Наведено результати експериментального дослідження коефіцієнтів тепловіддачі і втрат тиску при вимушеному ламінарному русі в трубі холодоносіїв на основі пропіленгліколю з домішками наночастинок Al_2O_3. Експериментальні дослідження проводилися на оригінальній установці. Показано інтенсифікація тепловіддачі і зростання втрат тиску при вимушеному русі нанохолодоносія, в порівнянні з базовим холодоносієм

Ключові слова: холодоносій, наночастинки, ламінарний режим, коефіцієнт тепловіддачі, втрати тиску, експериментальна установка

Представлены результаты экспериментального исследования коэффициентов теплоотдачи и потерь давления при вынужденном ламинарном течении в трубе хладоносителей на основе пропиленгликоля с примесями наночастиц Al_2O_3. Экспериментальные исследования проводились на оригинальной установке. Показана интенсификация теплоотдачи и рост потерь давления при вынужденном движении нанохлаоносителя, по сравнению с базовым хладоносителем

Ключевые слова: хладоноситель, наночастицы, ламинарный режим, коэффициент теплоотдачи, потери давления, экспериментальная установка

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EXPERIMENTAL STUDY OF HEAT EXCHANGE AND HYDRODYNAMICS AT THE LAMINAR FLOW OF NANOCOOLANT BASED ON PROPYLENE GLYCOL AND Al_2O_3 NANOPARTICLES

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1. Introduction

Intensification of the heat exchange process in different thermal power systems is an important task, since it contributes to the rational utilization of energy resources, as well as to the reduction in material consumption of heat exchange equipment.

One of the most promising ways of the heat exchange intensification, not requiring design changes in the equipment, is the use of nanofluids, obtained on the basis of traditional heat-transfer agents (water, mineral oils, ethylene glycol and other), instead of these agents. The term "nanofluid" is applicable to ultra-dispersed systems, which consist of basic liquid and the nanoparticles of dimensions on the order of 1–100 nm. The major advantage of the application of nanofluids as heat-transfer agents is in the fact that they have the higher value of thermal conductivity in comparison with the basic liquid. For this reason, nanofluids are promising heat-transfer agents for the heat exchange equipment under laminar conditions of the liquid motion [1–5]. An overview of studies of the heat transfer process with participation of nanofluids, performed in recent years [1–8], showed that experimental values of heat transfer coefficient do not coordinate with theoretical values, assessed by the existing dependences taking into account properties of nanofluids. At the same time, the detailed, physically substantiated ideas about the mechanism of the heat transfer intensification with the participation of nanofluids have not been sufficiently developed so far. For this reason, additional experimental studies, directed toward studying the influence of nanoparticles on the heat transfer intensity at the forced motion of nanofluids in the pipes and channels, are necessary.

In connection with traditional refrigerating systems, it is possible to state that heat transfer at the forced convection of coolants in their elements is characterized by low intensity. Therefore, a study of the possibilities of heat transfer intensification due to the use of nano-technologies for the coolant based on aqueous solution of propyleneglycol, which is widely used in practice, is a relevant task.

2. Literature review and problem statement

An analysis of articles [1–5], devoted to experimental studying of heat-transfer coefficient at the laminar motion of nanofluids in pipes revealed that the additives of nanoparticles always lead to the intensification of heat transfer. In paper [1], an increase in heat-transfer coefficient for laminar flow conditions in a pipe was shown for the nanofluid based on distilled water and nanoparticles of Al_2O_3 (0.1–2 % by volume). For the nanofluid of 2 % by volume, the increase in heat-transfer coefficient comprised approximately 32 % in comparison with pure water. Furthermore, in this paper, authors noted an increase in the losses of head at the use of nanofluids, which, as well as an increase in the heat-transfer coefficient, is proportional to the concentration of nanoparticles.

Article [2] explored influence of additives of nanoparticles of CuO and Al_2O_3 to distilled water on the heat transfer intensification at the laminar nanofluid flow in a pipe. In the conducted studies, an increase in the heat-transfer coefficient with an increase in the concentration of nanoparticles (0.2–3.0 % by volume) was shown. Furthermore, authors note that impurities of nanoparticles of Al_2O_3 lead to a somewhat larger increase in the values of heat-transfer coefficient than the impurities of nanoparticles of CuO at identical concentration.

In paper [3], heat transfer intensity at the laminar flow of two nanofluids on the water base was examined. For the nanofluid with the content of 3 % by volume of nanoparticles of Al_2O_3, an 8 % increase in thermal conductivity and a 20 % increase in heat-transfer coefficient (in comparison with pure water) were shown. For the nanofluid with the content of 3.5 % by volume of nanoparticles based on amorphous carbon, only an insignificant increase in thermal conductivity and an 8 % increase in heat-transfer coefficient (HTC) were observed.

In article [4], for the nanofluid based on water and nanoparticles of Al_2O_3 (0.6; 1.0 and 1.6 % by volume), an increase in heat-transfer coefficient at the laminar flow of liquid in the initial thermal section of a pipe proportional to the concentration of nanoparticles was indicated. Maximum effect was observed for the nanofluid with the content of 1.6 % by volume of nanoparticles, and accounted for 47 % of the magnitude of heat-transfer coefficient for the basic liquid.

