shows the pressure drop for both channels depending on Reynolds number for water and ZrO_2 nanofluid. Data analysis has shown that the nanofluid the increases pressure drop in the smooth and in the rough channels on the same value about 25%.

It is necessary to estimate of the thermal-hydraulic efficiency for the different channels. The heat transfer coefficient in the tube with a knurling ring to pure water is higher by 85%, than the heat transfer coefficient of smooth tube. However the pressure drop in the rough channel is also increased by 80% compared with a smooth tube. Nanofluid allows intensifying heat transfer in a smooth tube by 27% and increases the pressure drop is only 25% compared to pure water. To evaluate the thermal-hydraulic efficiency we used the ratio of heat transfer coefficients to the ratio pressure drops $\eta = \left(\dfrac{\alpha}{\alpha_{w,s}} \right) \left(\dfrac{dp}{dp_{w,s}} \right)^{-1}$. Here $\alpha_{w,s}$, $dp_{w,s}$ - average heat transfer coefficient and pressure drop in a smooth tube for pure water.

This dependence for different Reynolds numbers is shown in figure 5. The graph shows that the using of nanofluid in a round smooth tube has approximately the same efficiency, taking into account the measurement error. The applying of nanofluid in rough pipe is not effectively. The thermal-hydraulic for this case less than 1.
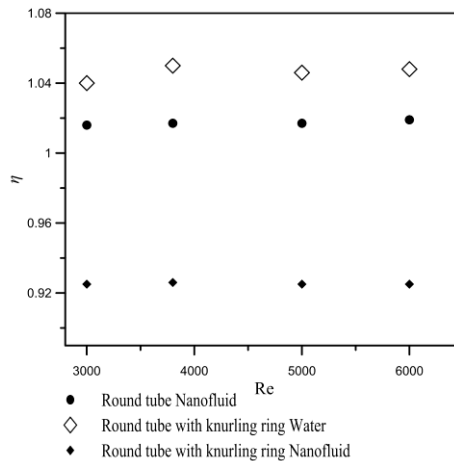

Figure 5. Thermal hydraulic efficiency versus Reynolds number

4. Conclusion

The turbulent forced convection of water-based nanofluid with zirconia nanoparticles was studied. The heat transfer coefficient and the pressure drop for pure water and nanofluid in the smooth and rough pipes been measured. The experiment showed that the heat transfer coefficient with using ZrO_2 nanofluid increases by about 25% in comparison with the pure water for both the smooth and rough channels. It was obtained that the nanofluid increases pressure drop in the smooth and in the rough channels on the same value about 25%. Thus, it has been shown that the intensification of heat transfer by using nanofluid weakly dependent on the shape of the channel, and determined only by the properties nanofluid.

A comparison of the thermal-hydraulic efficiency η of using nanofluid and using tubes with knurling ring has been done. It is shown that the using nanofluid in the smooth pipe has comparable performance with using tubes with knurling ring for pure water. The using of nanofluid in rough pipe is not effectively.

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