

Experimental study of turbulent forced convection of nanofluid in channels with artificial vortex generators

A.V.Minakov^{1,2,3}, D.V.Guzei^{1,2}, K.N.Meshkov¹, I.A.Popov³, A.V.Shchelchkov³

¹ *Siberian Federal University, Krasnoyarsk*

² *Kutateladze Institute of Thermophysics, SB RAS, Novosibirsk*

³ *Kazan National Research Technical University named after A.N.Tupolev,
KNRTU-KAI, Kazan*

E-mail: Tov-andrey@yandex.ru

Abstract

Experimental study of forced turbulent convection of water-based nanofluids with nanoparticles of zirconium oxide (ZrO_2) was carried out in smooth tubes and channels with wall heat transfer enhancers. Nanopowders with average particle size of 44 and 105 nm were used in the experiments. The Reynolds number ranged from 3000 to 8000. It is revealed that the increments in the heat transfer coefficient and the pressure drop when using nanofluids depend on the surface shape of the channel. It is shown that nanofluids allow reaching thermal-hydraulic efficiency comparable to that of the channels with artificial heat transfer enhancers.

Keywords: *nanofluids, turbulent heat transfer, forced convection, thermal conductivity, viscosity, pressure drop, nanoparticle size, channels with wall heat transfer enhancers.*

Introduction

One of the most common ways of heat transfer enhancement in turbulent convection is the use of surfaces with artificial roughness. Roughness structure may be both an inherent part of heat-exchange surface or a wire element or part of other inserts. In the first case this is uniform or discrete two- or three-dimensional gutters or protrusions applied by mechanical treatment. Therefore, the comparison of thermal-hydraulic efficiency of the artificial roughness, as well as the use of nanofluids for heat transfer enhancement and also the possibility of their application in channels with artificial roughness is an important issue.

The problem of convective heat transfer enhancement and the related issues of experimental and theoretical studies are currently becoming an independent and fast growing field in heat exchange doctrine. All machines, equipment and technologies need for intensive heat removal that can be carried out using various kinds of heat exchangers. The urgency of this problem is determined by the desire to achieve maximum compactness with minimum materials

consumption, and increase heat exchange performance factor combined with reducing energy costs.

One way of solving heat transfer enhancement problem is the use in heat-exchange equipment of shaped heat-transfer surfaces such as annular knurling, spherical protrusions, etc. [1, 2].

In [3-5] it was shown that heat transfer enhancement in tubes with annular knurling within the range of Reynolds numbers $600 \dots 3.8 \cdot 10^3$ may reach the factor of 1.06... 14.01. At that, the pressure loss may increase by 0.92...19.7 times. In [6] at $Re_D = 10^4 \dots 4 \cdot 10^5$ and $Pr = 0.7 \dots 50$ within a wide range of dimensionless geometric parameters $d/D = 0.9 \dots 0.87$, $t/D = 0.25 \dots 1$ heat transfer in such channels was enhanced by 1.2÷2.2 times at the growth of the pressure loss by 1.05÷10.5 times.

In order to improve thermal-hydraulic performance of heat exchanger tubes, in [7-13] it is proposed to use spherical protrusions located along circular or spiral lines inside the tubes, rather than solid annular knurling. These studies were conducted within the following ranges of Re and Pr numbers $5 \cdot 10^3 < Re_D < 10^5$ and $Pr = 2.9 - 100$, relative parameters of the protrusions $0.017 < h/D < 0.16$; $0.09 < t/D < 1.706$ and $0.16 < s/D < 0.55$. The increase in heat transfer coefficients was up to 1.5-5 times as compared with a smooth tube. At that, pressure loss increased by 1.2-4.5 times.

At the same time, studies related to the use in heat exchange devices of fluids with admixtures of nanoparticles of different composition called "nanofluids", are developing extremely fast. Experiments on laminar and turbulent forced convection of water-based nanofluids have shown that nanofluids can improve the heat transfer coefficient within the range from a few percent up to 350% for carbon nanotubes [14-25]. A huge number of works appeared in this area over the last two decades. Most studies revealed the increase in heat transfer when using nanoparticles. However, there are publications demonstrating the reduction of heat transfer when adding nanoparticles.

The research carried out by Pak and Cho [21] apparently should be considered the first work dealing with turbulent heat transfer of nanofluids. The results of this work have shown that the Nusselt number in nanofluids increases with increasing volume concentration of particles and the Reynolds number. However, in this work it was also shown that at high concentrations of nanoparticles the heat transfer coefficient may be lower than that of pure water (by 12% for three-percent nanofluid).

The results of the experiments [22] showed that the addition of nanoparticles into the coolant significantly increases heat transfer efficiency (by 60% for two-percent nanofluid), while the friction factor was almost the same as for pure water.

In [23] it was found that the heat transfer coefficient definitely increases with increase in the concentration of nanoparticles in laminar and turbulent regimes at fixed Reynolds number. The maximum rate of enhancement of the heat transfer coefficient, recorded in the experiment, was 40% for 1.1% nanofluid. At that, the pressure drop in the channels for nanofluids was very close to that for pure fluid. In addition, the authors investigated the effect of particle size. No effect of particle size on heat transfer was found in this work. Duangthongsuk and Wongwises [24] obtained 32% improvement of heat transfer at concentration of nanoparticles equal to just 1%, and 14% decrease in heat transfer coefficient at concentration of nanoparticles equal to 2% as compared to the base fluid.

Fotukian and Hasr Esfahany [25] recorded a definite increase in heat transfer coefficient and pressure drop with increasing particle concentration. The maximum increase in heat transfer coefficient amounted to 48% at a negligible volume fraction of the nanoparticles equal to 0.054%. In the next paper [26], the same authors investigated turbulent heat transfer of water-based nanofluid with particles of CuO in a circular pipe. The authors found that the heat transfer coefficient is almost independent of nanoparticles concentration.

The work [31] allowed answering partly some questions regarding turbulent heat transfer of nanofluids. It is shown that heat transfer enhancement due to the use of nanofluids in the turbulent regime is not a trivial task. The effect which is beneficial for heat transfer enhancement depends on the ratio between viscosity and thermal conductivity of nanofluid, and therefore the material of the particles and their size. It is shown that with increasing concentration of nanoparticles the local and average heat transfer coefficients at a fixed Reynolds number increase. Heat transfer coefficient may decrease with the increase of particles concentration at a constant flow rate of the coolant. When conducting investigations with nanofluid and their analysis, it is necessary to take into account and control many factors and parameters. Apparently, this fact explains extremely broad variations and inconsistency of data on turbulent heat transfer obtained by various authors.

From this brief review it follows that currently there is still no conclusive understanding of turbulent heat transfer of nanofluids. The results of various studies of forced convection of nanofluids are fragmented and largely inconsistent. Most of the works do not discuss in principle the issue of thermal-hydraulic efficiency of nanofluids, though in the meantime this problem is crucial from the viewpoint of the practical application.

As concerns the studies of turbulent convection in nanofluids flowing in channels with shaped surfaces, they are very few, and usually limited just by measurements of the heat transfer coefficient. Thus, for example, in the work of Liu [33] it is shown that the nanofluid with 2% volume concentration of ZnO particles enhances heat transfer by 33% relative to that in pure

water at a constant Reynolds number. Suresh [34] has shown that the use of nanofluids based on distilled water with CuO particles allows enhancing heat transfer coefficient in a channel with dimples. At the 0.3% volume concentration of nanoparticles this enhancement reaches 39% relative to that in pure water.

Currently, no comprehensive analysis of thermal-hydraulic efficiency when using nanofluids in channels with shaped surface is available. Also, there are virtually no data allowing direct comparison of the thermal-hydraulic efficiency of nanofluids with that of traditional methods of heat transfer enhancement. Exactly this point is the objective of the present study.

1. Description of the setup and experimental technique

The diagram of the setup for conducting research on forced convection is shown in Fig. 1. The setup is a closed loop with a circulating coolant. Working fluid is pumped through a heated measuring area from where it flows to the heat exchanger, where it is cooled by the thermostat. The flow rate of the working fluid in the loop is adjusted by changing the pump power. Pump output is regulated by laboratory transformer. Power supplied to the pump is measured by Omix meter.

