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NUMERICAL AND EXPERIMENTAL STUDY OF HYDRAULIC RESISTANCE OF TUBES WITH INTERNAL HELICAL FINNING BY DEFORMING CUTTING

An object of investigation is a tube having helical fins on the internal surface and different geometric sizes. Investigation methods: experiments to obtain quantitative results hydraulic resistance of tubes with internal helical finning and to verify computational algorithm; numerical simulation to visualize the flow structure in the tube. Studies of hydraulic resistance of tubes with internal helical finning over a wide range of operating and geometric parameters were made: $\mathrm{Re}_D=2\cdot10^3...2.5\cdot10^5$, under the variation of the angle of swirling α = 14–87°, the relative height of a protrusion h/d = (25–87,5)$\cdot10^{-3}$, the relative axial pitch p/d = 0.16–12.73. It is revealed that the hydraulic resistance of tubes with helical finning increases by a factor of 1.1 to 11.7. The numerical simulation results showed that, as the angle of helical swirling is increased, in the near-wall layers the share of the circumferential velocity component increases and the share of the longitudinal component decreases. And since the finning height exceeds the boundary layer thickness, the hydraulic resistance grows.

Keywords: turbulence, hydraulic resistance, experiment, numerical simulation, tube, deforming cutting

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ЧИСЛЕННОЕ И ЭКСПЕРИМЕНТАЛЬНОЕ ИССЛЕДОВАНИЕ ГИДРАВЛИЧЕСКОГО СОПРОТИВЛЕНИЯ ТРУБ С ВНУТРЕННИМ СПИРАЛЬНЫМ ОРЕБРЕНИЕМ, НАНЕСЕННЫМ МЕТОДОМ ДЕФОРМИРУЮЩЕГО РЕЗАНИЯ

В данной работе объектом исследования являются трубы с внутренним спиральным оребрением с различными геометрическими размерами. Методы исследования: эксперимент – для получения количественных результатов по гидравлическому сопротивлению труб с внутренним спиральным оребрением и для верификации расчетного алгоритма; численное моделирование – для визуализации структуры течения в трубе. Изучено гидравлическое сопротивления труб с внутренним спиральным оребрением в широком диапазоне режимных и конструктивных параметров: $\mathrm{Re}_D = 2\cdot10^3...2{,}5\cdot10^5$, при изменении угла закрутки α = 14–87°, относительной высоты выступов h/d = (25–87,5)$\cdot10^{-3}$, относительного шага по оси p/d = 0,16–12,73. Выявлено увеличение гидравлического сопротивления труб с внутренним спиральным оребрением от 1,1 до 11,7 раза. Результаты проведенного численного моделирования показали, что с увеличением угла спиральной закрутки в пристеночных слоях доля окружной составляющей скорости увеличивается, а доля продольной составляющей скорости уменьшается. Так как высота оребрения превышает толщину пограничного слоя, то в результате растет гидравлическое сопротивление.

Ключевые слова: турбулентность, гидравлическое сопротивление, эксперимент, численное моделирование, труба, метод деформирующего резания.

In forced heat carrier flow in the tube from the tube inlet, dynamic boundary layers start forming at the walls; their thickness gradually grows with increasing the distance from the tube inlet. At a time, thermal boundary layers are formed, which hinder heat transfer between the tube and heat carrier. The thickness of temperature and dynamic layers is related as $\delta_{\mathrm{temp}}/\delta_{\mathrm{dyn}} \sim \mathrm{Pr}^{-0.5}$. At some distance from the tube inlet boundary layers merge and flow becomes stabilized. To enhance heat transfer between the tube walls and the heat carrier, to perturb the boundary layer, or to decrease its thickness, or to control the boundary layer separation.

Heat transfer enhancement techniques can be divided into three types. The first type is concerned with active techniques requiring external power supply (induced vibration, acoustic action, boundary layer scraping), the second type – passive methods not requiring external power supply, third type – combined methods assuming the use of two or more active /passive methods.

