Heat transfer in helium-xenon mixture flowing in straight and twisted tubes with triangle cross-section

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Abstract. The article considers heat transfer in a flow of helium-xenon mixture in straight and twisted tubes with triangle cross sections. The main attention is paid to the heat transfer at large Reynolds numbers for tubes with small effective diameter. Such flows are often found in compact heat exchangers, assemblies of heat-generating elements, and energy-separating devices. CFD simulation data serve to analyze the influence of the cross-section shape and the twist step of the tube on hydraulic resistance and heat transfer. Various methods of generalization of heat exchange data are analyzed, taking into account the effect of gas compressibility on the flow acceleration in the tube and dissipative effects on the equilibrium wall temperature.

1. Introduction
Rare gas mixtures are increasingly offered as coolants in power equipment, for example, in closed gas turbine units (CGTU), operating in the Brighton cycle [1], in compact nuclear reactors designed to provide energy to spacecraft [2], and in devices of machine-free gas-dynamic energy separation [3-5] (in Leontiev tubes). On some characteristics (compressor turbine efficiency, mass of heat exchanger, refrigeration capacity), a helium-xenon mixture with helium mass fraction ranging within 5...10% (Pr = 0.2...0.3) is a more efficient heat carrier than pure gases or other mixtures [1]. For example, when working on a helium-xenon mixture, Leontiev tubes have an order of magnitude greater cooling capacity than when working on air or pure gases. The use of helium-xenon mixture in CGTU leads to an increase in the system efficiency by 4-5% compared to the efficiency in case of pure helium. A small value of Prandtl number leads to a more uniform heating of the coolant and a decrease in thermal deformations of the flow zone of the fuel assemblies, which somewhat likens properties of the mixed-gas coolant to the liquid metal coolant. The thermodynamic and transport properties of such mixtures have been studied quite well [6], but the data on the laws of heat transfer are not widely presented in the literature [3, 7].

Often the channels of power equipment, through which the coolant flows, have a complex shape. Acute angles and convergences can lead to uneven flows around the heat exchange surface and, as a consequence, uneven heat removal, which in turn can result in local overheating of the surface. Various spiral inserts [8], holes and fins [9,10], various types of eddy generators [11, 12] and channel twisting [13, 14] are used to intensify heat transfer.

This work presents new results of numerical simulation of heat transfer in helium-xenon mixture flowing in straight and twisted tubes with triangle cross-sections. The influence of tube twist on the
development of secondary vortex structures and their influence on heat transfer in a wide range of Reynolds numbers are studied. Various methods of generalization of heat exchange data are analyzed, taking into account the effect of gas compressibility on the flow acceleration in the tube and dissipative effects on the equilibrium wall temperature.

2. Problem formulation
The conjugate heat exchange in the helium-xenon mixture flow, pumped through a thin-walled nickel tube with a hydraulic diameter of 6.2 mm (wall thickness of 0.15 mm) was investigated. At the inlet and outlet of the tubes there were adiabatic areas of circular cross-section and a length of 120 mm. The total length of the studied work area was 1030 mm, and the twist step \( s \) was taken equal to \( \infty \) (no twist), 200, 100 and 50 mm. The properties of the wall material were: density – 8900 kg/m\(^3\), heat conductivity – 91.74 W/m·K, and specific heat – 460.6 J/kg·K. The shape of the studied tubes and the flow pattern are shown in figure 1 (from top to bottom: a tube with circular cross section; a straight tube with triangular cross-section; twisted tubes with triangular cross-section and twist step of 200, 100 and 50 mm).

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The properties of helium-xenon mixture were calculated by the parameters of flow deceleration at the tube inlet [6] and were taken constant in length (table 1).

| Property   | Value  |
|------------|--------|
| \( K_{He} \), % | 7.2    |
| \( \mu \), \( \mu \text{Pa} \cdot \text{s} \) | 25.58  |
| \( c_p \), J/kg·K | 527    |
| \( \lambda \), mW/m·K | 65.15  |
| \( \rho \), g/mole | 39.9   |
| Pr | 0.206 |

Along the entire length of the twisted part of the tube, a heat flux of 6000 W/m\(^2\) was supplied to the outer side of the wall. The gas stagnation temperature at the tube inlet was taken equal to 300 K. The pressure at the outlet was 1 atm. The mass flow rate of the gas varied from 0.001 to 0.02 kg/s, which corresponded to the range of Reynolds numbers \( Re_d \) from \( 4 \cdot 10^3 \) to \( 10^5 \).

