Numerical Analysis of Flow Characteristics in Large Diameter Helical Coils with Constant Pitch

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1. Introduction

Helical coil heat exchangers are widely used in various engineering applications due to their superior heat transfer efficiency, compact design, and enhanced fluid mixing. Compared to straight tubes, helical coils generate secondary flow structures, such as Dean vortices, which improve convective heat transfer while also increasing pressure drop. These features make helical coils advantageous for use in industries such as chemical processing, refrigeration, and power generation, where high thermal performance is required in limited space.

In nuclear Small Modular Reactor (SMR) designs, Once-Through Steam Generators (OTSGs) often employ helical coil configurations to maximize heat transfer efficiency while maintaining a compact structure. Traditional helical coil heat exchangers generally feature moderate curvature, but recent advancements in nuclear technology have led to the adoption of large helix diameter configurations. These large-curvature helical coils offer several benefits, including improved flow distribution, reduced risk of flow instability, and better thermal performance under high-pressure conditions. However, their thermal-hydraulic behavior differs significantly from conventional helical coils, making it crucial to assess their pressure drop and heat transfer characteristics accurately.

While numerous empirical correlations exist for predicting friction factors and Nusselt numbers in helical coils, most of these studies focus on small to moderate curvature ratios. The applicability of such correlations to large-curvature helical coils remains uncertain due to the lack of experimental validation and limited numerical studies in this area.

Ito [2] and Ju et al. [3] developed friction factor correlations based on pressure drop experiments conducted on helical coils with curvature ratios ranging from 3.19 to 7.69. Similarly, Xin et al. [7] and Schmidt et al. [5] proposed Nusselt number correlations derived from heat transfer experiments and modeling within curvature ranges of 5.25–17.42 and 4.02–6.5, respectively.

Experimental investigations on large-diameter helical coils are challenging due to the high costs and complexities associated with full-scale testing. Therefore, numerical approaches such as Computational Fluid Dynamics (CFD) provide an essential tool for analyzing flow and heat transfer in these configurations.

This study aims to investigate the flow characteristics in large-curvature helical coils with CFD approach and evaluate its accuracy by comparing with conventional empirical correlations for pressure drop and heat transfer performance. Using CFD simulations conducted in ANSYS CFX, we analyze flow behavior for Reynolds numbers ranging from 10,000 to 50,000 while comparing different turbulence models, including k-ε, k-ω SST, and the Reynolds Stress Model (RSM). The findings of this study contribute to the development of more accurate predictive models for large-diameter helical coil heat exchangers, particularly in SMR OTSG applications, where reliable thermal-hydraulic predictions are critical for safe and efficient operation.

2. Methodology

2.1. Computational Model and Meshing

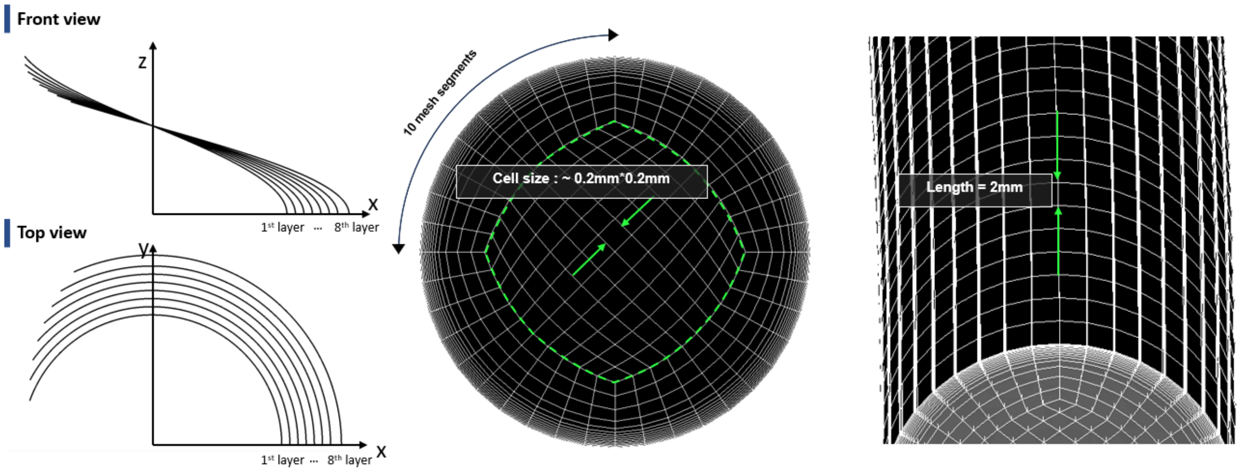
This study focuses on the analysis of a large-diameter helical coil, exhibiting a curvature below approximately 0.5 and a torsion below approximately 0.2. A total of eight coil geometries were designed, each with a different configuration. All coils were constructed with an identical pitch to ensure consistency in axial spacing. The following equations are employed to evaluate the curvature and torsion of the geometry.