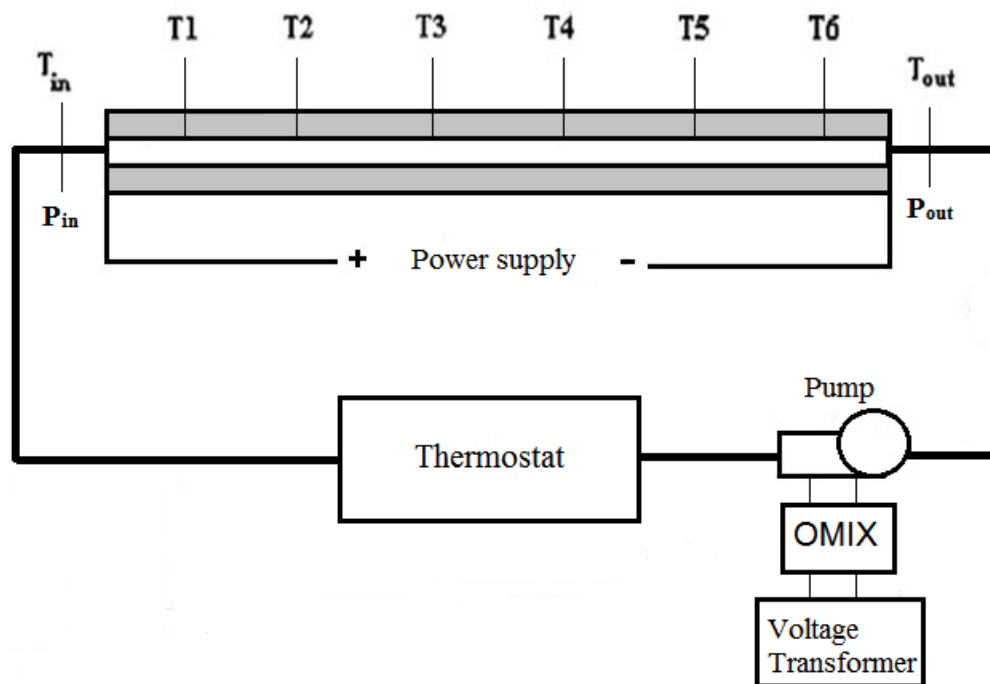


Fig.1. Scheme of the experimental setup.

Heated section is a stainless steel tube 6 mm in diameter and 1 m long. Thickness of the tube wall is 0.5 mm. The tube is heated by supplying electric current directly to the tube wall. This heating method allows providing heat at a constant heat flux density at the tube wall. In

addition, this heating method is universal and easily applicable to tubes of any cross section. The tube is insulated by multi-layered insulation. Heating power is controlled by a transformer. Six chromel-copel thermocouples are fixed at the tube wall at an equal distance from each other to measure the local temperatures of the tube. Temperature measurements were carried out by TRM-200 meters. In addition, temperatures at the inlet and outlet of the heated section were measured by means of thermocouples. At that, the thermocouple designed to measure the outlet temperature was located at a considerable distance from the end of the heated section to ensure temperature uniformity in the measurement point. The part of the loop between the heater and the cross section, in which the flow temperature was measured, was also insulated. Measurements of pressure drop were conducted using a differential pressure meter OWEN PD200.

Designed setup was tested based on known empirical data for heat transfer in pure water. A series of experiments was carried out for turbulent flow regime. The water flow rate was varied within the range from 0.65 to 2 l/min that corresponds to the range of Reynolds numbers from 2300 to 8000.

Figure 2a shows the comparison between the experimental dependence of the average Nusselt number on the Reynolds number and the well-known empirical correlation $Nu = 0.021 \cdot Re^{0.8} Pr^{0.43}$ [30]. Experimental Nusselt number was determined by the formula $Nu = \alpha d / \lambda$, where $\alpha = GC_p(T_i - T_o)S^{-1}(T_w - \bar{T})^{-1}$ – is the average heat transfer coefficient; C_p – is the heat capacity of the coolant; S – is the lateral surface area of the channel; T_o and T_i – are the outlet and inlet temperatures of the fluid; $\bar{T} = (T_i + T_o)/2$ – is the average temperature of the fluid; T_w – is the arithmetic mean of the channel wall temperature, obtained by averaging the indications of six thermocouples; Pr – is the Prandtl number, $Re = \rho U d / \mu$ – is the Reynolds number; λ and μ – are the thermal conductivity and viscosity coefficients of fluid, U – is the superficial velocity; d – is the tube diameter. As is obvious from the plot presented in Fig. 2a, the experimental data obtained in the turbulent flow regime are in a good agreement with the empirical correlation [30]. Disarrangement does not exceed 5% that is comparable to the accuracy of the correlation.

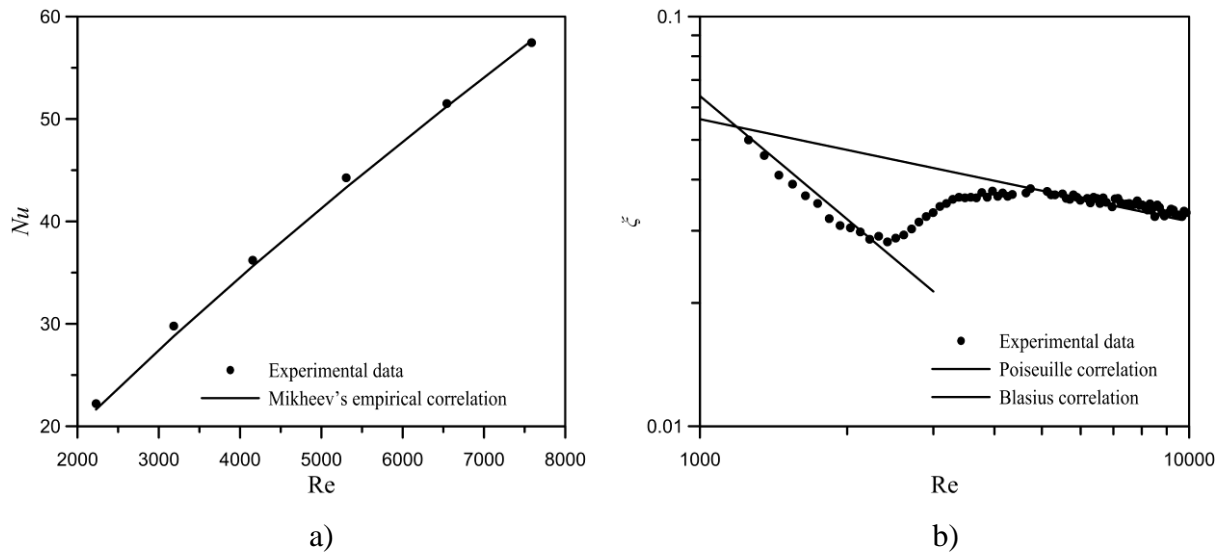


Fig.2. The Nusselt number (a) and friction factor (b) for pure water versus the Reynolds number.

In addition to the heat transfer experiments for pure water we have also measured pressure drop. Figure 2b shows the measured dependence of friction factor on the Reynolds number for pure water. The friction factor was calculated as follows: $\xi = \frac{2d \cdot \Delta P}{\rho U^2 L}$ where U – is the superficial velocity; d – is the diameter; L – is the length of the measuring section; ΔP – is the measured pressure drop. For comparison, we also showed the theoretical dependence of Poiseuille $\xi = \frac{64}{Re}$ for laminar flow and Blasius correlation $\xi = 0.316 Re^{-0.25}$ for turbulent flow. It is clear, that laminar-turbulent transition regime is observed within the range of Reynolds numbers from 2300 to 3000. The measured values of pressure drop agree with theoretical calculations with an accuracy of 5%.

Further we investigated turbulent forced convection of nanofluids in circular pipes with artificial vortex generators.

Studies were carried out with the use of the following types of tubes:

- 1) smooth tube with inside diameter of 0.01 m and the wall thickness of 0.001 m;
- 2) tube with annular knurling (see Fig.3) with the inner diameter $D=0.01$ m; height of the protrusions $-h=0.00045$ m; the diameter over the tops of the protrusions $d=0.0095$ m; relative height of the protrusions $d/D = 0.95$; the longitudinal pitch of the protrusions $t=0.01$ m;

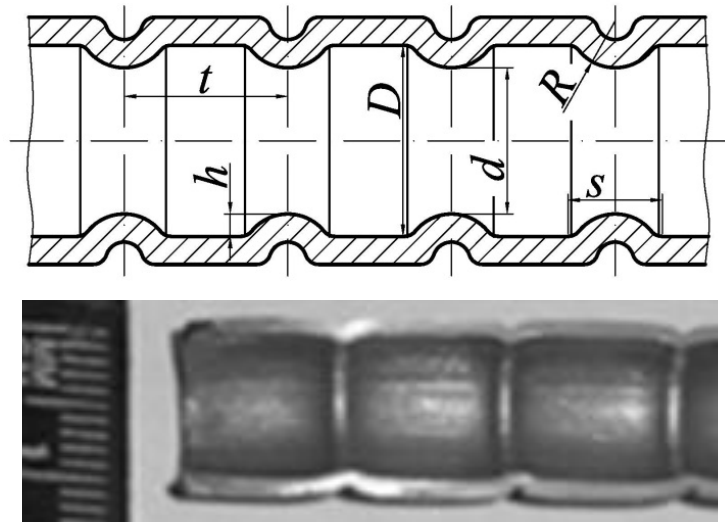


Fig.3. A longitudinal section of the heat exchanger tube with annular knurling.

3) the tube with spherical protrusions (see Fig. 4) with the inner diameter of the tube matrix $D=0.01$ m; protrusions height $h=0.0005$ m; diameter over the tops of the protrusions $d=0.009$ m; relative height of the protrusions $d/D = 0.9$; the longitudinal pitch of the protrusions $t=0.005$ m; transverse pitch of the protrusions $s=0.005$ m.