Many of heat transfer enhancement techniques include application with surface modification such as different kind of surface roughness on the tube inside or internally finned tubes. The thermal and hydraulic characteristics of tubes having helical fins on the internal surface – internal helical finning (for example, specially made ridging or helically corrugated internal surface) have been widely investigated over the last 30 years [1–11]. The installation of such designs allows a flow to be swirled in effort to make disturbances in the near-wall layers of heat carrier [1–11]. Most of the published experimental works are devoted to the possibility of industrial use of such tubes in shell-tube heat exchangers. This will permit one either to decrease the mass and size (metal consumption) of heat exchangers, or increase heart load per surface area at fixed overall sizes of the latter. The tubes with helical fins on the internal surface allows heat transfer coefficient to be increased due to disturbances made in the near-wall layers of heat carrier. However, at a time heat carrier flowing inside the tube becomes turbulent [4, 10–11].

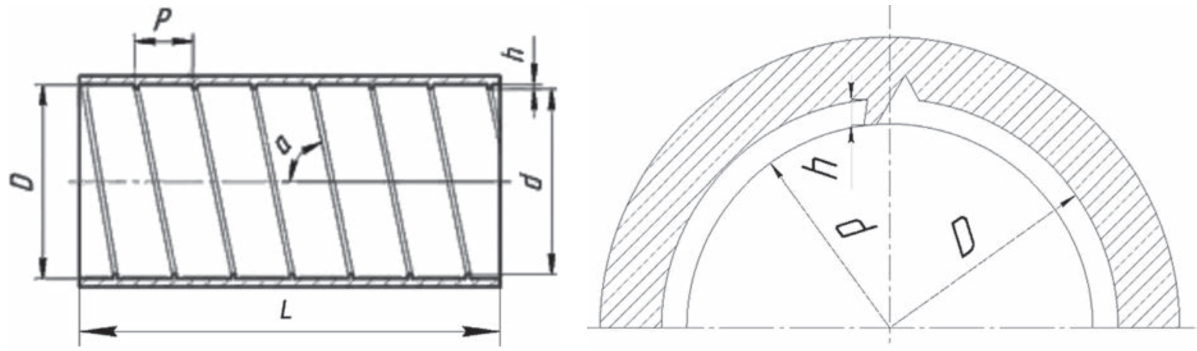
As shown in [2–9], thermal and hydraulic characteristics of tubes with internal helical finning are insufficiently studied for the standard flow conditions of viscous liquids in industrial heat exchangers. The studies of hydraulic resistance of tubes with single-threaded internal helical finning [2–9] have been made over the range of $Re_D = 3{\cdot}10^3 \ldots 1.2{\cdot}10^5$ for the transient and turbulent flow regions of heat carrier. Dimensional geometric parameters of internal helical finning – the ratio of a helical swirling pitch p to a tube diameter D and of a helical finning height h to a tube diameter D – was varied within $p/D = 0.14–1.2$ and $h/D = (6–88){\cdot}10^3$. The attempt to extend the range of the operating characteristics, namely, to make investigations at high Prandtl numbers $Pr_f = 10–90$ and low Reynolds numbers Re_D ranging from $2{\cdot}10^3$ is outlined elsewhere in [10]. The maximum values of thermal efficiency reach $Nu/Nu_0 = 2–2.1$ at a comparable growth of hydraulic resistance $\xi/\xi_0 = 1.8–2.4$, as shown in [8].

A short review permits a conclusion to be made that, despite a significant amount of experimental works on tubes with single-threaded internal helical finning, additional studies should be made of flow structures over the wide Reynolds and Prandtl number range, as well as of geometric tube parameters. In particular, experiments were not made on the flow structure visualization in such tubes because of technical difficulties. To get information about the flow structure in tubes with internal helical finning, numerical simulation methods can be adopted.

It should be emphasized that, for adequate results to be obtained by numerical simulation methods, first, the computational algorithm must be verified by the problems having physical analogs; second, designed computational grids allowing for the geometry features of the object of investigation and allowing boundary layer flow must be correctly used; third, the correct approaches to close the Navier – Stokes equations must be correctly used, i. e., for the equations to be closed, the turbulence model must be correctly chosen.

