THE EXPERIMENTAL STUDY OF THE EFFECT OF ADDING HIGH-MOLECULAR POLYMERS ON HEAT TRANSFER CHARACTERISTICS OF NANOFLUIDS

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Abstract. The experimental study of influence of adding high-molecular weight polymers on heat transfer characteristics of water-based nanofluids with Al2O3 nanoparticles. Volumetric concentration of particles was very low, 10-3 %. The volumetric particle concentration was equal to 2%. We investigated the dependence of heat transfer coefficient and pressure drop on the concentration of additives high polymers.

1 Introduction

Intensification of heat transfer and the related problems of experimental and theoretical studies are currently assuming an important and rapidly developing field of the teaching of heat transfer. In all machines, equipment and technologies there is a need of intensive heat removal, which uses a different kind of heat transfer equipment. One way to solve the problem of heat transfer enhancement of the process is the use of liquids with a dash of nanoparticles of various compositions, called "nanofluids".

The term "nanofluids" was first proposed in [1] to paper to the two-phase system consisting of a base fluid and nanoparticles high thermal conductivity material. Typical base fluids are water, organic liquids (ethylene glycol, oil), polymeric solutions. The material for the nanoparticles are metals, metal oxides, carbon nanotubes.

The aim of this study was to: Experimental study ways to reduce the friction in the turbulent heat transfer nanofluids using polymeric additives for the creation of nanofluids with desired transport properties and rheology.

2 Description of the experimental setup

The diagram of the installation to study the heat transfer coefficient is shown in fig 1. The installation is a closed loop with a circulating coolant. The working fluid is pumped through the heated test section to the heat exchanger, where heat is removed by the thermostat. The

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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.
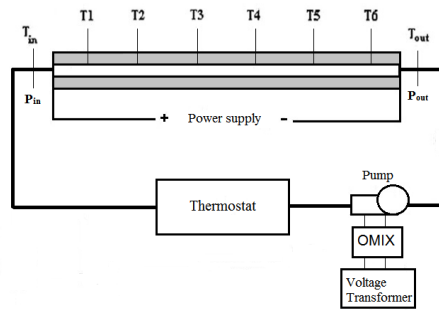
Fig. 1. A schematic diagram of the measuring system.

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. To determine the temperature of the liquid at the outlet the liquid must be very good to mix. To do this, at the exit of the heated channel equipped with a special mixer. The mixer is a cylindrical chamber with a diameter 3 cm and 5 cm in length, filled with polypropylene rings (diameter of 5 mm and a length of 7mm). The mixer and loop section between the heater and the fluid temperature metering point was also insulated.

In all experiments, electric power supplied heater was measured. This electric power compared with the heat capacity transferred to liquid. The installation was well heat insulated. Therefore, the difference between the thermal and electrical capacities not exceed 3-5%.

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.

## 3 The testing of the experimental setup

Figure 2 shows the comparison of the experimental data on the average Nusselt number depending on the Reynolds number with the known empirical correlation [2]:

$$Nu = 0.021 \cdot \mathrm{Re}^{0.8}\, \mathrm{Pr}^{0.43},$$

Where $Nu = \alpha d / \lambda$ – is the Nusselt number, $\alpha = GC_P(T_{out} - T_{in})S^{-1}(T_w - \bar{T})^{-1}$ – is the average heat-transfer coefficient; G – mass flow rate, C_p – is the specific heat of the

fluid; S – is the area of the lateral surface of the channel; T_{out}, T_{in} – are fluid temperatures at the channel inlet and outlet; $\bar{T} = (T_{in} + T_{out})/2$ – 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; $\mathrm{Re} = \rho U d / \mu$ – is the Reynolds number; λ and μ – are fluid thermal conductivity and viscosity coefficients, respectively; U – is the superficial velocity; and d – is the tube diameter. This values calculated at average temperature $\bar{T} = (T_{in} + T_{out})/2$.

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

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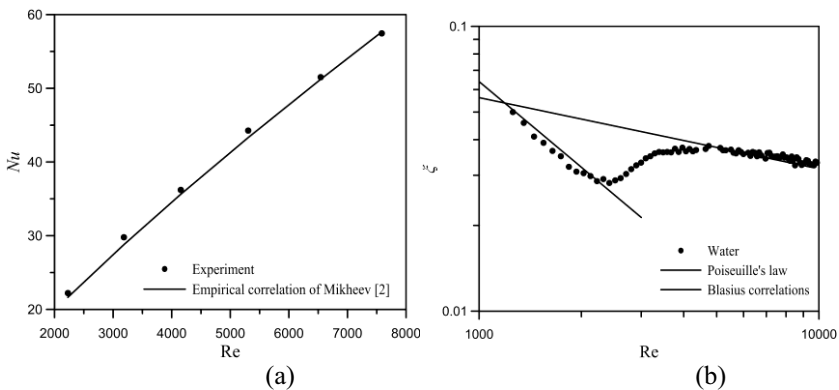
Fig. 2. Heat transfer coefficient (a) and friction factor (b) for pure water versus Reynolds number.