Article [5] demonstrates an increase in heat-transfer coefficient for the nano-fluid water/propylene glycol/nanoparticles of CuO in comparison with the same magnitude for the basic liquid. It was shown that an increase in heat-transfer coefficient is proportional to the concentration of nanoparticles of CuO (0.025; 0.10 and 0.50 % by volume). Authors of paper [5] argue that although the losses of head increase proportionally to the concentration of nanoparticles, an increase in heat-transfer coefficient is more essential.

In accordance with the overview of data from the literature, it is possible to draw a conclusion about the prospects of using nanoparticles as an additive to heat-transfer agents in order to increase heat-transfer coefficient. Furthermore, it was shown that there are no sufficient studies in the literature, dedicated to investigating heat transfer intensity with the use of propylene glycol as the base for preparing the nanofluids. It should be noted that this substance is the basic component of coolants, widely used in the refrigerating equipment and air-conditioning systems. A high anticorrosive durability and low toxicity are major advantages of coolants based on propylene glycol. At the same time, coolants based on aqueous solutions of propylene glycols have their shortcomings, for example, high viscosity at temperatures, characteristic for the area of application in refrigeration engineering.

Lately, there have been studies, dedicated to the research into thermophysical properties of nanofluids based on propylene glycol, showing the prospects of their applications as heat-transfer agents. Thus, in article [9], influence of the additives of nanoparticles of CuO (0.025; 0.1; 0.4; 0.8 and 1.2 % by volume) on the viscosity of the solution propylene glycol/water (60/40 % by weight) was explored. Paper [10] is dedicated to experimental estimation of thermal conductivity of nanofluids based on the aqueous solution of propylene glycol and nanoparticles of Al_2O_3 and TiO_2. Article [11] explores influence of the additives of nanoparticles, obtained by mechanical grinding of river sand, on the thermal conductivity and viscosity of pure propylene glycol. Although in the pure form propylene glycol is not used as a heat-transfer agent and coolant, results of the work are interesting, since they showed that additives of nanoparticles may lead to a decrease in the viscosity of nanofluid at low temperatures. Similar results on the reduction in viscosity at the addition of nanoparticles of ZnO both as pure propylene glycol and its aqueous solution were obtained in papers [12, 13].

Therefore, further study into influence of the impurities of nanoparticles in multi-component coolants, based on propylene glycol, on the heat transfer intensity at the forced flow in pipes and channels of heat exchangers at temperatures, characteristic for the application in refrigeration engineering, presents both scientific and practical interest.

3. The aim and tasks of the study

The aim of present work is an experimental study of the process of heat transfer under laminar flow of nanofluids based on aqueous solutions of propylene glycol, promising as coolants for refrigeratory and air conditioning systems.

To achieve the set aims, the following tasks were to be solved:

– creation of the test bench to study heat exchange and hydrodynamics in the process of the forced flow of nanofluids in a pipe;

– development of technology for the creation of nanofluid based on the aqueous solution of propylene glycol and nanoparticles of Al_2O_3, different in terms of aggregate stability;

– experimental and calculated estimation of thermophysical properties of nanocoolants based on the aqueous solution of propylene glycol and nanoparticles of Al_2O_3;

– experimental study of the mean and local values of heat-transfer coefficients, as well as losses of head in the initial thermal section of experimental installation under the laminar flow of nanocoolants.

4. Procedure of experimental study

4. 1. Schematic of the experimental installation

A schematic of the experimental installation to study the process of heat transfer and hydrodynamics under the flow of nanocoolants in a pipe is represented in Fig. 1.

The following designations are used in Fig. 1: 1 – pressure transducer Vika; 2, 17 – systems to control temperature of thermostats; 3 – storage reservoir; 4, 15 – thermostats; 5 – Dewar flask for the cold–soldered joint of absolute thermocouples; 6 – multipoint switch; 7 – computer; 8 – multimeter Picotest M3510A; 9 – power source for the pressure transducer Vika; 10 – power source for the circulation pump; 11 – the model resistor coil P321; 12 – vortex type circulating pump for the heat-transfer agent – (pumps 15WBX-12); 13 – regulating valve; 14 – refrigeration machine; 16 – gear flow meter Oval Gear EM008; 18 – circulation pump for thermostatic liquid; 19 – working section of the installation (mobile copper pipe); 20, 23 – Pitot tubes; 21 – block of temperature measurement of thermostatic liquid; 22 – external pipe of heat exchanger; 24 – output of thermocouples to electrical connector; 25 – electrical connector; 26 – oil seals; 27 – switch to measure full head of coolant; 28 – insulation.