$$Curvature(\kappa) = \frac{R}{R^2 + \left(\frac{p}{2\pi}\right)^2} \tag{1}$$

$$Torsion(\tau) = \frac{\left(\frac{p}{2\pi}\right)}{R^2 + \left(\frac{p}{2\pi}\right)^2} \tag{2}$$

Where R defines the radial distance from the helix centerline to the pipe axis, and p denotes the axial distance between successive turns of the coil. As indicated by Equations (1) and (2), in helical coil geometries with identical pitch, increasing the helix diameter results in a decreasing trend in both curvature and torsion. The geometries investigated in this study were designed with the same pitch, and their detailed configurations are presented in Figure 1(a).

The computational mesh was generated using SALOME, employing a structured grid with an O-grid topology for the pipe cross-section to accurately resolve the flow field. Boundary layer refinement was applied to capture near-wall velocity and temperature gradients with high resolution. To ensure compatibility with



(a) Numerical analysis domain (b) Cross-sectional meshes (c) Axial meshes

Figure 1. CFD analysis schematic domain geometry (a) and mesh shape (View of cross-sectional mesh(b) and axial mesh(c)).

turbulence models such as RSM, SST, and k-ω, the mesh was constructed to maintain y^+ values below 1 across all wall surfaces. The computational domain included multiple coil turns to capture the periodic nature of the flow, and mesh independence tests were performed to confirm numerical accuracy. The geometry and mesh structure used in this study are presented in Figure 1.

2.2. Governing equation

The CFD simulations were based on the conservation equations for mass, momentum, and energy:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho u) = 0 \tag{3}$$

$$\frac{\partial (\rho u)}{\partial t} + \nabla \cdot (\rho u u) = -\nabla P + \nabla \cdot \tau + \rho g \tag{4}$$

$$\frac{\partial (\rho h)}{\partial t} + \nabla \cdot (\rho h u) = \nabla \cdot (k \nabla T) + \phi \tag{5}$$

Where τ represents the viscous stress tensor, h is the enthalpy, k is the thermal conductivity, and Φ represents viscous dissipation. Water was used as the working fluid, and temperature-dependent material properties were incorporated into the simulation. Turbulence effects were modeled using the RSM, SST, and k-ε models, as discussed in the following section.

2.3. Boundary condition and turbulence modeling

At A mass flow rate corresponding to Reynolds numbers between 10,000 and 50,000 (based on the pipe's hydraulic diameter) was specified at the inlet, with a uniform velocity profile. The outlet boundary condition was defined as a gauge pressure of 0 Pa. A constant wall heat flux of 30 kW/m² was imposed as the thermal boundary condition.

Water was selected as the working fluid and modeled as an incompressible substance with temperature-dependent thermophysical properties to improve the fidelity of thermal analysis.

Three turbulence models were employed to test the dependency. Among the turbulence models considered, the standard k-ε model was employed as a baseline due to its widespread use and computational efficiency. To enhance near-wall resolution, the k-ω SST model was adopted, as it integrates the advantages of both the k-ε and k- ω formulations. Additionally, the Reynolds Stress Model (RSM) was utilized to account for turbulence anisotropy and to accurately capture the complex secondary flow structures that are characteristic of curved geometries, thereby providing more physically realistic predictions.

