Numerical Study on Heat Transfer Performance of Elliptical Internally-Ribbed Twisted Tubes: Postprint

Authors: Han Yong, Wang Dingbiao, Zhang Cancan, Youjian Zhu, Xiang Sa

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Abstract

Numerical investigations were performed on the heat transfer and flow resistance characteristics of elliptical twisted tubes with internal ribs. The effects of cross-sectional geometric dimensions, number of internal ribs n, and pitch S on heat transfer and flow resistance characteristics were analyzed, and the mechanism of heat transfer enhancement was investigated. The results demonstrate that: the number of internal ribs n has a negligible effect on heat transfer and flow resistance; the pitch S and cross-sectional dimensions exert significant influences on heat transfer and flow resistance performance—the smaller the pitch S and the greater the elliptical flattening degree, the better the heat transfer performance; spiral internal ribs further enhance convective heat transfer.

Full Text

Numerical Investigation on Heat Transfer Performance of Inner Fin-Twisted Oval Tubes

Authors: HAN Yong, WANG Ding-Biao, ZHANG Can-Can, ZHU You-Jian, XIANG Sa
(School of Chemical Engineering and Energy, Zhengzhou University, Zhengzhou, Henan 450001, China)

Abstract

This study numerically investigates the heat transfer and flow resistance characteristics of inner fin-twisted oval tubes. The effects of cross-sectional geometry, number of inner fins (n), and helical pitch (S) on heat transfer and flow resistance performance are analyzed, and the mechanism of heat transfer enhancement is examined. Results indicate that the number of inner fins n has negligible influence on heat transfer and flow resistance. In contrast, helical pitch S and cross-sectional dimensions exert substantial effects: smaller helical pitch and greater oval flattening degree yield better heat transfer performance, while the spiral inner fins further enhance convective heat transfer.

Keywords: inner fin-twisted oval tube; enhanced heat transfer; numerical investigation

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Introduction

Twisted tubes represent a novel type of heat transfer enhancement tube [?, ?]. Fluid rotates within the tube, generating secondary flow perpendicular to the main flow direction that significantly enhances convective heat transfer. Previous studies by domestic and international scholars have investigated laminar flow and heat transfer in twisted tubes. Kotorynski et al. [?] employed perturbation methods and symbolic algebra software to obtain low-order perturbation solutions for steady laminar flow in twisted tubes based on the Poiseuille flow model. Meng et al. [?] conducted theoretical analysis and numerical calculations of laminar heat transfer and flow resistance characteristics in helically twisted oval tubes, proposing correlations for Nusselt number and friction factor. Liu et al. [?] performed simulation studies for fluids with Reynolds numbers below 1000. Tan [?] used perturbation methods to solve the fully developed laminar flow and heat transfer equations in twisted oval tubes, elucidating the heat transfer enhancement mechanism under laminar conditions.

However, research on combining twisted tubes with other heat transfer enhancement techniques remains limited. Si et al. [?] experimentally investigated shell-side heat transfer and flow resistance in a heat exchanger with externally threaded spiral flat tubes. Zhang [?] experimentally studied the heat transfer and flow resistance performance of a double-shell-pass twisted tube heat exchanger with internally threaded twisted tubes, developing correlations for heat transfer coefficients and friction factors for both tube-side and shell-side flows.

The value of numerical simulation extends beyond design guidance to enabling the exploration of innovative designs. Inner fins represent an effective heat transfer enhancement method, and their combination with twisted tubes creates the inner fin-twisted oval tube configuration. This study employs Fluent software for numerical simulation.

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1 Numerical Simulation Method

1.1 Geometric Model

The geometric model of the inner fin-twisted oval tube is illustrated in [Figure 1: see original paper]. Key parameters include the inner major axis (A) and minor axis (B) of the oval cross-section, helical pitch (S), number of spiral inner fins (n), fin height (e), and fin helix angle (α), which correlates with pitch S. Additional parameters comprise fin base width (tb), fin tip width (tt), and fin tip angle (β). The geometric model is based on a Φ25\$$\times$2.5$mm base circular tube with spiral inner fins, featuring a tube length (L) of 1000 mm, fin height (e) of 0.3m$ of 41°, and fin tip width (tt) of 0.24 mm. The geometric parameters used in numerical simulations are summarized in .

For convenience, the following naming convention is adopted: A25-n16-S200 denotes an inner fin-twisted oval tube with A = 25 mm, n = 16, and S = 200 mm.

1.2 Mesh Generation

According to fundamental principles of numerical heat transfer [**?**], continuous physical fields must be discretized by generating a mesh that approximates the field using a finite set of variable values at discrete points. ICEM CFD is employed to generate predominantly hexahedral meshes. [Figure 2: see original paper] presents the mesh configuration for the A25-n24-S200 inner fin-twisted oval tube model.

1.3 Boundary Conditions and Solution Settings

Mathematical model assumptions: (a) constant fluid properties; (b) steady laminar flow in the tube; (c) negligible viscous dissipation and body forces.