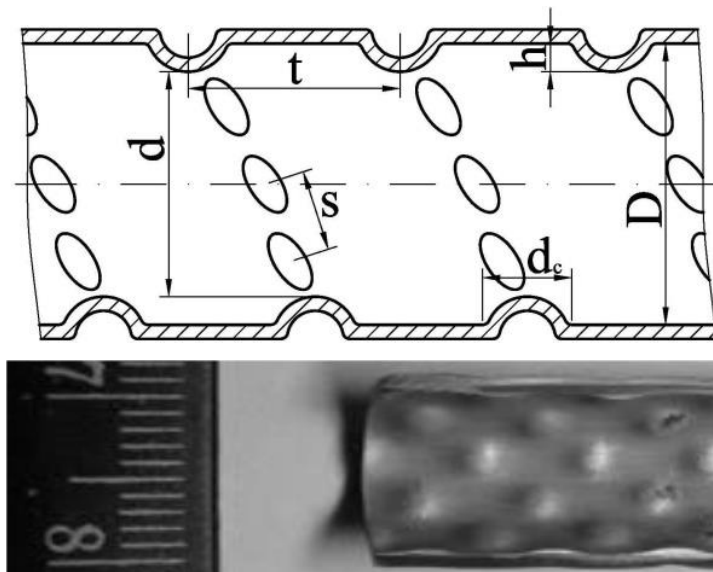


Fig.4. The scheme of spherical protrusions on the heat exchanger tube.

The nanofluid was prepared based on distilled water and zirconium oxide (ZrO_2) particles by standard two-step method. Required amount of nanoparticles were added to the base fluid and then mechanically stirred. The resulting suspension was subjected to a half-hour ultrasonic treatment to destruct the conglomerates of particles and obtain a uniform concentration. Zirconium oxide nanoparticles were purchased from JSC "Plazmotherm", Moscow. Zirconium

oxide nanopowder represents individual particles predominantly of spherical shape. Its phase composition is a mixture of monoclinic and tetragonal phases. Chemical composition (% wt.) includes ZrO_2 - 99.5%; Cl_2 <0.2%; and metal impurities <0.3%.

Nanopowders with average particle size of 44 and 105 nm were used in the experiments. Measuring the nanoparticles distribution by size directly in the fluid was carried out using the CPS Disk Centrifuge DC24000 device, while powder was controlled by means of electron microscopy (Fig. 5). Volume concentration of nanoparticles in all experiments was equal to 4 volume percent. The nanofluid was free from surfactants.

Conductivity and viscosity coefficients of nanofluids were measured in advance. The viscosity of nanofluids was measured using a rotational viscometer DV2T. Viscosity measurements were carried out within the range of shear rates from 10 to 200 1/s at a temperature of 25⁰C. The thermal conductivity coefficient of nanofluid was measured by nonstationary hot wire method. Detailed description of the setup and its testing is given in [35].

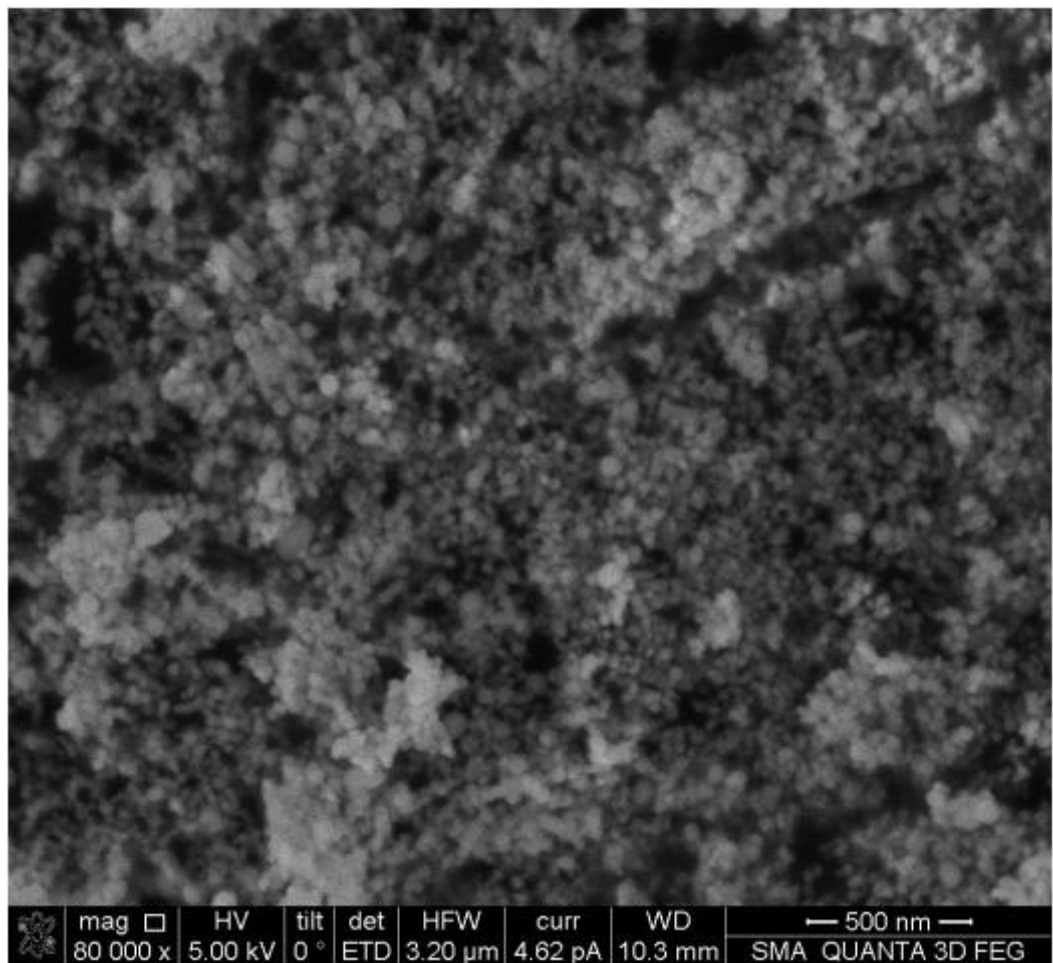


Fig.5. Micrograph of the zirconium oxide nanopowder.

Experimentally determined dependences of the relative viscosity and relative thermal conductivity coefficients of the nanofluid depending on particle size are shown in Figs.6a and 6b. As is obvious, the dependencies are oppositely directed. The viscosity of nanofluid decreases with increase in the average size of the particles, while the thermal conductivity coefficient, on the contrary, increases. This well agrees with present-day ideas about the behavior of the transfer coefficients of nanofluids [31-34].

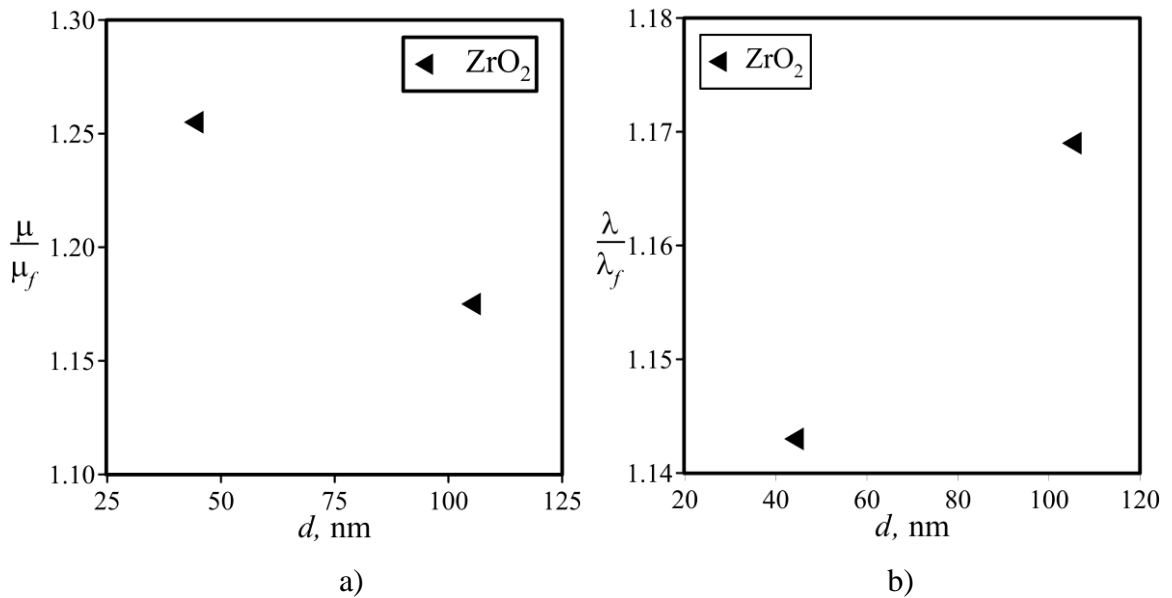


Fig.6. Transfer coefficients of the nanofluid with ZrO₂ nanoparticles:
a) relative viscosity coefficient versus particle size;
b) relative thermal conductivity coefficient versus particle size.