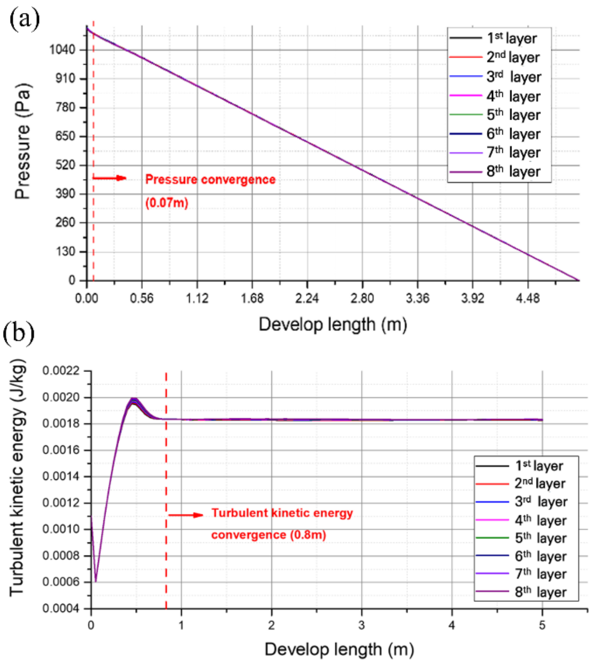


Figure 2. (a) Pressure development length and (b) turbulent kinetic energy development length for each layer of the helical coil.

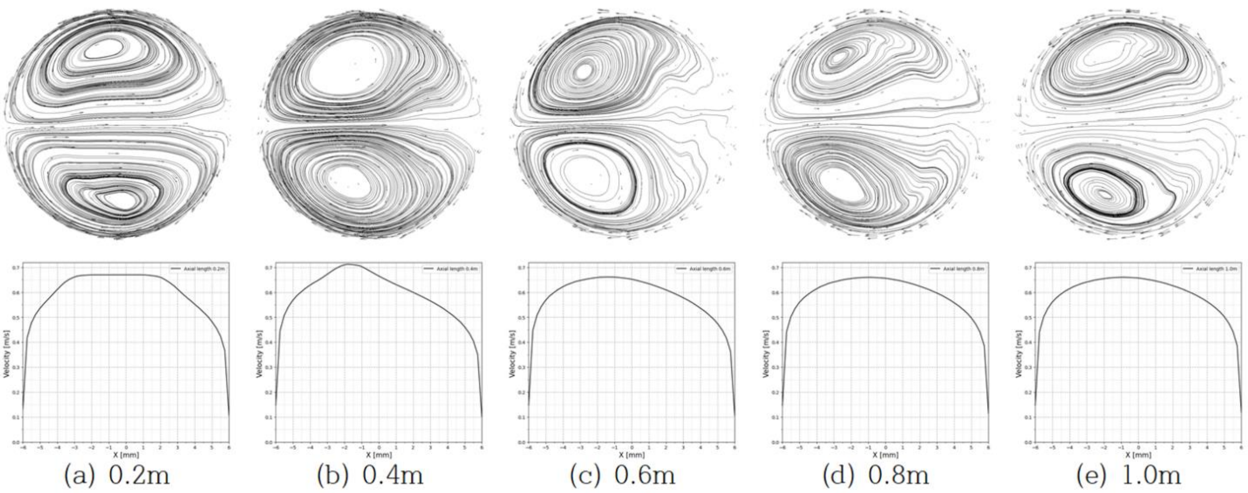


Figure 3. Development of velocity profiles and secondary flow structures along the axial direction in the first layer (Re ≈ 50,000).

2.4. Data analysis and Validation

The friction factor and Nusselt number were derived from the CFD results and subsequently compared with empirical correlations, such as those by Mori & Nakayama and Ito. The deviation between numerical predictions and empirical correlation values was examined to assess the applicability of these correlations to large-curvature helical coils. The friction factor and Nusselt number were derived from the CFD results and subsequently compared with empirical correlations, such as those by Mori & Nakayama and Ito. The deviation between numerical predictions and empirical correlation values was examined to assess the applicability of these correlations to large-curvature helical coils.

3. Result and Discussion

The heat transfer and pressure drop data used in this study were obtained after ensuring that the flow in the helical coil reached a fully developed state. Fully developed flow was confirmed by analyzing the pressure distribution, turbulence kinetic energy development, and velocity profile evolution along the coil. It was observed that the flow became fully developed when the axial flow distance exceeded approximately 1.0 m. The pressure development along the flow direction is shown in Figure 2, while the velocity profiles and secondary flow streamlines are illustrated in Figure 3. Only data from the fully developed flow region were considered for evaluating the heat transfer and pressure drop characteristics.