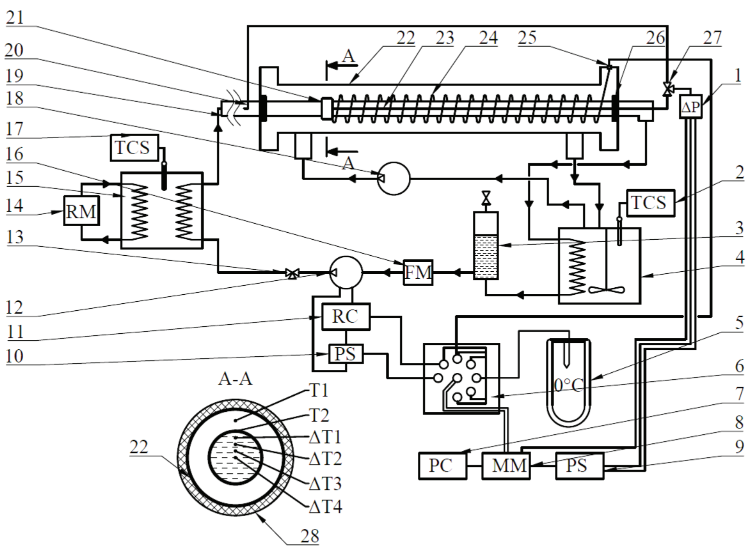


Fig. 1. Schematic diagram of experimental installation “pipe in a pipe”

The channel, along which the examined nanocoolant was moving, was the mobile copper tube 19 of diameter 0.01×0.001 m. The copper tube was encased (put in a pipe) 22. In the process of experiment, the heat-transfer agent (water) was pumped through the annular channel, formed by two tubes 19 and 22 at constant flow rate of 0.060 kg/s at the temperature, which corresponds to the conditions of conducting the experiment. Consumption of the coolant in the copper tube was varied with the help of regulating valve 13. Heat balance by the nano-heat-transfer agent and water coincided with accuracy of up to 5 %.

Measuring the temperatures, necessary for evaluating the local heat-transfer coefficients, was conducted at distance X=0.10; 0.25; 0.40; 0.55; 0.70 and 0.85 m from the inlet of the coolant into the working section. The measurements of temperatures of the coolant in the cross-section of the pipe were carried out with the help of four differential thermocouples, located in one cross-section at a different distance from the axis of the pipe. Additionally, temperature of the wall of the copper pipe in the examined cross-section was determined. Coolant consumption was determined with the help of the flow meter Oval Gear EM008 (Russia). The measuring section of experimental installation to study local heat-transfer coefficients was characterized by hydrodynamically stabilized flow conditions of liquid. For this purpose, the inlet of coolant into the measuring section was preceded by practically isothermal section of the pipe with length around 1 m.

Pressure losses at the forced flow of the coolant in the pipe were determined in the measuring isothermal section of experimental installation, located before the inlet into working section of HTC study.

Performed analysis of the extended uncertainty of magnitudes, measured in the experiment, reveals that absolute uncertainty of temperature measurement reaches 0.15 K, of diameter – 0.05 mm, of the distance between the inlet into the measuring section to the assigned cross-section of the pipe – 5 mm, and of coolant consumption – 1.0 %. Based on the indicated magnitudes, the extended uncertainty of determining heat-transfer coefficient, taking into account random component, does not exceed 15 %.

4. 2. Methods for processing experimental data

For determining experimentally the local heat-transfer coefficient in the cross-section of the pipe, in which the forced motion of the heat-transfer agent occurs, we used equation:

$$h_{\exp(X)} = q_{(X)} \Big/ \left(T_{w(X)} - T_{fl(X)}\right), \quad (1)$$

where $q_{(X)}$ is the heat flow density in the specific cross section X of the measuring section of pipe, W/m^2; $T_{w(X)}$ is the temperature of inner surface of the pipe in the specific cross-section of measuring section of the pipe, K; $T_{fl(X)}$ is the estimated temperature of the coolant in the particular cross-section of pipe, K.

Mean temperature of the coolant in the cross section of the pipe was calculated as

$$T_{fl} = \frac{1}{\bar{\upsilon}} \int_0^{f0} \upsilon_{(r)} T_{(r)} df, \quad (2)$$

where $\bar{\upsilon}$ is the mean flow rate (determined according to the value of coolant consumption), m/s; $T_{(r)}$ is the value of coolant's temperature at distance r from the axis of the pipe, K, measured during experiment; $\upsilon_{(r)}$ is the fluid velocity at distance r from the axis of the pipe under laminar mode, which was determined by equation (3), m/s:

$$\upsilon_{(r)} = 2\bar{\upsilon}\left(1 - \left(r/r_0\right)^2\right), \quad (3)$$

where r_0 is the radius of the pipe, m.

Mean heat flow density in the section of the pipe of length $L = X_i - X_{i+1}$ (where X_i and X_{i+1} are the distances from the inlet of coolant into working section of the pipe to cross-section i and i+1, respectively, m):

$$q_{(X_i - X_{i+1})} = \frac{G_{fl} \cdot C_{Pfl}\left(T_{fl\,i} - T_{fl\,i+1}\right)}{\pi \cdot 2 \cdot r_0 \cdot (X_i - X_{i+1})}, \quad (4)$$

where G_{fl} is the mass consumption of coolant in the pipe, kg/s; C_{Pfl} is the thermal capacity of coolant at mean temperature in section X_i–X_{i+1}, J/(kg·K); $T_{fl\,i}$ and $T_{fl\,i+1}$ are the mean by cross section of pipe temperatures of coolant in cross section X_i and X_{i+1}, respectively, K.

By dependence $q_{(X_i - X_{i+1})} = f(l)$ we determined magnitude $q_{(X)}$ for each examined cross section of the pipe.