Numerical simulation of the flow and heat transfer dynamics was carried out using the ANSYS Fluent CFD package under the license of the Siberian Supercomputer Center. For turbulence modeling, the Reynolds stress model (RSM) with nonequilibrium wall functions was used. The system of differential equations was solved by the SIMPLE method.

3. Results
The simulation has resulted in obtaining temperature and velocity fields and local values of heat transfer coefficients on the walls of twisted tubes. Figure 2 presents an example of data visualization in different sections along the length of a straight tube with triangular cross section. In the corners of the channel, areas with a low coolant rate are formed, and heat removal from the surface in these areas is difficult compared to the middle of the wall. At a large thermal load, this can lead to local overheating of the wall. Secondary low-intensity flows are directed from the flow core to the angles along the height of the triangle and back along its side.
Figure 2. Fields of longitudinal velocity and projections of velocity vectors on section planes (top) and temperature fields (bottom) for the straight triangular tube under thermal load of 6000 W/m² and flow rate of the helium-xenon mixture through the tube of 0.015 kg/s in sections: a – 0.014, b – 0.14, c – 0.47, d – 0.88 and e – 1 m, respectively.

Figure 3. Fields of longitudinal velocity and projections of velocity vectors on section planes (top) and temperature fields (bottom) for the triangular twisted tube with a twist step of 50 mm; the boundary conditions and notations are given in figure 2.

The structures formed in the heated area remain in the end adiabatic part of the tube. This flow pattern is in good agreement with the known literature data [16]. Twisting (figure 3) leads to some intensification of the coolant entrainment from the channel corners and to more uniform mixing of the coolant inside the channel. In the final adiabatic section, the flow of the coolant retains the swirling nature.

With an increase in the twist step the hydraulic resistance of tubes increases (figure 4). The greatest increase in resistance is observed at the transition from twist step of 100 to 50, which is associated both with the development of the surface area, and with the intensification of secondary flows. It is important to note that the effect of tube twist should be estimated on changes in the average coefficient of hydraulic resistance, i.e., the pressure drop between the tube inlet and the considered section along the tube length. In compressible flows, a significant pressure gradient necessitates choosing the conditions for determining the characteristic gas density. In our studies, the gas density was determined by the temperature and pressure in the considered section of the tube:
Figure 4. Changes in the coefficient of average hydraulic resistance in circular and triangular tubes with different twist step.

\[ \xi = \frac{\bar{\rho}_m - \bar{\rho}}{\lambda} \cdot \frac{2}{\rho w_s} \]  

(1)

From figure 4 it is apparent that the resistance of tubes does not become constant, which indicates the absence of hydrodynamic stabilization of the flow at more than 120 calibers, while in the flow of incompressible media at turbulent flow regime the stabilization area does not exceed 50 calibers. This can be explained by a significant acceleration of the gas flow due to the volume expansion at higher temperatures and lower pressure in the tube. With twist step decrease, the drop in resistance along the tube length becomes less pronounced.

It is known that in gas flows with a significant impact of compressibility, determining the heat transfer coefficient and generalizing data is more difficult than in incompressible flows [15]. Thus, as the flow rate increases, its thermodynamic temperature decreases, but this does not mean an automatic heat transfer enhancement as in incompressible flows. The decrease or increase in the wall temperature depends on the Prandtl number of gas. At \( Pr > 1 \) the wall will be heated, and at \( Pr < 1 \) it will be cooled. On the basis of simulation data in [7] various methods of generalizing data on heat exchange for air and helium-xenon mixture at flowing in small-diameter circular tubes have been analyzed. It is shown that selecting the mean mass recovery temperature as the defining temperature in the calculation of the Nusselt number, the range of applicability of the Mikheev formula can be expanded:

\[ \overline{Nu} = \frac{q}{T_w - T_{\text{def}}} \cdot \frac{d_j}{\lambda} = 0.021 \cdot Re_d^{0.8} \cdot Pr_d^{0.41} \cdot T_{\text{def}} = T, \]

(2)

to higher Reynolds numbers. For air, \( Nu \) calculations using the Mikheev formula remained valid up to \( 10^5 \) and slightly depended on the pressure. For the helium-xenon mixture, a reduction in inlet pressure while maintaining a high (sonic) velocity at the outlet, which is equivalent to an increase in flow acceleration, significantly reduced the range of Reynolds numbers, in which formula (2) remains fair.