Thus, the objective of this study is to conduct joint experimental and numerical research of hydraulic resistance and flow structure in heat carrier flow in tubes with internal helical finning. Experimental investigate will make it possible to obtain characteristics curves for the hydraulic resistance of tubes with internal helical finning for laminar, transient and turbulent flow regimes, as well as to determine optimal geometric parameters of tubes with internal helical finning depending on operating parameters at minimum hydraulic loss ξ/ξ_0. Numerical simulation results will allow one to visualize the flow structure in tubes having different helical fins on the internal surface and to qualitatively explain the effects obtained.

1. Object of investigation. An object of investigation is a tube having helical fins on the internal surface and different geometric sizes. The basic geometric parameters of the object of investigation are shown in Fig. 1. The present work studies six tubes having single-threaded helical fins made on the internal surface by the deforming cutting technique combining at a time the processes of deforming and cutting the surface layers of the tube [12–13]. The copper tubes of inner diameter $D = 16$ mm and length $L = 800$ mm were investigated; thus, their relative length was $L/D = 50$, which is indicative of the fact that studies of heat transfer in such tubes must allow for the influence of the thermal entry length. The basic geometric parameters of tubes are cited in Table. Water at 20° ($Pr = 6.97$) served as heat carrier.



a *b*

Fig. 1. Longitudinal (*a*) and transverse (*b*) sections of tubes with internal helical finning: *D* – diameter of a starting smooth tube *d* – deformed tube diameter taken from the height of an intensifier (helical protrusion), *p* – helical swirling pitch of an intensifier, *h* – height of an intensifier (helical protrusion), α – helical swirling angle, $\phi = \dfrac{h^2}{pd}$ – dimensionless parameter responsible for the influence of created swirling on flow characteristics

Basic geometric parameters of tubes under study

α°	*p*, mm	*h*, mm	*d*/*D*	*h*/*D*·10³	*p*/*D*
14	198	1.4	0.825	87.5	12.73
32	80	0.6	0.925	37.5	5
46	48	0.7	0.913	43.75	3
61	28	0.7	0.913	43.75	1.75
76	12	0.4	0.95	25	0.75
87	2,5	0.7	0.913	43.75	0.16

2. Research techniques. Test bench for investigation of hydraulic resistance of tubes with internal helical finning. The hydraulic scheme of the test bench (Fig. 2) is designed in the form of an open loop with a system of forced supply of heat carrier to the working (measuring) section. The test bench comprises the facilities of water storage *2* and water supply *3*, *4* to working section *12*, a piping, a measuring system of bulk (mass) flow rate by reference flowmeters 6, *7* and a control system.

Working section *12* for experimental study of hydraulic resistance of tubes with internal helical finning is a channel with its inlet and outlet located on the axis. Heat carrier temperature at the working section inlet and outlet was controlled by chromel-alumel thermocouples *8, 9*. To measure static pressure, 0.8 mm dia pressure taps were envisaged in connecting pipes at inlet *10* and outlet *11* respectively.

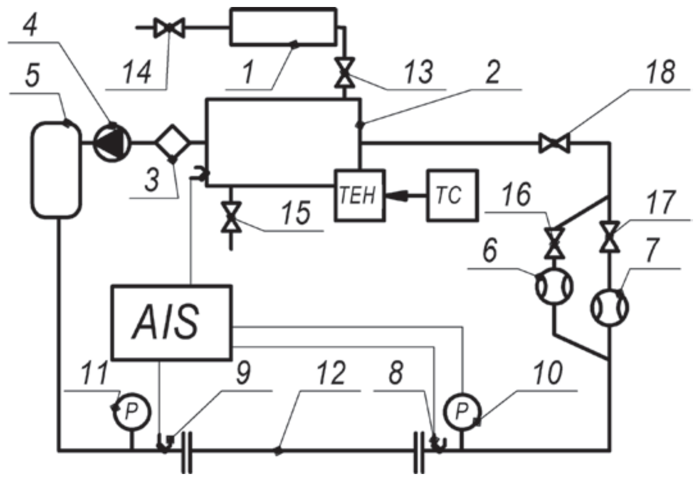


Fig. 2. Test bench for investigation of hydraulic resistance of tubes with internal helical finning: *1*– distiller; *2* – tank-heater; *3* – filter; *4* – high pressure pump; *5* – receiver; *6, 7* – flowmeters; *8, 9* – thermocouples; *10, 11* – pressure sensors; *12* – working section; AIS – automated information system; TEH – tubular electric heater; TC – temperature controller; *13–17* – valves

Pressure drop on the working section was measured by OBEH pressure sensors for isothermal conditions of heat carrier flow.