2. The study of the heat transfer coefficient and pressure drop for pure water in different channels

At the beginning we have conducted a series of measurements of the heat transfer coefficient and pressure losses in shaped channels for pure water. The results of the measurements are shown in Figs. 7-8. Measurements of heat transfer coefficient in channels with enhancers showed that the channels with an annular knurling on the walls allow achieving the enhancement of the average heat transfer coefficient by 85% relative to that in the smooth tube at constant Reynolds number. The average heat transfer coefficient in channels with spherical protrusions is by 30% higher than that in the smooth tube at constant Reynolds number (Fig. 7).

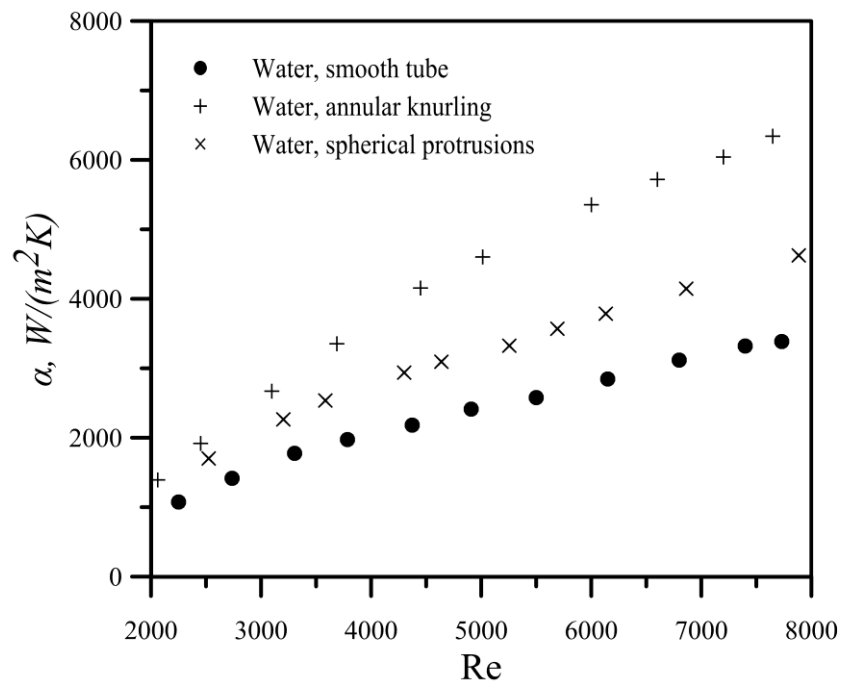


Fig.7. The average heat transfer coefficient of pure water Versus the Reynolds number in different channels.

The presence of heat transfer enhancers inevitably leads to an increase in the pressure drop required for pumping the fluid. Thus, in channels with an annular knurling, pressure drop increases by 1.8 times compared to smooth tube at the same Reynolds number. Channels with spherical protrusions increase the pressure drop by 1.4 times compared with the smooth tube at the same Reynolds number. The dependence of pressure drop on the Reynolds number is presented in Fig. 8.

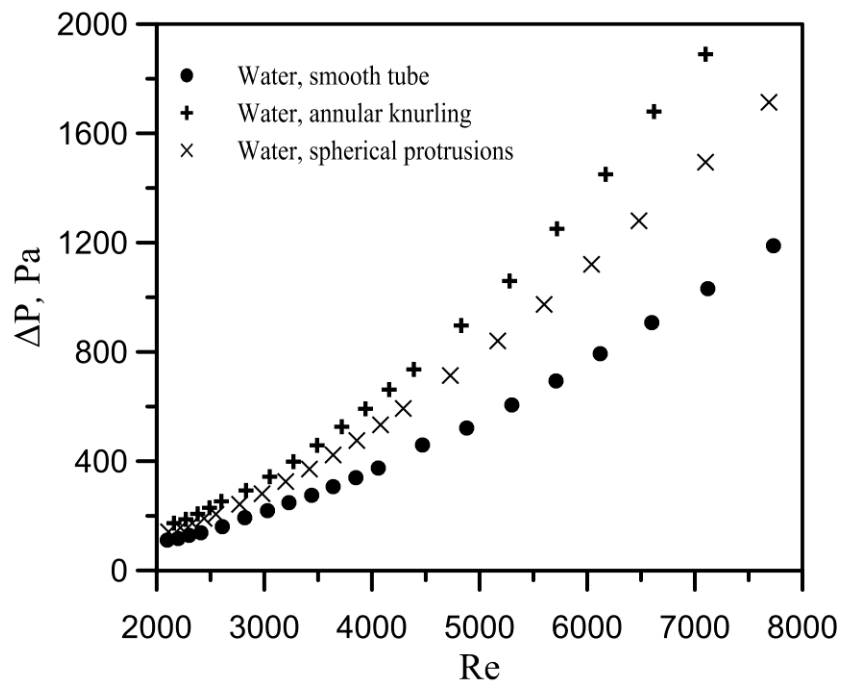


Fig.8. Pressure drop versus the Reynolds number in different channels.

3. The study of the heat transfer coefficient and pressure drop for nanofluids in different channels

Further experiments were carried out with the nanofluid. The first series of experiments was conducted on a smooth tube. First, the experimental setup was tested using pure water. Figure 9 shows experimental dependences of the average heat transfer coefficient versus the Reynolds number for pure water and nanofluids with an average size of ZrO_2 particles equal to 44 and 105 nm, as well as Mikheev's empirical correlation for pure water.

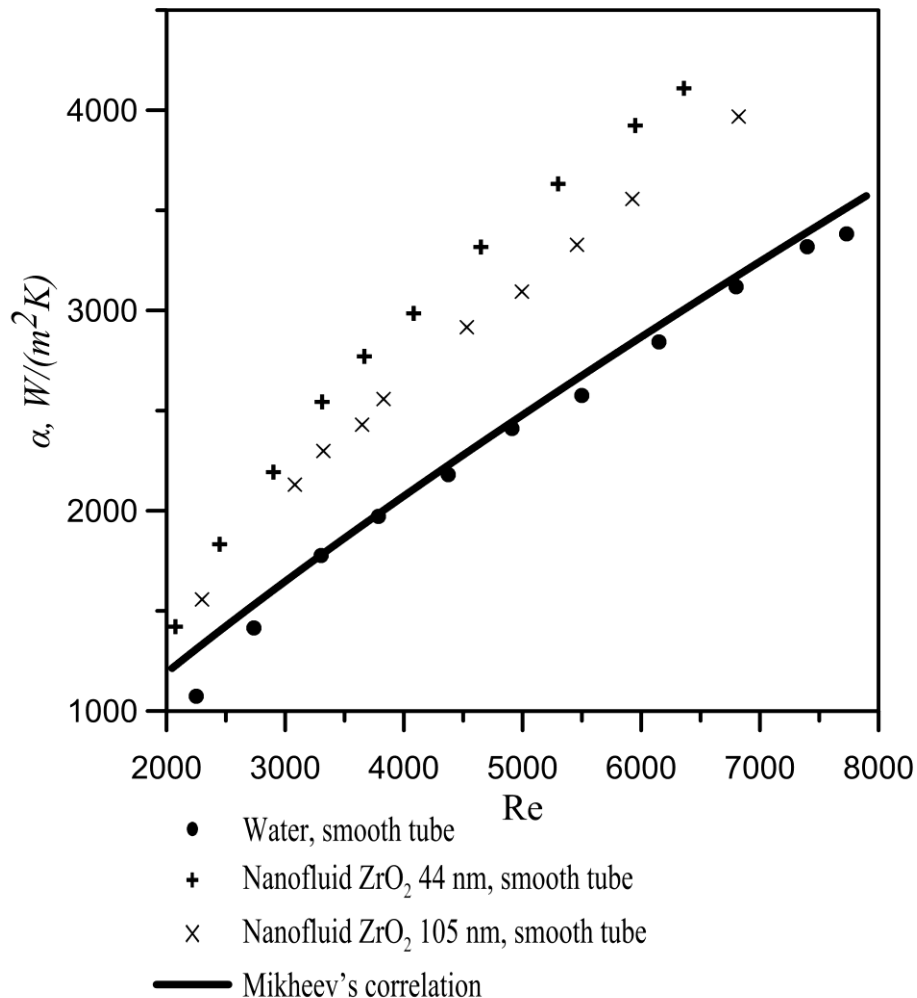


Fig.9. The average heat transfer coefficient in the smooth tube versus the Reynolds number for water and nanofluids with different sizes of particles.

