Hydraulic resistance of tubes with internal helical finning designed by deforming cutting

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Abstract: 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 for hydraulic resistance of tubes with internal helical finning and to verify computational algorithm; numerical simulation to visualize the flow structure in the tube. The hydraulic resistance of tubes with internal helical finning was studied over a wide range of operating and geometric parameters: \( Re_D = 2 \cdot 10^3 \ldots 2.5 \cdot 10^5 \), under the variation of the angle of swirling \( \alpha = 14^\circ \ldots 87^\circ \), the relative height of a protrusion \( h/d = (25-87.5) \cdot 10^{-3} \), and the relative axial pitch \( p/d = 0.16 \ldots 12.73 \). It has been revealed that the hydraulic resistance of tubes with helical finning increases by a factor of 1.1 to 11.7. The numerical simulation results show that with an increase in the angle of helical swirling, 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.

1. Introduction

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 \( \delta_{temp}/\delta_{dyn} \sim \text{Pr}^{-0.5} \). At some distance from the tube inlet, boundary layers merge and flow becomes stabilized.

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

Many of heat transfer enhancement techniques include surface modification such as different kinds 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. Such designs provide flow swirling producing 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 allow either decreasing the mass and size (metal consumption) of heat exchangers, or increasing heart load per surface area at fixed overall sizes of the latter. The tubes with helical fins on the internal surface increase heat transfer coefficient due to disturbances 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 \( \text{Re}_D=3 \cdot 10^3 \ldots 1.2 \cdot 10^5 \) 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 \( D \) – 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 \( \text{Pr}=10-90 \) and low Reynolds numbers \( \text{Re}_D \) ranging from \( 2 \cdot 10^3 \) is outlined elsewhere in [10]. The maximum values of thermal efficiency reach \( \text{Nu}/\text{Nu}_0=2.2-2.1 \) at a comparable growth of hydraulic resistance \( \xi/\xi_0=1.8-2.4 \), as shown in [8].

A short review brings to a conclusion 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 wide ranges of Reynolds and Prandtl numbers, as well as of geometric tube parameters. In particular, the flow structure visualization has not been realized 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, to obtain adequate results by numerical simulation methods, first, the computational algorithm must be verified by problems having physical analogs; second, it is necessary to correctly use the designed computational grids taking into account the geometry features of the object of investigation and the boundary layer flow; 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 investigation will allow obtaining characteristic curves for hydraulic resistance of tubes with internal helical finning for laminar, transient and turbulent flow regimes, as well as determining optimal geometric parameters of tubes with internal helical finning, depending on operating parameters at minimum hydraulic loss \( \xi/\xi_0 \). Numerical simulation results will enable visualization of the flow structure in tubes having different helical fins on the internal surface and will qualitatively explain the effects obtained.

### 2. 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 \( D=16 \text{ mm} \) and length \( L=800 \text{ mm} \) were investigated; 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 1. Water at \( 20^\circ \) (\( \text{Pr}=6.97 \)) served as heat carrier.
Figure 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), $\alpha$ – helical swirling angle, $\phi = h^2/(p \cdot d)$ – dimensionless parameter responsible for the influence of created swirling on flow characteristics.

Table 1. Basic geometric parameters of tubes under study

| $\alpha$, $^\circ$ | $p$, mm | $h$, mm | $d/D$ | $h/D \cdot 10^3$ | $p/D$ |
|-------------------|--------|--------|-------|-----------------|-------|
| 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  |

3. 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.

![Diagram](image-url)

Figure 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, 14, 15, 16, 17 – valves.

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, pressure taps with 0.8 mm diameter were envisaged in connecting pipes at inlet 10 and
outlet \( \Omega \), respectively. Pressure drop on the working section was measured by OWEN pressure sensors for isothermal conditions of heat carrier flow.

Hydraulic resistance coefficient was determined by formula (1):

\[
\xi = \frac{2 \Delta p D}{\rho w^2 L}
\]

where \( \Delta p \) is the pressure drop on the working section, \( \rho \) 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 \( \kappa-\omega \) 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 assumed constant and independent of temperature and pressure.