Hydraulic resistance coefficient was determined by formula (1):

$$\xi = \frac{2\Delta p}{\rho w^2}\frac{D}{L}, \quad (1)$$

where Δp is the pressure drop on the working section, ρ is the heat carrier density, and w is the bulk velocity of heat carrier. A relative error in determining hydraulic resistance did not exceed 6.5 %.

Numerical simulation. Numerical simulation was performed with the use of gasdynamic solver ANSYS Fluent 14.5. Steady-state Reynolds-averaged Navier – Stokes equations (Reynolds equations) and the continuity equation were solved. The Reynolds equations were closed by the κ-ω Menter shear stress transfer model [14]. A medium moving in a tube was assumed to be incompressible and its thermophysical properties (density and viscosity) were assigned constant and independent of temperature and pressure.

A computational grid consisted of tetrahexagonal and hybrid elements closely packed near the tube finning. Total capacity of a computational grid was from 4 mln cells for a tube without finning to 8.5 mln cells for a tube with finning. A minimal size of a computational cell was $0.2 \cdot 10^{-3}$ m.

Massflow rate and operating pressure were assigned at the computational domain inlet; a tube inner surface was assumed to be smooth, on which no slip conditions were realized. Outflow boundary conditions were set at the computational domain outlet. Gravity was directed perpendicular to the incoming flow. In the course of numerical simulation, boundary conditions were predetermined to be consistent with experiment conditions.

3. Analysis of the results obtained. Analysis of the experimental data on hydraulic resistance of tubes with internal helical finning. Experiments on hydraulic resistance of tubes with single-threaded internal helical finning were performed within the turbulent regime of forced water flow over the Reynolds number range $\mathrm{Re}_D = 2 \cdot 10^3 ... 2.5 \cdot 10^5$ (Fig. 3). The experimental results obtained are compared with the data on a tube without internal helical fining and with those calculated by the formula $\xi_0 = 0.3164/\mathrm{Re}^{0.25}$ (Blasius' law).

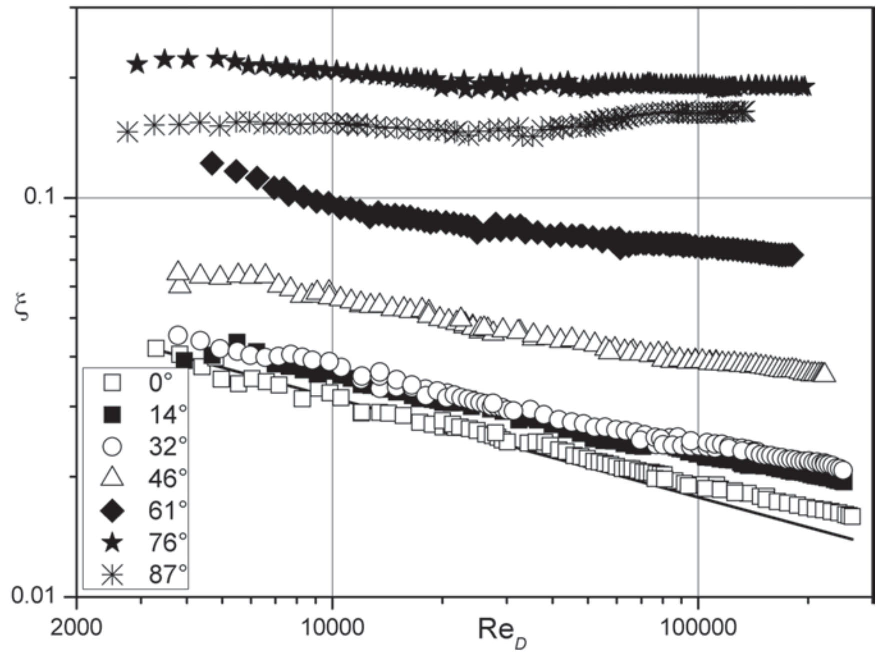


Fig. 3. Experimental data on hydraulic resistance: line – the calculation by the relation $\xi = 0.3164/\mathrm{Re}^{0.25}$ for a tube without internal finning