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:

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

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 $\zeta = 64/\mathrm{Re}$ for laminar flow and Blasius correlation $\zeta = 0.316\,\mathrm{Re}^{-0.25}$ 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%.

Initial conditions ($t=0$): $T=T_{g0}$, $C_w=0$, $C_g=1$, $\psi=0$, $\omega=0$ at $0<x<H$, $0<y<L$; $T=T_{w0}$ at $0<r<R_d$, $0<\varphi<2\pi$.

Boundary conditions ($t>0$):

$$T = T_g, \; C_w = 0, \; C_g = 1, \; \frac{\partial \psi}{\partial y} = U_g \;\; \text{at} \;\; y = 0, \, 0 < x < H; \tag{1}$$

$$T = T_a, \; \frac{\partial^2 C_w}{\partial y^2} = 0, \; \frac{\partial^2 C_g}{\partial y^2} = 0, \; \frac{\partial \psi}{\partial y} = -U_w \;\; \text{at} \;\; y = L, \, 0 < x < H; \tag{2}$$

$$\frac{\partial T}{\partial t} = 0, C_w = 0, C_g = 0, \frac{\partial \psi}{\partial x} = 0 \ \text{at} \ x = 0, 0 < y < L; \tag{3}$$

$$\frac{\partial T}{\partial t} = 0, C_w = 0, C_g = 0, \frac{\partial \psi}{\partial x} = 0 \ \text{at} \ x = H, 0 < y < L. \tag{4}$$

Where t – time, s; T – temperature, K; T_{g0} – initial temperature of gases, K; T_{wo} – initial temperature of water, K; C_w – concentration of water; C_g - concentration of high-temperature gases; $\omega = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}$ – rotation vector velocity, s^{-1}; ψ – flow function, m^2/s; x, y – coordinates of the Cartesian coordinate system, m; H, L – dimensions of solution area, m; r, φ – coordinate of the spherical systems; R_d – radius of droplet, m; T_g – temperature of gases, K; U_g – velocity of high-temperature gases, m/s^2; U_w – velocity of spray flow, m/s^2; T_a – temperature of air, K.

In this formulation of the problem there are adiabatic horizontal and vertical walls with constant temperature $T_g \approx 1170\,\text{K}$ and $T_a \approx 298\,\text{K}$. Velocity of sray flows and velocity gases equal $U_g = 1.5\,\text{m/s}^2$ and $U_a = -5\,\text{m/s}^2$.

Boundary conditions for liquid – gas were set taking into account vaporization:

$$R = R_1, 0 < \varphi < 2\pi, \lambda_1 \cdot \frac{\partial T_1}{\partial R} = \lambda_2 \cdot \frac{\partial T_2}{\partial R} - W_e \cdot Q_e, \rho_2 D_2 \frac{\partial C_w}{\partial r} = W_e, t > 0. \tag{5}$$

Where R_1 – droplet radius at the output of high-temperature gases, m; λ_1 – thermal conductivity of water, W/(m·K); T_1 – water temperature, K; λ_2 – thermal conductivity of water vapor, W/(m·K); T_2 – water vapor temperature, K; W_e – evaporation rate, kg/(m^2·s); Q_e – water vaporization heat, J/kg; ρ – water vapor density, kg/m^3; D_2 – diffusion coefficient, m^2/s.

The simulation used Ansys package. To solve this problem used time steps $\Delta\tau = 0.01$ s and grid $\Delta x = 0.01$ mm.

4 The results of the experiments

Numerical and experimental investigation of the influence of additives of high molecular polymers on the characteristics of turbulent heat transfer of nanofluids was conducted.. The Reynolds number is varied from 2000 to 10000. Distilled water was used as a base fluid. The polymers used range of acrylic polymers with a molecular weight of from 1 to 20MDa anionic charge and from 3 to 40 %. As nanoparticles was used a powder of aluminum oxide. The diameter of the nanoparticles ranged from 10 to 100 nm. The volumetric particle concentration was equal to 2%. The obtained suspension was subjected to hour ultrasonic bath "Sapphire".

The experiments were measured and the average value of the local heat transfer coefficient at the channel walls, and the pressure drop at the inlet and outlet of the channel at different concentrations and polymer nanoparticles.