For determining experimentally the mean value of heat-transfer coefficient lengthwise the pipe at the forced motion of heat-transfer agent, we used equation:

$$q_{(X_1-X_6)}=\frac{G_{fl}\cdot C_{Pfl}\left(T_{fl\,1}-T_{fl\,6}\right)}{\pi\cdot 2\cdot r_0\cdot (X_1-X_6)}, \quad (5)$$

$$h_{exp}=q_{(X_1-X_6)}\Big/\left(T_w-T_{fl}\right), \quad (6)$$

where T_w is the mean temperature of inner surface of measuring section of the pipe, K; T_{fl} is the mean value along the length of measuring section temperature of the coolant, K.

To determine coefficient of friction resistance under the flow of coolant along the pipes, we used dependence (7):

$$f_{exp}=\frac{4\cdot\Delta P\cdot r_0}{\bar{\upsilon}^2\cdot\rho\cdot l}, \quad (7)$$

where ΔP is the experimentally defined value of pressure losses in the measuring section, Pa; ρ is the fluid density at mean temperature, kg/m^3; l is the length of section, m.

Theoretical value of friction coefficient was calculated according to formula:

$$f_{teor}=64/\mathrm{Re}. \quad (8)$$

Dependence (8) is applicable for the laminar mode of motion of liquids in pipes and channels.

5. Technology of preparing the objects of study and their thermophysical properties

One of the most important tasks that the researchers must solve before starting measurements of thermophysical properties of nanofluids and study the process of heat exchange using them is the estimation of aggregate stability of the examined samples. A correct choice of technology for the preparation of nanofluid samples and the evaluation of their aggregate stability in the process of studies define reliability of obtained data in many cases.

5. 1. Preparation of objects of study and the aggregate stability of nanocoolants

Aqueous solutions of propylene glycol, which, when used as coolants, have main disadvantage – a high value of viscosity, were examined in present work as the objects of study. As studies [1–3, 8, 9] demonstrated, the addition of nanoparticles to the base liquid contributes to an increase in viscosity. Although, as authors of papers [11–13] note, the opposite effect from the existence of nanoparticles in the liquid may also be observed. Therefore, to decrease viscosity and to intensify the heat transfer process, we made a decision on adding to the base coolant a component, which contributes to a decrease in viscosity of aqueous solutions of propylene glycol at low temperatures, characteristic for the application in refrigeratory equipment. Preliminary studies into the influence of different concentrations of substances, which reduce viscosity, on the magnitude of density and viscosity of aqueous solutions of propylene glycol, were carried out by authors of the present article [14, 15].

As the objects of study in the work, we explored:

– coolant with the content of propylene glycol/water (54.00/46.00 % by weight) – C (PG/W);

– coolant with the content of propylene glycol/water/substance, regulating viscosity (48.60/46.52/4.88 % by weight) – C (PG/W/RV);

– nanocoolant with the content of propylene glycol/water/substance, regulating viscosity/nanoparticles Al_2O_3 (48.24/46.38/4.85/0.53 % by weight) – nano C 0.53 %;

– nanocoolant with the content of propylene glycol/water/substance, regulating viscosity/nanoparticles Al_2O_3 (48.00/46.15/4.82/1.03 % by weight) – nano C 1.03 %.

For the preparation of nanofluid, we used nanoparticles α-Al_2O_3 (Wenzhou Jingcheng Chemical Co) with dimension in powder 10±5 nm (data provided by manufacturer).

For the preparation of nanocoolants, a two-stage method was used, in accordance with which nanoparticles in the form of powder were mixed up with a part of the base liquid and underwent dispersion (for 24 hours). The process of dispersion took place in the beaded mill, filled with ZrO_2 balls, 2 mm in diameter. Additionally, after each 3–4 hours, the nanofluid was exposed to ultrasonic treatment for 30 min. The obtained nanofluid with the high concentration of nanoparticles was diluted to the required composition before putting into the experimental installation. Nanocoolants were prepared without using surface active substances because their admixtures may influence thermophysical properties of the examined object.

It should be noted that by using a two-stage method for the preparation of nanocoolants, it is impossible to obtain the size of nanoparticles in liquid, which corresponds to the size of nanoparticles in powder. A part of nanoparticles exists in liquid in the form of aggregates, which, at correctly implemented dispersion technology, must remain as nano-dimensional. To control the processes of agglomeration and sedimentation of nanoparticles, dispersed in the liquid over the course of time, average size for two identical samples of nanocoolant was determined by using the turbidimetry method [16]. Practical implementation of this method implied measurement of spectral dependence of optical density of nanocoolants' samples on the wavelength of transmitted light (in the interval λ=700–990 nm). The studies of optical density of the samples were carried out in the spectrophotometer Shimadzu UV-120-02 in the airtight plane-parallel optical cells at the length of optical path 4.05 mm.

Optical density of the samples was measured repeatedly for 30 days, for the first sample without preliminary shaking, for the second sample – with pre-mixing directly in the optical cell before each measurement. Thus, for the second sample, conditions of constant mechanical stirring of nancoolant during its circulation in the installation for studying heat-transfer coefficient were simulated.