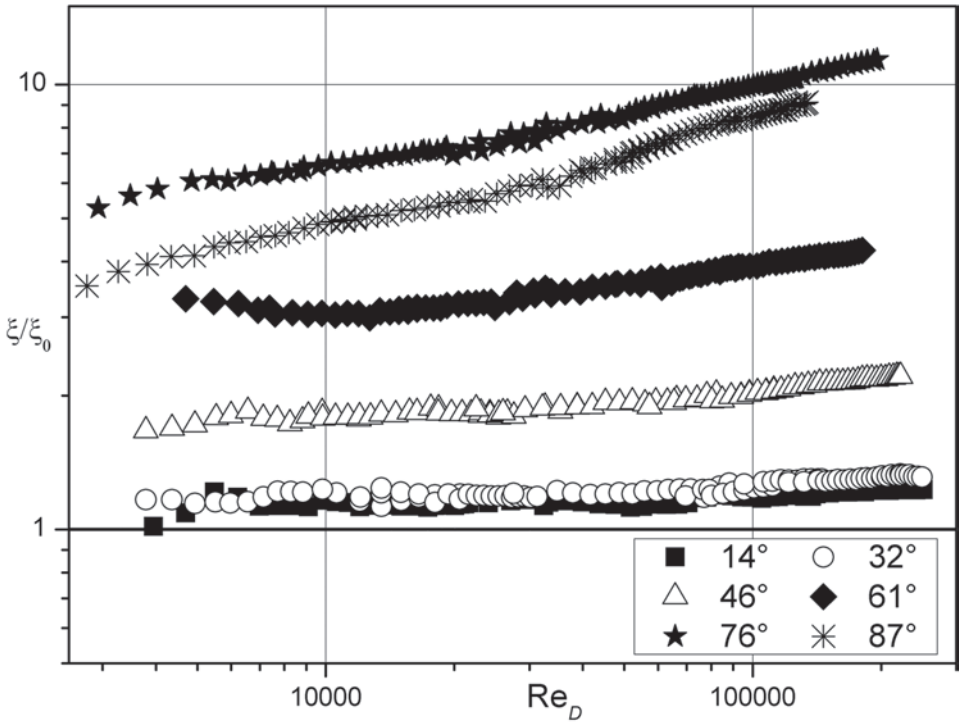


Fig. 4. Comparison of hydraulic resistance coefficients of tubes at different angles of swirling

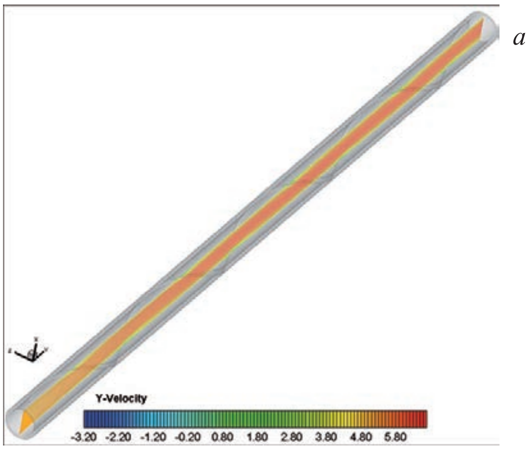
To evaluate the adequacy of the data obtained, the experimental turbulent regime results of the authors of the present article and presented in Fig. 4 were compared with the data [10]. Comparison was made for a tube at an angle of swirling $\alpha = 46°$ and for tube No. 08 [10], as well as for a tube at an angle of swirling $\alpha = 61°$ and for tube No. 06 [10]. The comparison of the results yielded a satisfactory agreement; the disagreement did not exceed 5 %.