It has been shown that the addition of polymer does not reduce the value of the coefficient of heat transfer at turbulent forced convection of nanofluid (see fig. 3).
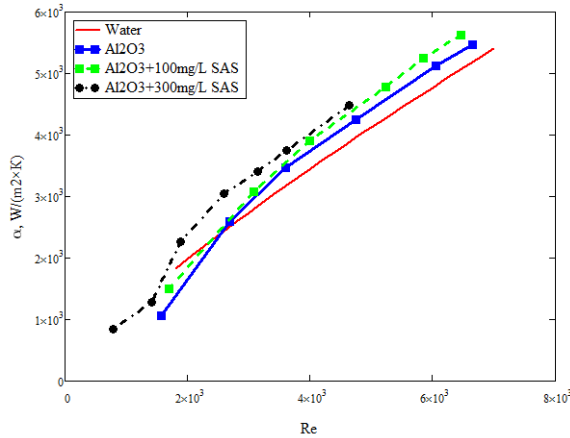
Fig. 3. The dependence of the average heat transfer coefficient on the Reynolds number and the surfactant concentration (anionic polyacrylamide flocculant, $M = 20 \cdot 10^6 – 22 \cdot 10^6$ anionic charge - 40%).

The measured values of the pressure drop in the channel for turbulent flow of nanofluids with additives of polymers (fig. 4). It is experimentally shown that the addition of high molecular weight polymers can reduce friction during turbulent flow of nanofluids. So in particular, it is shown that adding polyacrylamide in the nanofluid reduces the pressure drop in the channel in the turbulent flow regime by approximately 60% compared to the nanofluid without polymer. With the increase of polymer concentration, the pressure drop decreases monotonically and goes to the next level after that does not depend on the concentration of the polymer. This reduction occurs only in turbulent flow regime, which suggests that the reason for the decline is due to the Toms effect. It was found that for lower pressure drops in turbulent flow of nanofluids are needed in some times higher concentration of polymer than needed for pure liquid. Thus, in particular, to reduce the pressure drop by 67 % for pure water, the concentration of polyacrylamide of about 100 ml/L, in nanofluids the concentration of polymer for the same reduction of friction should be at 250-300 mg/L.
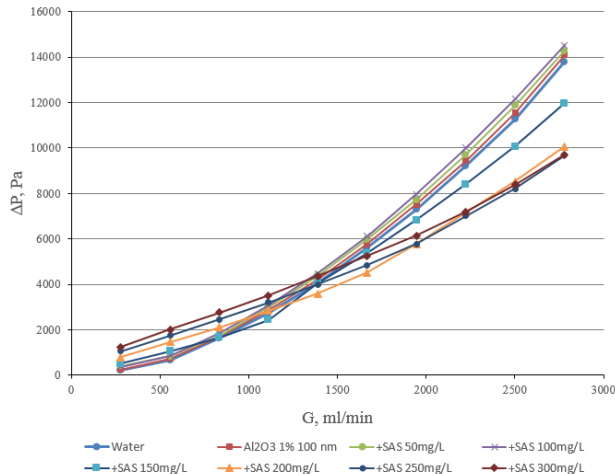
Fig. 4. The differential pressure dependence of nanofluids based on Al_2O_3 particles on the flow rate and different concentrations of surfactants. (anionic polyacrylamide flocculant, $M = 20 \cdot 10^6 – 22 \cdot 10^6$ anionic charge – 40 %).

This is, adsorption of the molecules of the polymer nanoparticles. A study of the molecular weight of the polymer to reduce the value of the pressure when pumping nanofluids. The results are shown in fig. 5.
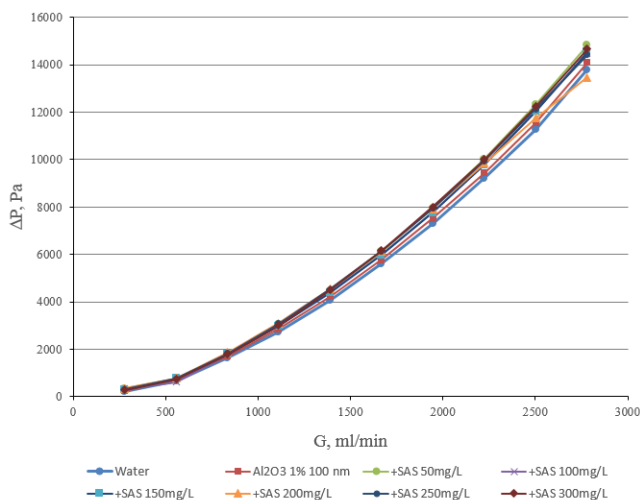
Fig. 5. The differential pressure dependence of nanofluids based on Al_2O_3 particles on the flow rate and different concentrations of surfactants. (anionic polyacrylamide flocculant, $M = 1 \cdot 10^6$ anionic charge – 3 %).

The influence of molecular weight and anionic charge of the polymer to reduce the value of friction pumping nanofluids in the turbulent regime. So in particular, it was shown that the additive of polyacrylamide with a molecular weight of 1 MDA and a concentration of 300 mg/L for nanofluids with particles of aluminum oxide almost does not reduce friction. While the same additive of polyacrylamide with a molecular mass of 20 MDA reduces friction by about 50 %. Thus it is shown that for maximum effect to reduce friction it is necessary to apply a polymer with a higher molecular weight.

Acknowledgments

References

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