A computational grid consisted of tetrahedral and hybrid elements closely packed near the tube finning. Total capacity of the 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 \( 0.2 \times 10^{-3} \) m.

Mass flow 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.

4. 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 \( \text{Re}_D=2 \times 10^3 \ldots 2.5 \times 10^5 \) (Fig. 3). The obtained experimental results are compared with the data on the tube without internal helical fining and with those calculated by the formula \( \xi_0=0.3164/\text{Re}^{0.25} \) (Blasius’ law).

![Graph](image-url)
To evaluate the adequacy of the data obtained, the results for the experimental turbulent regime of the authors of this article, presented in Fig. 4, were compared with the data [10]. Comparison was made for a tube at an angle of swirling \( \alpha=46^\circ \) and for tube No. 08 [10], as well as for a tube at an angle of swirling \( \alpha=61^\circ \) and for tube No. 06 [10]. The comparison of the results has 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 \( \alpha=(14-87)^\circ \). It should be noted that the values of hydraulic resistance coefficient for tubes with \( \alpha=14^\circ \) and \( \alpha=32^\circ \) 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.

\[
\xi=0.3164/Re^{0.25} \text{ for a tube without internal finning}
\]

Figure 3. Experimental data on hydraulic resistance: line – the calculation by the relation

It should be noted (Fig. 4) that at \( Re_D=4500 \), the ratio of hydraulic resistance coefficient of tubes with internal helical finning \( \xi \) to hydraulic resistance coefficient of tubes without finning \( \xi_0 \) is \( \xi/\xi_0=1.05-1.1 \) for a tube with an angle of fin twisting \( \alpha=14^\circ \); \( \xi/\xi_0=1.1-1.15 \) at \( \alpha=32^\circ \); \( \xi/\xi_0=1.7-1.8 \) at \( \alpha=46^\circ \); \( \xi/\xi_0=3.0-3.1 \) at \( \alpha=61^\circ \); \( \xi/\xi_0=5.9-6.1 \) at \( \alpha=76^\circ \); \( \xi/\xi_0=3.95-4.05 \) at \( \alpha=87^\circ \). Attention should be paid to the fact that the tubes with swirling angles \( \alpha=76^\circ \) and \( 87^\circ \) (Fig. 3) are characterized by the similar behavior of hydraulic resistance over the Reynolds number range. At that, the hydraulic resistance coefficient \( \xi \) for a tube at the angle of swirling \( \alpha=76^\circ \) is 1.25 time larger than for a tube at \( \alpha=87^\circ \). To explain this phenomenon, the flow structure should be visualized. It may 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 \( \alpha=14^\circ, \alpha=32^\circ, \alpha=46^\circ \) and \( \alpha=61^\circ \), as well as tubes with \( \alpha=76^\circ \) and \( \alpha=87^\circ \), 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 $\xi_0 = 0.3164 \text{Re}^{-0.25}$. Over the Reynolds number range $3 \times 10^3 - 2.5 \times 10^5$ 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 on the flow axis has the highest magnitude and sharply changes near the wall [15]. Increasing the angle of helical swirling (Fig. 5, b, c) disturbs the boundary layer. The finning height exceeds the boundary layer thickness, and this results in a hydraulic resistance growth.

![Figure 5. Velocity distribution over the tube middle cross section: (a) $\alpha=14^\circ$; (b) $\alpha=32^\circ$; (c) $\alpha=46^\circ$.](image)

It was earlier mentioned that 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.

![Figure 6. Temperature distribution over the tube middle cross section: (a) $\alpha=14^\circ$; (b) $\alpha=32^\circ$.](image)
Figure 6. Flow structure in the tube with internal helical finning: (a) – α=14°; (b) – α=32°; (c) – α=46°

Conclusion
1. Hydraulic resistance of tubes with internal helical finning has been studied over a wide range of operating and geometric parameters: Reₜ=2·10⁶...2·5·10⁶, under the variation of the angle of swirling α=14°-87°, the relative height of a protrusion hld= (25-87,5)·10⁻³, the relative axial pitch pld=0.16-12.73. It has been revealed that the hydraulic resistance of tubes with helical finning increases by a factor of 1.1 to 11.7.
2. The numerical simulation results have shown 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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