Results of the study demonstrate that the dimensions of nanoparticles, assessed at the assigned height of optical cell for 12 days, decreased from 240 nm to 145 nm under static storage conditions. Subsequently, no decrease in dimensions was observed. A conclusion on the establishment of sedimentation-diffusion equilibrium in the nanocoolant was made. For the samples, subjected to stirring, the dimensions of nanoparticles remained approximately constant and equal to 240 nm. Thus, the obtained data indicate aggregate stability of the examined nanofluid samples. This result is important as it shows that the obtained samples of nanofluids might be used in devices with the forced circulation of coolant without changing its dispersed composition over time.

5. 2. Thermophysical properties of the objects of study

For the estimation of experimental values of pressure losses and heat-transfer coefficients, it is necessary to possess reliable information on the thermophysical properties of the objects of study.

The study of density of the base coolant C (PG/W/RV) was conducted by pycnometric method. A pycnometer was placed in the thermostat, equipped with the automatic temperature control system. Temperature fluctuations in the thermostat did not exceed ±0.02 K. To measure the level of coolant sample in the pycnometer, the cathetometer KM-8 was used. The experiment was conducted in the range of temperatures from 253 to 313 K. Performed analysis demonstrates that the extended uncertainty of density measurement of the coolants did not exceed 0.2 %.

The obtained experimental data on the density for C (PG/W/RV) were approximated by equation (9) with the following coefficients: A=–0.6245, B=1217.6 for C (PG/W/RV).

$$\rho = A\cdot T + B, \quad (9)$$

where T is temperature, at which thermophysical properties of substance are determined, K.

Density ρ_{nf} for nanofluids of nanocoolant 0.53 % and nanocoolant 1.03 % was evaluated according to the rule of additivity (10).

$$\rho_{nf} = \rho_{np}\cdot\varphi + \rho_{bf}\cdot(1-\varphi), \quad (10)$$

where ρ_{np} is the density of nanoparticles, kg/m^3; ϕ is the volumetric share of nanoparticles in nanofluid; ρ_{bf} is the density of the base C (PG/W/RV), kg/m^3.

Data on the density of C (PG/W) were taken from a reference book [17].

Experimental studies of viscosity of the base coolant and nanocoolants were carried out in the experimental installation whose main element is a glass capillary viscometer with the suspended level. The conducted studies demonstrated that temperature fluctuations in the thermostat did not exceed ±0.05 K. Uncertainty in measurement of temperature was lower than 0.2 K. Performed analysis reveals that the extended uncertainty in measurement of kinematic viscosity in the examined range of parameters from 253 to 313 K did not exceed 0.5 mm^2/s.

Obtained experimental viscometric data on the coolants were approximated by the Walter equation (11) with the following coefficients: A=1.2231, B=13.25504, C=5.385791 for the C (PG/W); A=1.3908, B=12.72117, C=5.174600 for the (PG/W/RV); A=1.5929, B=12.52073, C=5.087327; for the nanoC 0.53 %; A=1.9429, B=11.91202, C=4.839401 for the nanoC 1.03 %.

$$\lg(\lg(\nu + A)) = B - C\cdot\lg(T), \quad (11)$$

where ν is the kinematic viscosity, mm^2/s.

A study of thermal capacity of the base coolant and the nanocoolant was conducted by the method of monotonous heating in the adiabatic calorimeter [18]. The experiment was performed in the range of temperatures from 243 to 328 K. The extended uncertainty in the measurement of heat capacity of the coolants did not exceed 0.45 %.

Obtained experimental data on heat capacity were approximated by equation (12) with the following coefficients: A=0.00010815, B=0.053257 for the C (PG/W/RV); A=0.000096026, B=0.057539 for the nanoC 0.53 %; A= =0.000084106, B=0.062054 for the nanoC.

$$C_p = (A + B/T)^{-1}, \quad (12)$$

where C_p is the heat capacity, J/(kg·K).

Data on heat capacity of the C (PG/W) were taken from a reference book [17].

For calculating thermal conductivity of nanocoolants, correlation (13), proposed in paper [19], was used:

$$k_{nf} = \left[\frac{k_{np} + 2\cdot k_{bf} + 2(k_{np} - k_{bf})(1+\beta)^3\varphi}{k_{np} + 2\cdot k_{bf} - (k_{np} - k_{bf})(1+\beta)^3\varphi}\right]k_{bf}, \quad (13)$$

where k_{np} is the thermal conductivity of nanoparticles, W/(m·K); k_{bf} is the thermal conductivity of the base liquid, calculated by procedure, proposed in [20] using the properties of components of heat-transfer agents, data on which were cited in [17, 21], W/(m·K); β is the ratio of thickness of layer of the base liquid, adsorbed at the surface of nanoparticles, to the true radius of nanoparticle, value β=0.1 was accepted for calculations [19].