The plot shows that the experimental data for pure water are in good agreement with the empirical correlation of Mikheev, with an accuracy of 3%. The average heat transfer coefficient depends on the particle size. Thus, the nanofluid with ZrO_2 particles of 44 nm allows enhancing heat transfer by 40% compared with that for pure water at a fixed Reynolds number. For nanofluids with ZrO_2 particles of 105 nm in size, the enhancement reaches 27% compared to that for pure water. With increasing particle size at a fixed Reynolds number the average heat transfer coefficient of nanofluids is reduced. The reason for this behavior lies in the dependency of the

transfer coefficients of nanofluid on particle size. According to the Mikheev's formula, at a fixed Reynolds number, the heat transfer coefficient is proportional to the complex $\mu^{0.43}\lambda^{0.57}$. In this case, with increasing particle size, we have two competing trends - the viscosity decreases while the conductivity increases. Experimentally determined dependences for relative viscosity and thermal conductivity coefficients versus particle size are shown in Figs.6a and 6b, respectively.

Figure 10 demonstrates pressure drop versus the Reynolds number in the smooth tube for pure water and nanofluids with different particle size.

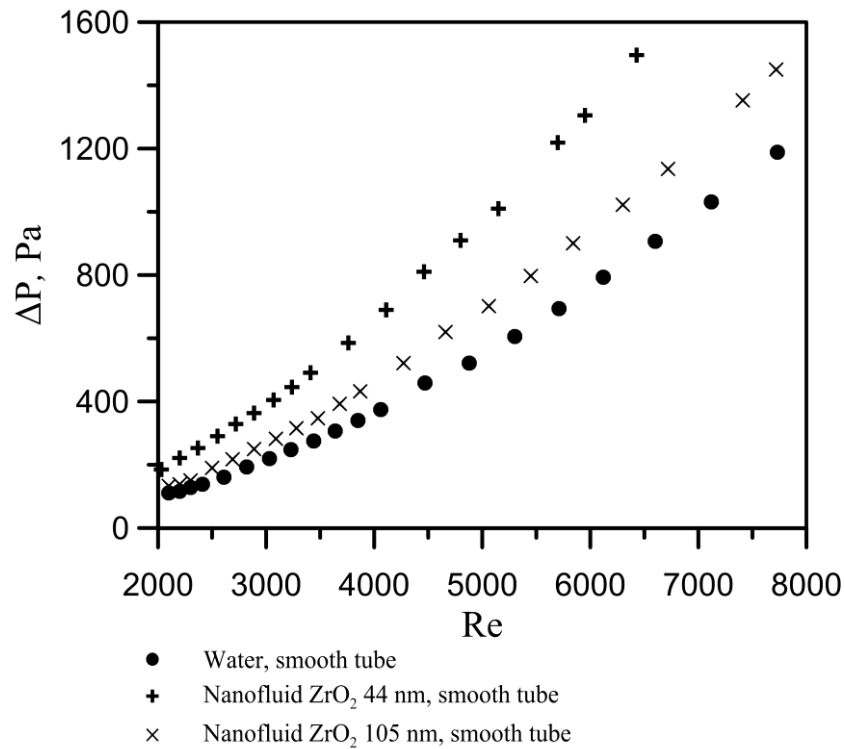


Fig.10. Pressure drop in the smooth tube versus the Reynolds number for pure water and nanofluids with different particle sizes.

As is obvious, with increasing size of nanoparticles the pressure drop at a fixed Reynolds number decreases. This is due to the lower viscosity (see Fig. 6a). The nanofluid with ZrO₂ particles size of 44 nm increases the pressure drop required for pumping the fluid by 1.8 times as compared to pure water. The nanofluid with ZrO₂ particles of 105 nm increases the pressure drop by 1.25 times.

The next series of experiments with nanofluids was conducted for tubes with annular knurling. The experiments have shown that with increasing of ZrO₂ particle size at a fixed Reynolds number, the average heat transfer coefficient of nanofluids is reduced similarly as in the smooth tube.

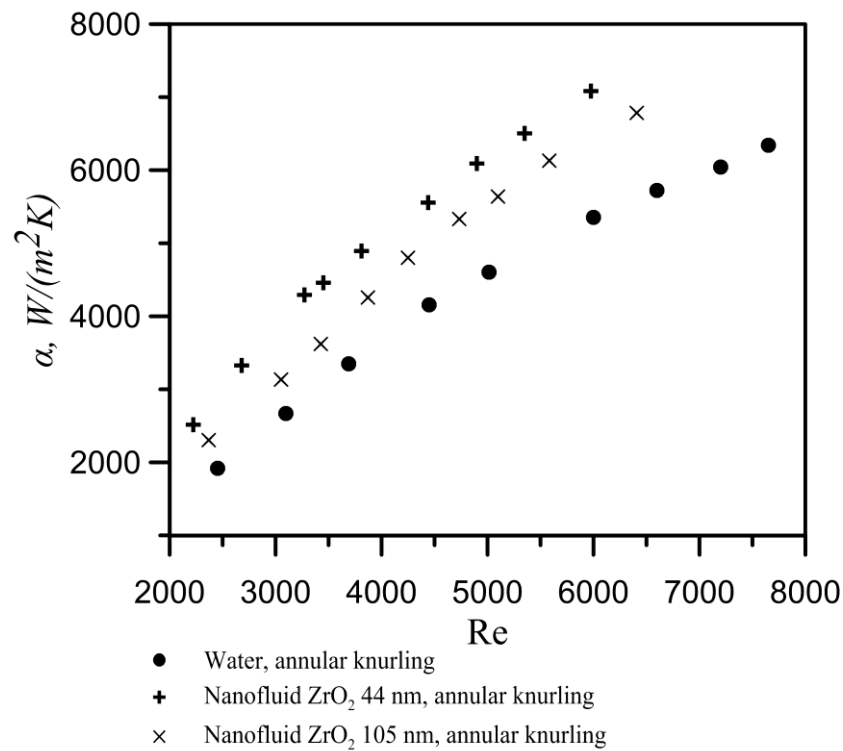


Fig.11. The average heat transfer coefficient in the tube with annular knurling versus the Reynolds number for pure water and nanofluids with different particle sizes.

As is seen from Fig. 11, the heat transfer coefficient at a fixed Reynolds number for nanofluid with particle size of 44 nm is by 37% higher than that for pure water in the tube with an annular knurling. The nanofluid with particle size of 105 nm enhances heat transfer by 23% relative to pure water.

Figure 12 shows the dependence of the pressure drop on the Reynolds number for nanofluids in channels with annular knurling. The pressure drop required to pump the nanofluid in a channel with annular knurling has increased by 1.85 times for ZrO_2 particles with a size of 44 nm and by 1.3 times for particles with a size of 105 nm relative to pure water in the channel with annular knurling.

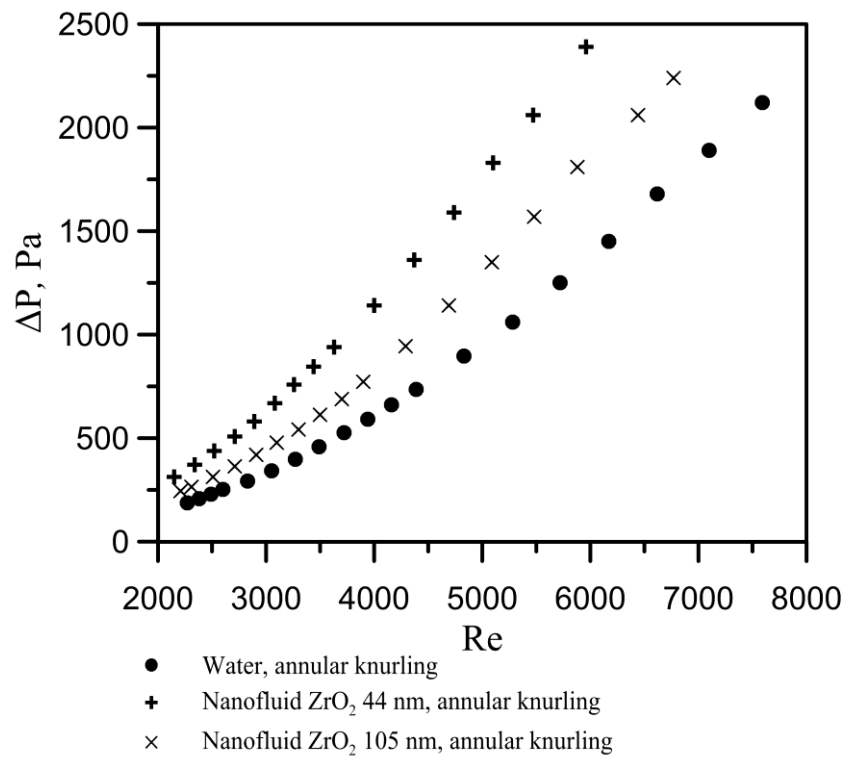


Fig.12. Pressure drop in the tube with annular knurling versus the Reynolds number for water and nanofluids with different particle sizes.

Experiments carried out with nanofluids in the tube with spherical protrusions showed that the heat transfer coefficient for the nanofluid with ZrO₂ particle size of 44 nm is by 35% higher than that for pure water. The nanofluid with ZrO₂ particle size of 105 nm enhances heat transfer by 20% relative to pure water flowing in the tube with spherical protrusions (Fig.13).