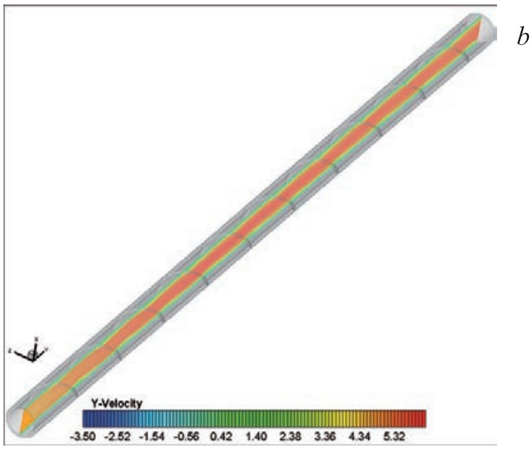
The analysis of the experimental results (Fig. 3) on tubes with internal helical finning revealed that the hydraulic resistance coefficients increased with increasing angle of swirling $\alpha = (14–87)°$. It should be noted that the values of hydraulic resistance coefficient for tubes with $\alpha = 14°$ and $\alpha = 32°$ were close. A maximal increase in hydraulic resistance coefficient in the turbulent Reynolds number range was $\xi/\xi_0 = 11.7$ times in comparison with a smooth channel.

It should be noted (Fig. 4) that at $\mathrm{Re}_D = 4500$, the ratio of hydraulic resistance coefficient of tubes with internal helical finning ξ to hydraulic resistance coefficient of tubes without finning ξ_0 is $\xi/\xi_0 = 1.05–1.1$ for a tube with an angle of fin twisting $\alpha = 14°$; $\xi/\xi_0 = 1.1–1.15$ at $\alpha = 32°$; $\xi/\xi_0 = 1.7–1.8$ at $\alpha = 46°$; $\xi/\xi_0 = 3.0–3.1$ at $\alpha = 61°$; $\xi/\xi_0 = 5.9–6.1$ at $\alpha = 76°$; $\xi/\xi_0 = 3.95–4.05$ at $\alpha = 87°$. Attention should be paid to the fact that the tubes with the angles of swirling $\alpha = 76°$ and 87° (Fig. 3) are characterized by the similar behavior of hydraulic resistance over the Reynolds number range. At that, the hydraulic resistance coefficient ξ for a tube at the angle of swirling $\alpha = 76°$ is 1.25 time larger than for a tube at $\alpha = 87°$. To explain this phenomenon, the flow structure should be visualized. It can be assumed that, as the angle of swirling is increased and the axial pitch is decreased, the circumferential velocity component becomes much smaller than the axial one.

Analysis of numerical simulation results. Because of the difficulties associated both with flow structure visualization in the tube with internal helical finning and with effect explanations, particularly the similarity effect of hydraulic resistance in tubes with $\alpha = 14°$, $\alpha = 32°$, $\alpha = 46°$ and $\alpha = 61°$, as well as tubes with $\alpha = 76°$ and $\alpha = 87°$, numerical simulation results were adopted. The computational algorithm was preliminarily tested for flow in the tube without internal finning; the numerical simulation results were compared with the Blasius law $\xi_0 = 0.3164\,\mathrm{Re}^{-0.25}$. Over the Reynolds number range $3\cdot10^3–2.5\cdot10^5$ the numerical simulation results differed from the Blasius' law by 8–12%.

At small angles of helical swirling (Fig. 5, *a*) finning slightly influences the flow structure in the tube, as well as in the case of the tube without finning the following flow structure is formed: the velocity