Boundary conditions: The inlet is specified as a velocity inlet at 300 K, with Reynolds number ranging from Re = 500 to 1800. The tube wall is maintained at a constant temperature of 373.15 K. The outlet is designated as a pressure outlet with specified static pressure and appropriate backflow conditions. The characteristic length for the inner fin-twisted oval tube is taken as the equivalent diameter of the base circular tube.

Solution settings: The SIMPLE algorithm is used for pressure-velocity coupling. The Green-Gauss Cell-Based scheme is employed for gradient calculations, pressure is discretized using the Standard scheme, and all other equations utilize second-order upwind schemes. The convergence criterion is set to 10^{-6}.

1.4 Grid Independence Verification

Grid independence is verified using the A25-n24-S200 tube with five different mesh configurations (cross-section mesh count × axial mesh count) at Re = 1000. The results, presented in , demonstrate maximum deviations of less than 1.16%

for Nusselt number and 0.7% for friction factor. Based on these findings, all subsequent simulations employ at least 5900 mesh elements in the cross-section and 250 elements in the axial direction, ensuring grid-independent solutions.

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2 Results and Analysis

2.1 Effect of Fin Number (n) on Nusselt Number and Friction Factor

[Figure 3: see original paper] illustrates the influence of fin number n (16, 24, 32, 40) on Nu and f for tubes with A = 25 mm and S = 200 mm. The results show that Nu increases with Reynolds number, following identical trends across different fin numbers. Similarly, friction factor f decreases with increasing Re, exhibiting consistent declining trends. Variations in fin number n produce negligible changes in both Nu and f, particularly for the friction factor, which shows minimal differences and remains essentially constant. This indicates that flow and heat transfer characteristics are similar for fin numbers of 16, 24, 32, and 40, suggesting that fin number n has minimal impact on laminar heat transfer and flow resistance in inner fin-twisted oval tubes.

2.2 Effect of Helical Pitch (S) on Nusselt Number and Friction Factor

[Figure 4: see original paper] presents results for cases with varying helical pitch S only. The data show that as Re increases, the increment in Nu (ΔNu) grows for all tubes, while the increment in f (Δf) remains comparable. At constant Re, Nusselt number Nu increases significantly with decreasing helical pitch S, indicating that smaller pitch enhances heat transfer. However, friction factor f also increases as pitch decreases. Compared with fin number n, helical pitch variation exerts a much more pronounced influence on both Nu and f.

[Figure 5: see original paper] displays secondary flow contours at Re = 1200 and x = -800 mm for different pitch values. The results demonstrate that smaller helical pitch induces stronger secondary flow due to greater tube deformation and larger fin helix angle α, which creates greater disturbance in the boundary layer and enhances heat transfer. The secondary flow is most intense at the major axis of the oval cross-section, where the spiral flow is strongest.

2.3 Effect of Cross-Section Dimensions on Nusselt Number and Friction Factor

[Figure 6: see original paper] shows the influence of cross-sectional dimensions on Nu and f. At constant Re, Nusselt number Nu increases with increasing A/B ratio, indicating improved heat transfer capability. However, friction factor f also increases under the same conditions. The cross-section A×B = 22$×$17.9 mm exhibits minimal flattening with an equivalent diameter of 19.68 mm, similar to a circular tube, resulting in modest heat transfer enhancement. Compared with fin number n, cross-sectional dimensions have a substantially greater impact on flow and heat transfer performance in inner fin-twisted oval tubes.

2.4 Local Nusselt Number Analysis

[Figure 7: see original paper] compares the local Nusselt number distribution along half the perimeter in the major axis direction at Re = 800 and z = -800 mm for the A25-n24-S200 tube versus a circular tube. The x-axis indicates coordinates along the oval major axis. The results reveal that regions where the local Nu of the inner fin-twisted oval tube is lower than that of the circular tube constitute a relatively small proportion of the total perimeter, while the average Nu of the spiral inner fin-twisted tube exceeds that of the circular tube. Furthermore, local Nu exhibits parabolic-shaped sudden increases at each inner fin location, significantly exceeding nearby values with peak values approximately five times those of the circular tube. This confirms that inner fins promote local Nu enhancement at the wall surface, substantially strengthening the convective heat transfer process.

Conclusions

This study numerically simulated the heat transfer and flow resistance characteristics of inner fin-twisted oval tubes, yielding the following preliminary results:

1. Increasing the number of inner fins n produces negligible changes in both Nusselt number Nu and friction factor f.
2. As helical pitch S decreases, flow disturbance intensifies, secondary flow becomes more pronounced, and Nu increases, thereby enhancing heat transfer, though friction factor f also increases.
3. Cross-sectional dimensions significantly affect heat transfer and flow resistance performance, with greater oval flattening yielding better heat transfer enhancement.

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Note: Figure translations are in progress. See original paper for figures.