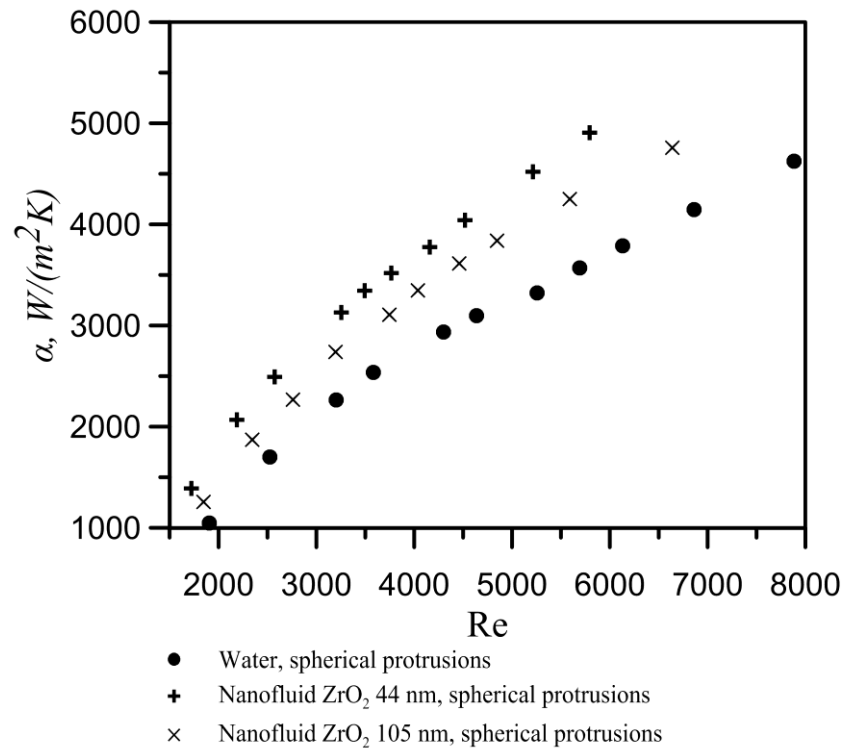


Fig.13. The average heat transfer coefficient in a tube with spherical protrusions versus the Reynolds number for water and nanofluids with different particle sizes.

The pressure drop required to pump the nanofluid in a channel with spherical protrusions has increased by 2 times for ZrO_2 particles with a size of 44 nm and by 1.35 times for particles with a size of 105 nm relative to pure water flowing in a channel with spherical protrusions (Fig.14).

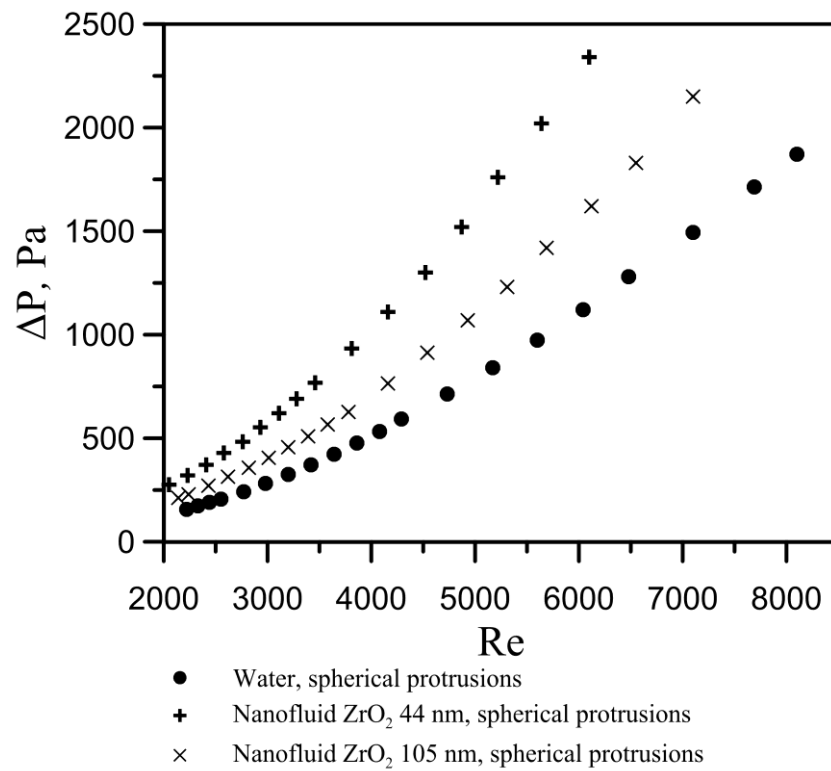


Fig.14. Pressure drop in a tube with spherical protrusions versus the Reynolds number for water and nanofluids with different particle sizes.

Tables 1 and 2 represent the increments of the heat transfer coefficient and pressure drop, averaged within the entire range of the Reynolds numbers, for nanofluids and pure water flowing in various channels.

Table 1. The increment of the heat transfer coefficient and pressure drop for nanofluid with ZrO₂ particles with a size of 44 nm relative to pure water.

ZrO ₂ 44 nm	$\frac{\alpha}{\alpha_B}$	$\frac{\Delta P}{\Delta P_B}$
Smooth tube	1.4	1.8
Annular knurling	1.35	1.85
Spherical protrusions	1.35	1.95

Table 2. The increment of the heat transfer coefficient and pressure drop for nanofluid with ZrO₂ particles with a size of 105 nm relative to pure water.

ZrO ₂ 105 nm	$\frac{\alpha}{\alpha_B}$	$\frac{\Delta P}{\Delta P_B}$
Smooth tube	1.27	1.25
Annular knurling	1.23	1.3
Spherical protrusions	1.2	1.35

It is obvious that in the channels with shaped surfaces, the increment in heat transfer coefficient due to the use of nanofluids is slightly lower than that in the smooth channel, while the increment of the pressure drop, on the contrary, is higher. This trend remains for both particle sizes. Thus, it was shown that heat transfer enhancement when using nanofluids and the increment of the pressure drop depend on the surface shape of the channel. As it follows from Tables 1 and 2, the use of nanofluid is most effective for smooth channels. The difference in increments of the heat transfer coefficient and pressure drop for pure water and nanofluid in channels with different surface shape, in our opinion, can be caused by local heterogeneity in the nanoparticles concentration. In [36, 37] it is shown that experimental data on heat transfer of nanofluids in smooth circular channels are well described by the approximation of a homogeneous mixture. The homogeneous nanofluid model assumes a constant concentration of nanoparticles all over the channel. In channels with discrete roughness, this approach may not work. Annular knurling and spherical protrusions play the role of local generators of vortices, in which nanoparticles can separate changing their local concentration. Local change of the particle concentration changes locally thermophysical properties of the nanofluid that in our opinion leads to the difference in the increments of the heat transfer coefficient and pressure drop in the shaped channels. In the future we plan to test this assumption through numerical simulations.

4. Analysis of thermal-hydraulic efficiency

From a practical point of view, for the analysis of thermal-hydraulic efficiency it is convenient to use the following value:

$$\eta = \frac{\frac{\alpha}{\alpha_B}}{\frac{\Delta P}{\Delta P_B}}$$

Let evaluate the heat transfer enhancement against the increase in the cost of pumping the working fluid through the channel. Figure 15 shows the dependence of thermal-hydraulic

efficiency on the Reynolds number for nanofluids with different size of particles flowing in smooth-walled tube. From Fig. 15 it is seen that the use of nanofluid with a large particle size is more advantageous, since an increase in particle size reduces the viscosity of the nanofluid and consequently decreases the cost of pumping.

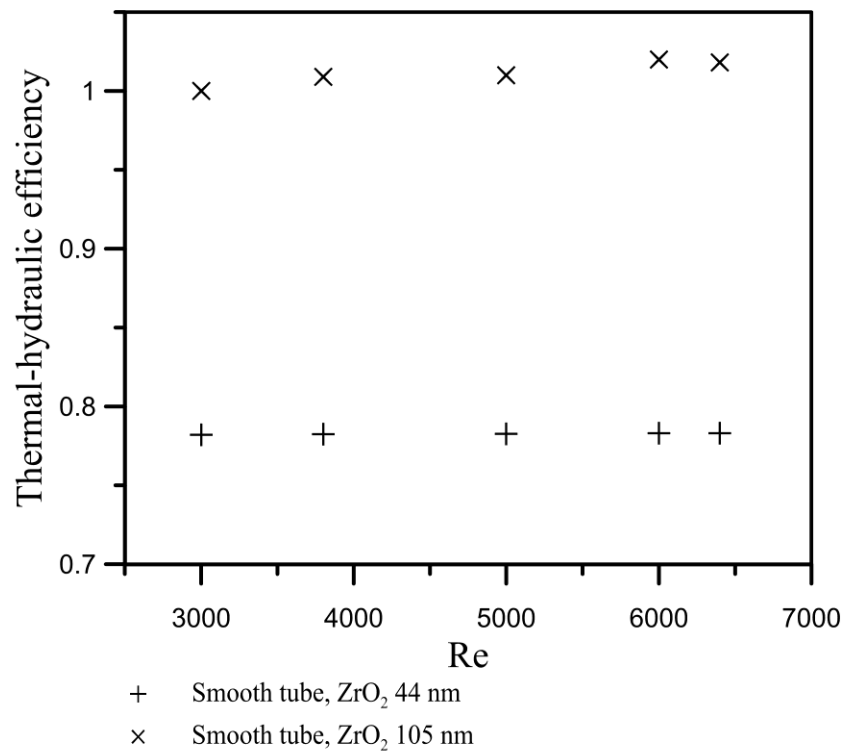


Fig. 15. The thermal-hydraulic efficiency versus the Reynolds number for nanofluids flowing in a smooth tube.

