The choice of optimum parameters of the tubes with inner helical finning

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Abstract. In this work the studies of the hydraulic resistance and the heat transfer of tubes with inner helical finning over a wide range of the geometric and operating parameters are presented: \( \text{Re}_D=4 \cdot 10^3 - 2 \cdot 10^5 \), under the variation of the angle of swirling \( \theta=14^\circ -87^\circ \), the relative height of a protrusion \( h/D=(25-87.5) \cdot 10^{-3} \), 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 and the augmentation of the Nusselt number varies in the range 1.05-3.35.

1. Introduction

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. Dimensionless geometrical parameters and operating conditions in the studied works [1-10] are listed in Table 1.
Table 1 – The range of the geometrical and operating parameters in the prospected works

| Work | Number of studied tubes, n | D, mm | h/D, 10⁻³ | p/D | Re₀, 10⁴ | Pr |
|------|---------------------------|-------|-----------|-----|-----------|----|
| [2]  | 11                        | 15,9  | 8-88      | 0.20-0.78 | 10-80 | 4-6 |
| [3]  | 12                        | 22,0-25,0 | 20-62 | 0.20-1.20 | 6-20 | 5-0 |
| [4]  | 14                        | 13,9-29,9 | 16-52 | 0.30-0.91 | 10-120 | 5-0 |
| [5]  | 16                        | 17,0-17,9 | 10-69 | 0.14-0.97 | 10-80 | 5-7 |
| [6]  | 4                         | 25,4-25,7 | 21-30 | 1.17-1.18 | 10-100 | 4-3 |
| [7]  | 2                         | 25.4-25.7 | 26-30 | 1.17-1.18 | 3-70 | 4.7-35 |
| [8]  | 25                        | 24.9-26.1 | 17-47 | 0.25-0.67 | 10-60 | 2.2-3.4 |
| [9]  | 3                         | 23.6  | 6-22 | 0.81 | 3-50 | 5-8-10 |
| [10] | 10                        | 18    | 26.7-57.2 | 0,886-1,158 | 2-90 | 2.9-92 |

In Figure 1,(a) and 1,(b) are shown the comparison of the range of dimensionless geometrical parameters of the tubes and the turbulence promoters in present work and presented in Table 1.

![Figure 1](image_url)

Figure 1. Comparison of the range of the dimensionless parameters in the present work with the literature data

It could be mentioned, that of the ratio \( \frac{p}{d} \) in the works [2-10] varies in the narrow range and could be further extended.

2. Object of research

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. 2. 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 2. Water at \( 20^\circ \) (\( Pr=6.97 \)) served as heat carrier.
Figure 2. Longitudinal (a) and transverse (b) section of tubes with internal helical finning. D – diameter of a starting smooth tube, d – deformed tube diameter taken from the height of helical projection, p – helical axial pitch of an intensifier, e – height of a helical projection, θ – angle of swirling, L – length of the tube

Table 2. Basic geometric parameters of tubes under study.

| θ° | p, mm | h, mm | d/D | h/D·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. Experimental installation scheme
The hydraulic scheme of the test bench (Fig. 3) is designed in the form of two open loops for the heating and cold fluids. The water was used as the working fluid for both ducts with the forced supply of the heat carrier to the measuring section.

Figure 3. Hydraulic scheme of the installation. 2, 29 - distillation units, 4, 27 - water tanks, 16, 33 - filters, 34, 17 - water pumps, 35, 15 - dampers, 8, 9, 24, 25 - water flow meters, 11, 12, 18, 20, 37, 32 - thermocouples, 10, 14, 21, 36 - pressure sensors, 12 – working area, 1, 3, 6, 7, 19, 24, 25, 28, 30, 31 - valves, TR - temperatureregulator device, HE - heating element, AMS – automatic measuring system

The measuring section 12 for experimental study of the friction and heat transfer of tubes with inner helical finning is realised as a counter flow double pipe heat exchanger with the hot flow in the outer
and cold flow in the inner ducts. The heat carrier temperature at the working section inlet and outlet for both ducts were controlled by chromel-copel thermocouples, and the pressure drops were measured by OVEn pressure sensors. The inlet and the outlet for the channel for the cold flow was located on the axis, while the heating flow was supplied tangentially to the outer duct. During the experiments were realised a constant heat flux boundary conditions for the wall of the tested tubes.