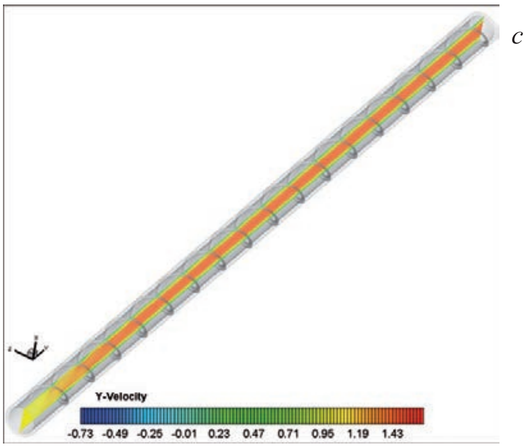
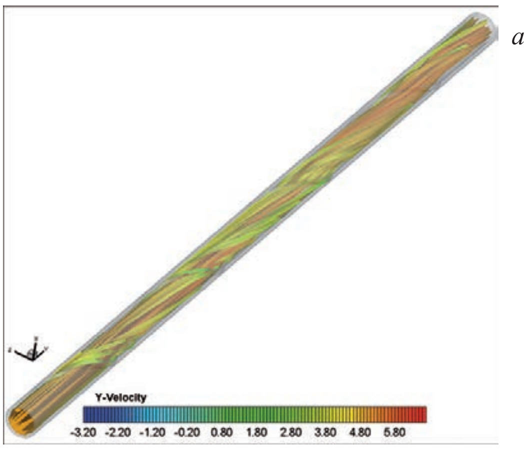
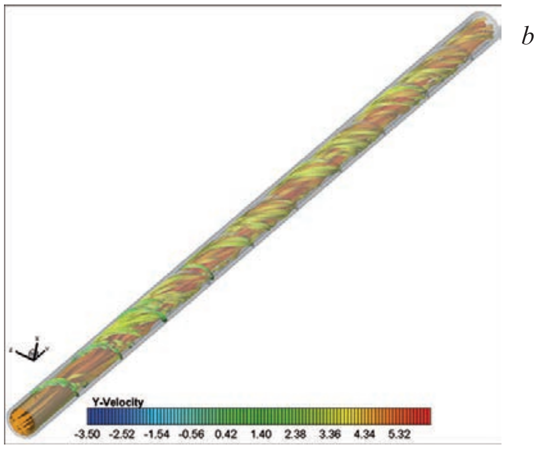


Fig. 5. Velocity distribution over the tube middle cross section:
a – α = 14°; *b* – α = 32°; *c* – α = 46°





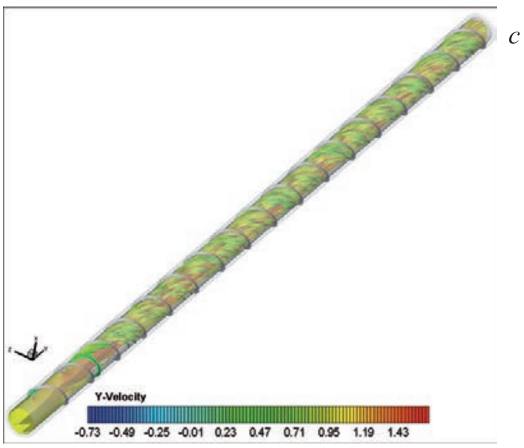


Fig. 6. Flow structure in the tube with internal helical finning:
a – α = 14°; *b* – α = 32°; *c* – α = 46°

on the flow axis has a maximum value and it sharply changes near the wall [15]. Increasing the angle of helical swirling (Fig. 5, *b*, c) disturbs the boundary layer. As the finning height exceeds the boundary layer thickness, this results in a hydraulic resistance growth.

As previously mentioned, as the angle of helical finning (Fig. 6) is increased, in the near-wall layers the share of the circumferential velocity component increases and the share of the longitudinal velocity component decreases. Thus, tubes with a large angle of swirling and a small axial pitch can be considered as tubes with transverse rolling.

Conclusion. Studies of hydraulic resistance of tubes with internal helical finning over a wide range of operating and geometric parameters were made: $Re_D=2\cdot10^3...2.5\cdot10^5$, under the variation of the angle of swirling α = 14°–87°, the relative height of a protrusion h/d = (25–87,5)$\cdot10^{-3}$, the relative axial pitch p/d = 0.16–12.73. It is revealed that the hydraulic resistance of tubes with helical finning increases by a factor of 1.1 to 11.7.

The numerical simulation results showed that, as the angle of helical swirling is increased, in the near-wall layers the share of the circumferential velocity component increases and the share of the longitudinal component decreases. And since the finning height exceeds the boundary layer thickness, the hydraulic resistance grows.

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