6. Results of experimental study of heat-transfer coefficients and pressure losses under the laminar flow of coolants in the pipe

6. 1. Results of the calibration experiment

Reliability of experimental data in the study of heat-transfer coefficients at the forced convection of nanofluids in the pipes is usually proved by conducting a calibration experiment. In the present study, within the framework of calibration experiment, heat-transfer coefficient at the forced motion of distilled water in the interval of the Reynolds numbers, characteristic for laminar conditions, was determined. Values of the Nusselt numbers (both mean by length and the local ones), obtained in the experiment, were compared to theoretical values. For this purpose, we ran an analysis of empirical dependences, found in the literature, to calculate a heat-transfer coefficient at the laminar motion of liquids in pipes and channels under the boundary conditions of first order (constant value of wall temperature) [22–24].

In article [22, 23], for calculating the mean lengthwise value of heat-transfer coefficient in the initial thermal section (hydrodynamic boundary layer was formed) at T_w=const, it is recommended to use dependence:

$$\overline{Nu} = 3.66 + \frac{0.0668\cdot(d/l)\cdot Re\cdot Pr}{1+0.04\cdot\left[(d/l)\cdot Re\cdot Pr\right]^{2/3}}. \quad (14)$$

When computing the values of similarity numbers in equation (14), data on thermophysical properties of liquid are accepted at the mean lengthwise temperature of the flow.

In papers [23, 24], for calculating the mean lengthwise heat-transfer coefficient in the initial thermal section at T_w=const (as well as at slightly changing lengthwise temperature of the wall), it is recommended to use dependence:

$$\overline{Nu} = 1.55\left(\frac{Re\cdot Pr}{l/d}\right)^{1/3}\left(\frac{\mu_l}{\mu_w}\right)^{0.14}. \quad (15)$$

In papers [23, 24], it is recommended to use equation (15) at $\frac{1}{\text{Re}\cdot\text{Pr}}\cdot\frac{X}{D}\leq 0.05.$

In article [23], for calculating the local lengthwise heat-transfer coefficient in the initial thermal section at T_w=const (as well as at slightly changing lengthwise temperature of the wall) it is recommended to use dependence:

$$\overline{\text{Nu}} = 1.03\left(\frac{\text{Re}_d\cdot\text{Pr}}{X/d}\right)^{1/3}\left(\frac{\mu_l}{\mu_w}\right)^{0.14}. \qquad (16)$$

In paper [23], it is recommended to use equation (16) at

$$\frac{1}{\text{Re}\cdot\text{Pr}}\cdot\frac{X}{D}\leq 0.01.$$

The properties of liquid in equations (15) and (16) must be accepted at the temperature, which is mean between temperature of the flow and that of the wall.

The inner diameter of the pipe is the determining size, introduced to numbers Nu and Re in all the examined equations (14)–(16).

Results of comparison of experimental values of the mean Nusselt number for distilled water to theoretical values, estimated by dependences (14) and (15), are given in Fig. 2. Local values of heat transfer coefficient (experimental and theoretical values) for two different Reynolds numbers are given in Fig. 3.

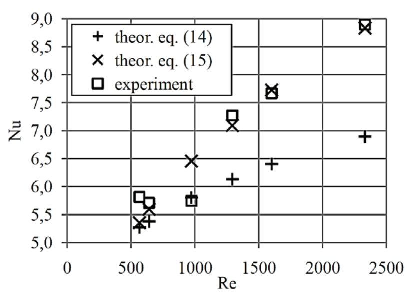
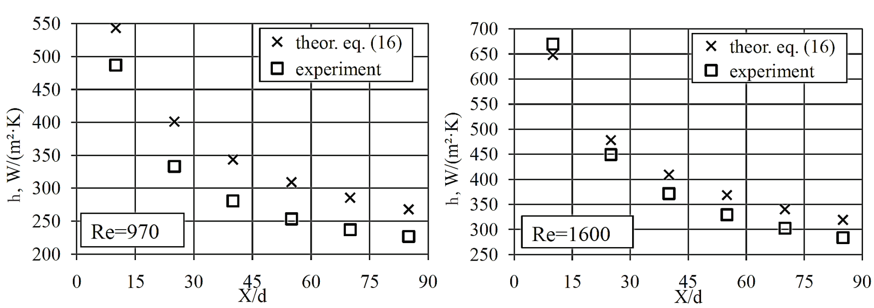


Fig. 2. Dependence of the mean value of Nusselt number on the Reynolds number at the forced motion of distilled water in the pipe at d=0.01 and l=0.85 m (initial thermal section)



a *b*

Fig. 3. Dependence of local heat-transfer coefficient lengthwise initial thermal section at the forced motion of distilled water in the pipe at d=0.01 and l=0.85 m: *a* – Re=970; *b* – Re=1600

As the conducted study shows (Fig. 2), there is a good agreement between the obtained experimental data and the results of calculation according to equation (15) in the interval of numbers Re≤2300. This conclusion proves correctness of both the data, obtained in the experimental installation and of the procedure for processing primary initial information. Dependence of local heat-transfer coefficient lengthwise the initial thermal section (Fig. 3), obtained in the experiment, also satisfactorily agrees with the values of heat-transfer coefficient, calculated according to equation (16).

6. 2. Results of experimental study of heat-transfer coefficient and pressure losses coefficient under the laminar flow of nanocoolants in the pipe

Parameters of conducting the experiment on determining local and mean heat-transfer coefficients under the forced flow of coolants in the pipe are given in Table 1.