Hydraulic resistance coefficient was determined by Equation 1:

$$\xi = \frac{2 \Delta p}{p w^2 L}$$

(1)

where $\Delta p$ is the pressure drop on the working section, $p$ is the heat carrier density, and $w$ is the bulk velocity of heat carrier.

The heat transfer coefficient $h_c$ of the water in the inner duct was determined from the equation for the linear overall heat transfer coefficient $U_l$ (Eq. 2 ). This methodology explained in [15].

$$\frac{1}{U_l} = \frac{1}{h_c D} + \frac{1}{2 \lambda_{cop}} \ln \frac{D_w}{D} + \frac{1}{h_h D_w}$$

(2)

where $\lambda_{cop}$-thermal conductivity of the tube wall material, $h_h$-heat transfer coefficient from the hot flow side. The linear overall heat transfer coefficient $U_l$ were defined from the average value of the heat transfer rate $Q_{avg}$, between the heat was supplied by the hot flow and absorbed by the cold flow, as

$$U_l = \frac{Q_{avg}}{\pi \Delta T_{log}}$$

(3)

where $\Delta T_{log}$-mean logarithmic temperature difference.

4. Analysis of results obtained

4.1. 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 $Re_D=2 \cdot 10^3...2.5 \cdot 10^5$ (Fig. 4). The experimental results obtained are compared with the data on a tube without internal helical finning and with those calculated by the formula $\xi_0=0.3164/Re^{0.25}$ (Blasius’ law).

Figure 4. Experimental data on hydraulic resistance: line – the calculation by the relation $\xi=0.3164/Re^{0.25}$ for a tube without internal finning

Figure 5. Comparison of hydraulic resistance coefficients of tubes at different angles of swirling. Index”0” corresponds to the smooth tube
The analysis of the experimental results (Fig. 4) on tubes with internal helical finning revealed that the hydraulic resistance coefficients increased with increasing angle of swirling \( \theta=14°-87° \). It should be noted that the values of hydraulic resistance coefficient for tubes with \( \theta=14° \) and \( \theta=32° \) 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. 5) 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 \( \alpha=14° \); \( \xi/\xi_0=1.1-1.15 \) 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. 4) are characterized by the similar behavior of hydraulic resistance over the Reynolds number range. At that, the hydraulic resistance coefficient \( \xi \) at the angle of swirling \( \alpha=76° \) is 1.25 time larger than for a tube at \( \alpha=87° \).

4.2. Analysis of the experimental data on heat transfer coefficients of tubes with internal helical finning

The result of heat transfer experiments for the Nusselt number measurements versus the Reynolds number is shown in Fig. 6. The analysis of the experimental results shows that with increasing the angle of swirling the heat transfer is rising up. For example, for the Reynolds number equal to \( Re_D=5\cdot10^3 \) the Nusselt number augmentation for the tube with the angle of swirling \( 14° \) is 1.2 times, for the tube \( 87°-3.38 \) times. With increasing the Reynolds number, the Nusselt number also increasing, but the Nusselt number augmentation decreasing. Then for \( Re_D=5\cdot10^4 \) the Nusselt number augmentation in comparison to the smooth tube for the tube with the angle of swirling \( 14° \) is 0.97 times, while for the tube \( 87° \) this value is larger-2.5 times.

The maximal value of the Nusselt number augmentation corresponds to the tube with angle of swirling \( \theta=87° \) and equals 3.4 times for the Reynolds number values \( Re_D=1\cdot10^4 \).

![Figure 6. Experimental data on Nusselt number: line – the calculation by the relation \( Nu=0.023\cdot Re^{0.8}\cdot Pr^{0.4} \) for a tube without internal finning](image)

![Figure 7. Comparison of Nusselt number of tubes at different angles of swirling. Index”0” corresponds to the smooth tube](image)

**Conclusion**

1. Studies of hydraulic resistance and heat transfer 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 \( \theta=14°-87° \), the relative height of a protrusion \( h/D=(25-87,5)\cdot10^3 \), the relative axial pitch \( p/D=0.16-12.73 \).

2. It is revealed that the hydraulic resistance of tubes with helical finning increases by a factor of 1.1 to 11.7, and the Nusselt number augmentation ranges from 1.05-3.5 times.
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