Compare dependences of thermal-hydraulic efficiency on the Reynolds number for pure water and nanofluids flowing in smooth tubes and channels with artificial heat transfer enhancers.

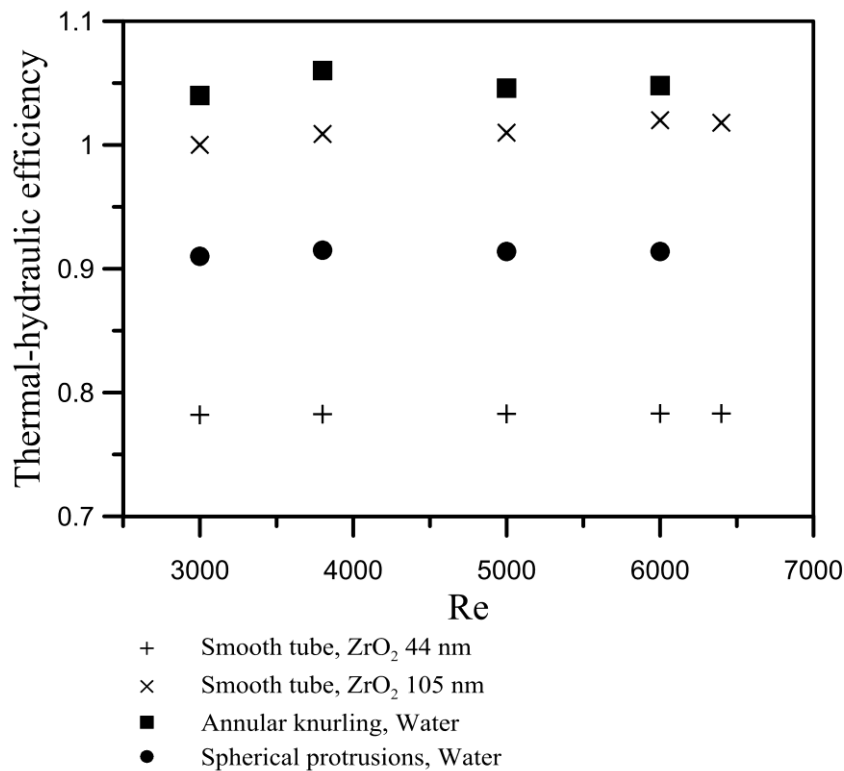


Fig. 16. The thermal-hydraulic efficiency versus the Reynolds number for nanofluids flowing in a smooth tube, and pure water flowing in channels with artificial roughness.

As is obvious from Fig. 16, at the same Reynolds number, the nanofluid with ZrO₂ particles 105 nm in size allows obtaining thermal-hydraulic efficiency comparable to that for pure water flowing in the channel with annular knurling. The thermal-hydraulic efficiency of the nanofluid with a particle size of 105 nm is by 11% higher than that for the channel with spherical protrusions.

Despite the fact that in terms of enhancement of heat transfer coefficients, the nanofluid with particle size of 44 nm is superior to the nanofluid with a particle size of 105 nm, the thermal-hydraulic efficiency is higher in nanofluids with larger particles (see Fig.16), since decrease in particle size leads to increase of viscosity of the nanofluid and, consequently, increase of the pressure drop required for pumping of the fluid. Consequently, the thermal-hydraulic efficiency decreases.

Thermo-hydraulic efficiency of the nanofluid with particles of 44 nm for a smooth tube is lower than thermal-hydraulic efficiency of the channels in artificial intensifiers of heat transfer with pure water. Thus, in channels with spherical protrusions thermal-hydraulic efficiency for pure water is higher by 16%, while in channels with annular knurling it is higher by 33%.

Thus, in this work, we were the first who conducted a direct comparison of the thermal-hydraulic efficiency of nanofluids flowing in the smooth channel with that for pure water flowing in the channels with traditional heat transfer enhancers, i.e. channels with shaped

surface. It is shown that choosing the relevant parameters of nanofluid (particles size, material, and concentration), we can achieve a thermal-hydraulic efficiency comparable with that for the best heat transfer enhancers.

Thus, the use of nanofluids as the coolant in the channels with heat transfer enhancers is low-efficient that is evident from the plot, presented in Fig. 17. Thermal-hydraulic efficiency of nanofluids flowing in the channels with shaped surface in all cases was lower than that in smooth channels. This is caused by the additional increase in already high pressure drop related to the viscosity of the nanofluid.

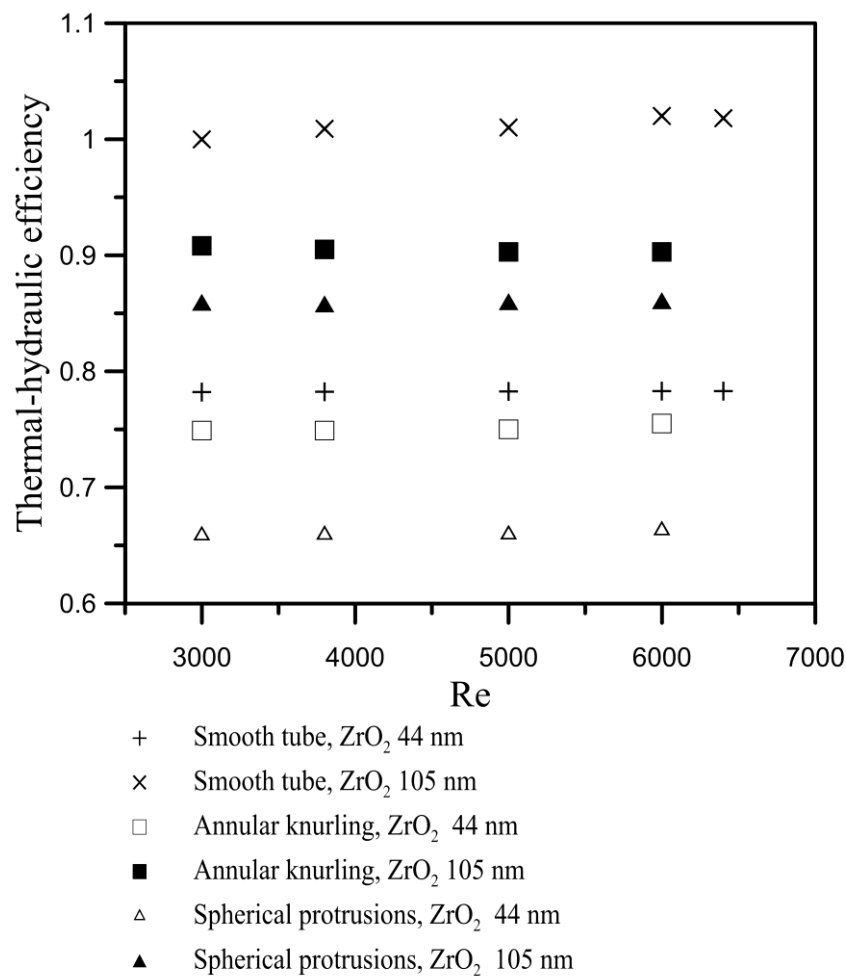


Fig.17. The thermal-hydraulic efficiency versus the Reynolds number for nanofluids flowing in different channels.

Conclusion

1. Experimental study of forced turbulent convection of water-based nanofluids with nanoparticles of zirconium oxide (ZrO₂) was carried out in smooth tubes and channels with wall heat transfer enhancers.

2. The experiments have shown that the nanofluid with ZrO_2 particles 44 nm in size enhances heat transfer by 35% as compared to heat transfer in pure water at a fixed Reynolds number. In the turbulent flow regime at a fixed Reynolds number, with increasing particle size, the rate of enhancement decreases, and for the nanofluid with particle size of 105 nm this decrease is 20%.

3. It is revealed that the increments in the heat transfer coefficient and the pressure drop when using nanofluids depend on the surface shape of the channel. When using tubes with heat transfer enhancers, the increment of the heat transfer coefficient in nanofluids decreases, while the increment of the pressure drop, on the contrary, increases. In our opinion, this is caused by a change in the local concentration of nanoparticles in the vortices formed behind the artificial roughness.