Table 1

Parameters of heat-transfer agents when conducting experimental studies into the process of convective heat transfer at the forced motion of coolant in a pipe

$T_{C\,entr}$, K	$\overline{\upsilon}$, m/s	Re	$T_{C\,entr}$, K	$\overline{\upsilon}$, m/s	Re
C (PG/W)			283,2	0,355	405
278.5	0.267	280	283.2	0.709	785
278.2	0.532	543	283.2	1.180	1372
273.4	0.267	192	nanoC 0.53 %		
273.4	0.462	326	278.5	0.267	280
283.2	0.355	432	278.2	0.532	543
283.2	0.709	837	273.4	0.267	192
283.2	1.179	1294	273.4	0.462	326
C (PG/W/RV)			283.2	0.355	432
278.2	0.267	254	283.2	0.709	837
278.2	0.532	496	283.2	1.179	1294
273.4	0.267	182	nanoC 1.03 %		
273.4	0.461	295	278.2	0.267	254
283.2	0.355	405	278.2	0.532	496
283.2	0.709	785	273.4	0.267	182
283.2	1.180	1372	273.4	0.461	295
278.2	0.532	496	283.2	0.355	405
273.4	0.267	182	283.2	0.709	785
273.4	0.461	295	283.2	1.180	1372

The following information was used in the computation of the Nusselt number values: thermophysical properties of the coolant, experimental values of coolant consumption, temperature of the heat exchange surface, mean across the section temperature of the coolant. Dependence of the mean Nusselt number values for different coolants on the Peclet number for the initial thermal section of the round pipe, obtained as a result of the conducted studies, is represented in Fig. 4.

Dependence of the local values of heat-transfer coefficients of the examined coolants in different cross-sections of the initial thermal section for number Re=715±20 is presented in Fig. 5. Fig. 5, in addition to experimental data, also shows the values of local heat-transfer coefficients for the base coolant (PG/W/RV), calculated by dependence (16).

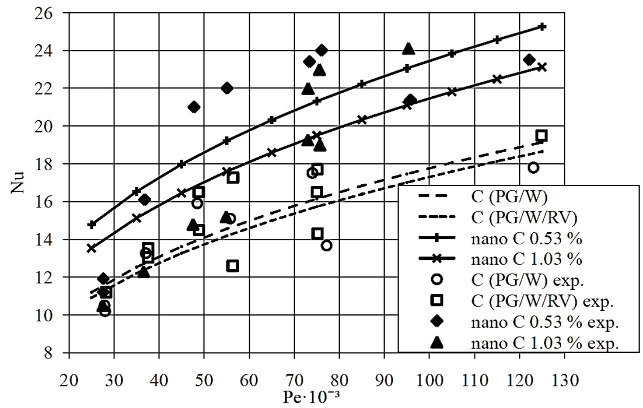


Fig. 4. Dependence of the mean Nusselt number values on the Peclet number at the forced motion of coolants in the pipe at d=0.01 and l=0.85 m (initial thermal section) for the coolant PG/W, base coolant PG/W/RV and nanofluids of the nanoC 0.53 % and the nanoC 1.03 %

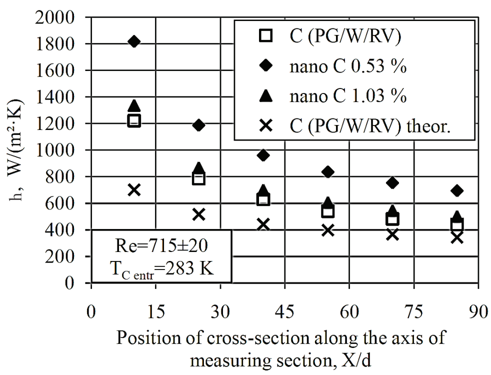


Fig. 5. Dependence of local heat-transfer coefficient lengthwise the initial thermal section at the forced motion of coolants in the pipe (d=0.01 and l=0.85 m)

Pressure losses at the laminar flow of heat-transfer agents in the pipe were determined for the coolant C (PG/W/RV) and the nanocoolant C 1.03 %. The measurements of pressure losses were conducted under the modes, given in Table 1. Ratio of pressure losses, determined for the nanoC 1.03 % to pressure losses for the base coolant C (PG/W/RV) depending on the Reynolds number is given in Fig. 6.

By using experimental data on pressure losses, the values of friction coefficient – f – were calculated by formula (7). Dependence of value f on the Reynolds number for C (PG/W/RV) and the nanoC 1.03 % is presented in Fig. 7.