The defining recovery temperature was calculated using the formulas [7]:

\[ T_{\text{def}} = r \sqrt{T} + (1-r) \overline{T}, \quad r = 0.9 \cdot Pr^{0.1} \]

(3)

This approach (red lines and dots in Fig. 5-7), along with the approach from [15] (blue lines and dots in figures 5-7), has been also adopted for the analysis of heat exchange in triangular cross-section tubes. In the work [15] generalization of data on heat exchange at the sound air flow from a small diameter tube was achieved using the recovery temperature, obtained in the experiment with adiabatic
Air flow with the same flow rate, as the determining one. Figure 6 presents data on heat exchange in all considered geometries of working sections. The lines with hollow circles show the dependences of the Nusselt number on the coordinate along the circular tube: \( T_{\text{def}} = T_{\text{ad}} \) (blue), \( T_{\text{def}} = 10 \) (green), \( T_{\text{def}} = 10 \) (red), the calculation by the formula (3), \( T_{\text{def}} = T_{\text{ad}} \) (blue), and the calculation at adiabatic outflow with the same flow rate. It is seen that for a circular tube, the calculation of the recovery temperature by formula (3) leads to a constant Nusselt number over the length, which is close to the value calculated by the Mikheev formula at the Reynolds number of \( 1.2 \times 10^5 \) corresponding to the considered heat exchange conditions. Other ways of data generalization lead to several-fold underestimation of the Nusselt number.

The calculation results for triangular tubes show a slightly different picture. The approach from [7] allows obtaining a little changing value of the Nusselt number on most part of the tube. However, when approaching the output cross section in the region of sound flow velocities, the Reynolds number increases sharply, which indicates an increase in the error of changes of the recovery temperature determined by formula (3) in the region with strong acceleration (pressure drop). This in part determines the spread of heat exchange data in twisted triangular tubes. Figure 5 shows distributions of the recovery temperature along the tube length, resulted from calculations of the wall temperature at the adiabatic gas outflow and from formula (3) for the same conditions.

The application of the approach from [15] has shown that the heat exchange data are well generalized, but the Nusselt number along the tube length decreases. It is thought that this approach allows us to correctly consider the effect of the pressure gradient (flow acceleration), which occurs due to friction pressure losses and the associated gas expansion, on the recovery temperature. However, the gas expansion due to heat supply and the actual change in the stagnation temperature of the heated gas remains unaccounted for in this case. Formula (3) allows taking into account the influence of these factors, but the expression for determining the recovery factor requires a modification to consider the effect of the pressure gradient. This question is still open. Fig. 7 presents a comparison of the known correlations [3] and the Nusselt number averaged for the tube length, obtained using the approaches of [7] and [15], depending on the Reynolds number. The Reynolds number varied due to changes in the gas flow rate through the tubes. It is seen that depending on the choice of the defining temperature in the Nusselt number calculation, the generalization of data leads to different correlations. Thus, the calculation of the defining temperature by formula (3) gives results close to the Mikheev correlation, and the approach from [15] gives results close to the correlation [17].
Conclusion

The influence of the twist of the tubes with a triangular cross section on the hydraulic resistance and heat transfer rate at the outflow of helium-xenon mixture with a small Prandtl number has been analyzed. It is shown that the known approaches to determine the Nusselt number by mass-average recovery temperature do not allow fully taking into account the effect of the volume gas expansion and the associated effect of the pressure gradient on the heat exchange at large Reynolds numbers. The obtained data on heat transfer have not resulted in clear detection of the influence of the tube twist step on the heat transfer intensity. The flow cooling due to increasing outflow rate when approaching the speed of sound has a more significant effect on heat transfer than the presence of secondary flows due to the shape of the tube cross section and the flow twist.

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Nomenclature

\( c_p \) – specific heat at constant pressure, [J/kg·K];
\( d_{ef} \) – effective hydraulic diameter of a tube, [m];
\( K_{He} \) – helium mass fraction in mixture, [kg/kg];
\( M \) – molar mass of gas mixture, [g/mole];
\( r \) – recovery factor, [-];
\( \Pr \) – Prandtl number, [-];
\( \text{Re}_{df} \) – Reynolds number defined by the \( d_{ef} \), [-];
\( s \) – twist step, [m];
\( q \) – specific heat flux, [W/m\(^2\)];
\( T_{\text{def}} \) – defining temperature, [K];
\( T_{ax} \) – thermodynamic temperature at the axis of tube, [K];
\( \overline{T}, \overline{T'}, \overline{T''} \) – mass-average: thermodynamic, total and recovery temperature, accordingly, [K];
\( \mu \) – dynamic viscosity, [Pa·s];
\( \lambda \) – heat conductivity, [W/m·K].

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