4. It is shown that nanofluids allow reaching thermal-hydraulic efficiency comparable to that of the channels with artificial heat transfer enhancers. The use of nanofluids in channels with artificial enhancers appeared to be ineffective in terms of thermal-hydraulic efficiency.

References

1. Popov I.A., Gortyshov Yu.F., Olimpiev V.V. Industrial applications of heat exchange enhancement: The modern state of the problem (a Review). *Thermal Engineering*. Volume 59, Issue 1, January 2012, pp. 1-12.

2. Popov I. A., Yakovlev A. B., Clicks A.V., Ryzhkov D. V., Obukhova, L. A., Advanced methods of heat transfer enhancement for heat energy equipment // *Power Engineering of Tatarstan*. 2011. No. 1. pp. 25-29.

3. Kalinin E.K., Dreitser G.A., Kopp I.Z. et al. Efficient heat exchange surfaces. M.: Energoatomizdat, 1998. 407 p.

4. Webb R.L., Eckert E.R.G., and Goldstein R.J. Heat exchange and friction in tubes with repeated-rib roughness, *Int. J. Heat Mass Exchange*, 1971. vol. 14, pp. 601–618.

5. Migay V.K. Improving the efficiency of modern heat exchangers. – L.: Energy. Leningr. otd-nie, 1980. – 144 p.

6. Popov I.A., Chshelchikov A.V., Yarkaev M.Z. Thermal-hydraulic characteristics of discrete-rough pipes in transient flow regimes // *Bulletin of Higher Educational Institutions. Aviation Equipment*. 2013. No. 1. Pp. 61-65.

7. Hwang S.D., Cho H.H. Heat exchange enhancement of internal passage using dimple/protrusion // *Annals of the Assembly for International Heat Exchange Conference 13*. – Sydney, Australia, 2006. – HTE, №24. – Pp. 10-17.

8. Kuwahara H., Takahashi K., Yanagida T., Nakayama W., Sugimoto S., Oizumi S. Heat exchange tube for single phase flow: Patent USA № 4690211. CIC B21C 37/20. 01.09.1987. Assignee Hitachi, Ltd.
9. Rabas T.J., Webb R.L., Thors P., Kim N.K. Influence of roughness shape and spacing on the performance of three-dimensional helically dimpled tubes // J. Enhanced Heat Exchange. – 1993. – No.1. – Pp. 53-64.
10. Vicente P.G., Garcia A., Viedma A. Heat exchange and pressure drop for low Reynolds turbulent flow in helically dimpled tubes // Int. J. Heat and Mass Exchange. – 2002. – Vol. 45. – Pp. 543-553.
11. Vicente P.G., Garcia A., Viedma A. Experimental study of mixed convection and pressure drop in helically dimpled tubes for laminar and transition flow // Int. J. of Heat and Mass Exchange. – 2002. – Vol. 45. – Pp. 5091-5105.
12. Gotovsky, M.A., Belenky M.Ya., Fokin, B.S., Heat transfer and resistance of flow in a circular pipe enhanced by the regular system of spherical dimples and spherical protrusions // Heat and mass transfer and hydrodynamics in swirling flows: Abstracts of the 2nd Russian Conf. – M.: Publishing MEI, 2005. – Pp. 49-50.
13. Chen J., Müller-Steinhagen H., Duffy G.G. Heat exchange enhancement in dimpled tubes// Applied Thermal Engineering. – 2001. – Vol. 21. – Pp. 535-547.
14. Ahuja A.S. Augmentation of heat transport in laminar flow of polystyrene suspensions. II. Analysis of the data // J. Appl. Phys. 1975. V. 46. P. 224.
15. Das S.K., Choi S.U.S., Patel H. Heat exchange in nanofluids. A Review // Heat Exchange Eng. 2006. V. 20. № 10. P. 3.
16. Das S.K., Choi S.U.S., Yu W., Pradeep T. Nanofluids science and technology. New Jersey: Wiley-Interscience, 2007. P. 397.
17. Wang X.-Q., Mujumbar A.S. Heat exchange characteristics of nanofluids: a Review // Int. J. Thermal Sci. 2007. V. 46. P. 1.
18. Yu W., France D.M., Choi S.U.S., Routbort J.L. Review and assessment of nanofluid technology for transportation and other application // Argonne National Laboratory, ANL/ESD/07-9. 2007. P. 78.
19. Terekhov V.I., Kalinin S.V., Lemanov, V.V., Heat transfer mechanism in nanofluids: Current state of the problem. Part 2. Convective heat transfer // Thermophysics and Aeromechanics. 2010. No. 2. P. 173.
20. A.V. Minakov, A. S. Lobasov, D.V. Guzei, M. I. Pryazhnikov, V. Ya. Rudyak. The experimental and theoretical study of laminar forced convection of nanofluids in the round channel. Applied Thermal Engineering (2014), doi: 10.1016/j.applthermaleng.2014.11.041.

21. B.C. Pak, Y. I Cho, Hydrodynamic and heat exchange study of dispersed fluids with submicron metallic oxide particles, *Exp. Heat Exchange* 11 (1998) 151.
22. QLi, Y. Xuan, Convective heat exchange and flow characteristics of Cu-water nanofluid, *Sci. China E* 45 (2002) 408.
23. Y. He, Y. Jin, H. Chen, Y. Ding, D. Cang, H. Lu, Heat exchange and flow behavior of aqueous suspensions of TiO₂ nanoparticles (nanofluids) flowing upward through a vertical pipe, *Int. J. Heat Mass Exchange*. 50. (2007) 2272.
24. W. Duangthongsuk and S. Wongwises. An experimental study on the heat exchange performance and pressure drop of TiO₂-water nanofluids flowing under a turbulent flow regime. *Int J Heat Mass Exchange*, vol. 53, p. 334–344, 2010.
25. S. Fotukian, M. Nasr Esfahany. Experimental investigation of turbulent convective heat exchange of Al₂O₃/water nanofluid inside a circular tube. *Int J Heat Fluid Fl*, vol. 31, p. 606–612, 2010.
26. S. Fotukian, M. Nasr Esfahany. Experimental study of turbulent convective heat exchange and pressure drop of dilute CuO/water nanofluid inside a circular tube. *Int Commun Heat Mass*, vol. 37, p. 214–219, 2010.
27. C.T. Nguyen, G. Roy, C. Gauthier, N. Galanis, Heat exchange enhancement using Al₂O₃-water nanofluid for electronic fluid cooling system, *Appl. Therm. Eng.* 28 (2007) 1501.
28. Arttu Meriläinen, Ari Seppälä, Kari Saari, Jani Seitsonen, Janne Ruokolainen, Sakari Puisto, Niko Rostedt, Tapio Ala-Nissila. Influence of particle size and shape on turbulent heat exchange characteristics and pressure losses in water-based nanofluids. *International Journal of Heat and Mass Exchange* 61 (2013) 439–448.
29. E. V. Timofeeva, Wenhua Yu, David M. France, Dileep Singh, and Jules L. Routbort. Base fluid and temperature effects on the heat exchange characteristics of SiC in ethylene glycol/H₂O and H₂O nanofluids. *Journal of applied physics* 109, 014914 (2011)
30. Tsvetkov, F.F., Grigoriev B.A. *Heat and mass transfer*. M.: Publishing MEI, 2005.
31. Faris Mohammed Ali, W. Mahmood Mat Yunus, Zainal Abidin Talib. Study of the effect of particles size and volume fraction concentration on the thermal conductivity and thermal diffusivity of Al₂O₃ nanofluids. *International Journal of Physical Sciences*.2013. Vol. 8(28), pp. 1442-1457.
32. Michael P. Beck, Yanhui Yuan, Pramod Warriar, Aryn S. Teja. The effect of particle size on the thermal conductivity of alumina nanofluids. *J Nanopart Res* (2009) 11:1129–1136.
33. Liu, K. (2014). Heat exchange enhancement by three-dimensional surface roughness technique in nuclear fuel rod bundles. (Master's thesis).

34. S. Suresh, M. Chandrasekar, S. Chandra Sekhar. Experimental studies on heat exchange and friction factor characteristics of CuO/water nanofluid under turbulent flow in a helically dimpled tube. 2011. *Experimental Thermal and Fluid Science* 35 p.542–549.

35. Minakov A.V., Rudyak V.Ya., Guzei D.V., Pryazhnikov M.I., Lobasov A.S. Measurement of the thermal-conductivity coefficient of nanofluids by the hot-wire method // *Journal of Engineering Physics and Thermophysics*. 2015. Vol. 88. No. 1. pp. 149-162.

36. Minakov A.V., Lobasov A. S., Guzei D.V., Pryazhnikov M. I., Rudyak V. Ya. The experimental and theoretical study of laminar forced convection of nanofluids in the round channel // *Applied Thermal Engineering*. 2015. Vol. 88. pp. 140-148.

37. Minakov A.V., Lobasov A.S., Pryazhnikov M.I., Guzei D.V. Experiment-calculated investigation of the forced convection of nanofluids using single fluid approach // *Defect and Diffusion Forum*. 2014. Vol. 348. pp. 123-138.