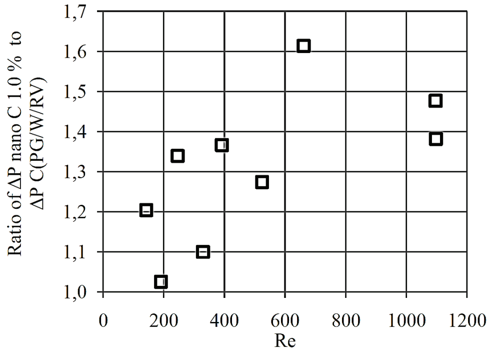


Fig. 6. Dependence of ratio of pressure losses for the nanoC 1.0 % to pressure losses for the base coolant C (PG/W/RV) on the Reynolds number

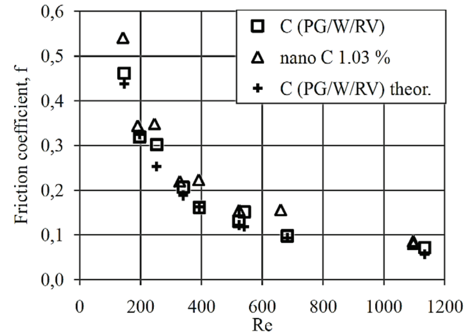


Fig. 7. Dependence of friction coefficient lengthwise on the Reynolds number for the base coolant (PG/W/R) and nanofluid of the nanoC 1.03 %

Fig. 7 additionally presents theoretical values of magnitude f for the base coolant C (PG/W/RV), evaluated according to formula (8). This information will make it possible to estimate a possibility of using traditional dependences of calculating f to the estimation of pressure losses at the flow of nanofluids.

7. Discussion of results of experimental study into heat-transfer coefficient and pressure losses under the laminar flow of nanocoolants in the pipe

Results of the experimental study, given in Fig. 4, demonstrate an increase in mean lengthwise Nusselt number for both examined nanofluids in comparison to the base liquid. At the same time, the influence of admixtures of nanoparticles on the heat exchange intensity is not proportional to the concentration of nanoparticles. Heat exchange intensity for the nanocoolant with the content of nanoparticles 0.53 % by weight is larger than for the nanocoolant with 1.03 % of nanoparticles by weight (Fig. 4, 5). It should be noted that analogous results were obtained by other authors. Thus, in studies [7, 8], no significant change of heat-transfer coefficient depending on the change in concentration of nanoparticles (maximum concentration amounted to 0.2 % by volume) was registered. But article [9] shows an increase in heat-transfer coefficient by approximately 26 % at the concentration of 1.5 % of TiO_2 nanoparticles by volume in water and a decrease in heat-transfer coefficient by approximately 14 % at the concentration of nanoparticles at 2.0 % by volume.

Taking into account obtained results of the experimental study of heat-transfer coefficient under the laminar flow of nanofluids in the pipe, it is possible to formulate a conclusion: an increase in thermal conductivity is not the only reason for the intensification of the heat transfer process.

An increase in pressure losses for the nanofluid (Fig. 6) was expected in connection with the increase in viscosity of coolants at the addition of nanoparticles to them, registered in the experimental studies. It is evident from Fig. 7 that for the nanocoolant, the magnitude of friction coefficient f is somewhat larger than for the base coolant. In accordance with the obtained results, we made a conclusion that an increase in viscosity is not the only explanation for the increase in pressure losses under the laminar motion of nanocoolants in the pipe.

The results we obtained indicate a rather complicated nature of influence of nanoparticles on the value of heat-transfer coefficient and an increase in pressure losses under the laminar flow of nanocoolants in the pipe. A more detailed study of the heat transfer process under the laminar flow of nanocoolants requires consideration of not only their thermophysical properties but also of additional mechanisms of the heat exchange intensification, related to the existence of nanoparticles.

The results, obtained in present work, are necessary for further improvement in refrigeration equipment for the purpose of increasing its energy effectiveness. The studies might also prove useful for the development of heat-transfer agents and coolants in other sectors, such as thermal power engineering, electronics, etc.

Taking into account the relevance of task on the introduction of nano-technologies into the refrigerating and energy industry, we consider it expedient to conduct further studies into the influence of nanoparticle additives to heat-transfer agents and coolants on the intensity of heat transfer processes in heat exchangers.

8. Conclusions

1. A schematic of the original experimental installation to study the processes of heat exchange and hydrodynamics under the forced flow of nanofluids was devised; the calibration experiment was performed using distilled water as coolant, which demonstrated reliability of the obtained experimental data.

2. A technology for the preparation of coolant based on the aqueous solution of propylene glycol and Al_2O_3 nanoparticles of was explored; aggregate stability of the obtained samples of nanofluids and the possibility of using the examined nanofluids in heat exchange equipment under the forced circulation of coolant were shown.

3. It was demonstrated that the additives of nanoparticles to the coolant lead to an increase in viscosity in comparison to the base coolant by 2 % for the nanoC 1.03 % at temperature 253 K and by 12 % for the nanoC 1.03 % at temperature 313 K; an insignificant increase in viscosity at low temperatures indicates the prospects for using the nanofluids, examined in present work, as coolants in the refrigeration systems.

4. It is shown that the admixtures of nanoparticles in a coolant lead to the intensification of heat transfer; the existence of Al_2O_3 nanoparticles in the base coolant in the amount of 0.53 % by weight contributed to a larger increase in the mean Nusselt number lengthwise the working section (about 25 % in comparison with the base coolant) than the presence of 1,03 % by weight of admixtures of nanoparticles (about 20 %); the admixtures of Al_2O_3 nanoparticles in the base coolant lead to an increase in pressure losses under the flow of nanocoolant in the pipe. We made a conclusion that an increase in thermal conductivity of nanofluids is not the only factor, which contributes to the heat transfer intensification; an increase in pressure losses is not determined only by an increase in viscosity of the base fluid at the presence of nanoparticles in it.

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