3.1 Flow Characteristics and Pressure Drop

The CFD simulations revealed the presence of strong secondary flow structures within the helical coil. These secondary flows resulted in enhanced fluid mixing and a corresponding alteration in pressure distribution along the coil. Due to the curvature-induced centrifugal forces, the pressure drop across the coil was significantly higher than that of a straight tube with the same hydraulic diameter. The results showed that the RSM model provided the most accurate predictions of pressure drop when compared to empirical correlations, capturing the effects of secondary flow and turbulence anisotropy more effectively than the k-ε and k-ω SST models.

The comparison of friction factors obtained from CFD and empirical correlations proposed by Mori and Nakayama, Ito, Yao, and Ju [1-4] indicated that while conventional correlations provide reasonable estimates, deviations occur at high Reynolds numbers. The Darcy friction factor was calculated from CFD results using the following expression:

$$f = \frac{2 \cdot \Delta P \cdot d}{\rho \cdot L \cdot \bar{u}^2} \tag{6}$$

where ΔP is the pressure drop along the flow path, d is the hydraulic diameter, L is the flow development length, $\bar{u}$ is the cross-sectional average velocity, and ρ is the fluid density. All quantities were extracted from the fully developed region of the flow.

The Mori & Nakayama correlation [1] exhibited better agreement with CFD results than other available

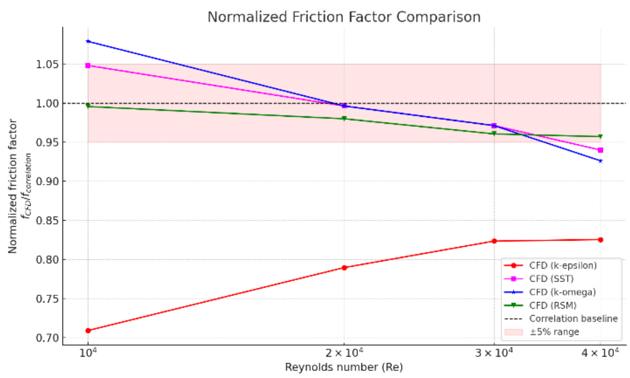


Figure 4. Normalized friction factor($f_{CFD}/f_{correlation}$) in a helical pipe as a function of Reynolds number for different turbulence models.

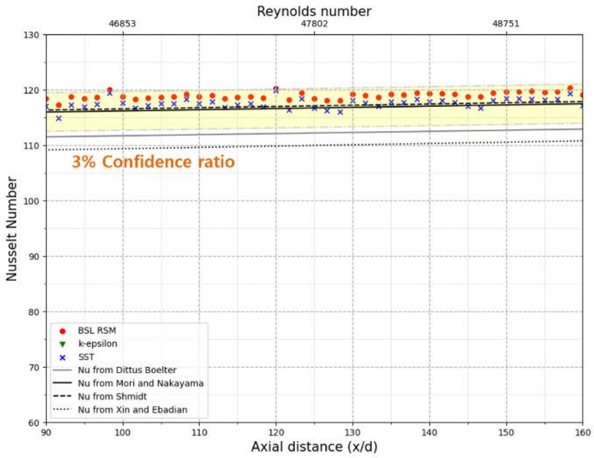


Figure 5. Comparison of CFD-based Nusselt number with empirical correlations for different turbulence models.

correlations, particularly for Reynolds numbers above 20,000. The k-ε model consistently underestimated pressure drop due to its limitations in resolving complex flow structures, whereas the RSM model demonstrated excellent agreement within a 5% deviation from empirical predictions.

3.2 Heat Transfer Performance

The Nusselt number was calculated using the following relation:

$$Nu = \frac{q'' \cdot d}{k \cdot (T_w - T_b)} \quad (7)$$

where q″ is the imposed wall heat flux, **d** is the hydraulic diameter, k is the thermal conductivity of the fluid, T_w is the average wall temperature, and T_b is the bulk fluid temperature. The bulk temperature T_b was determined using a mass flow average formulation to ensure accurate representation of thermal transport under flow-dominant conditions:

$$T_b = \frac{\int_A \rho u T dA}{\int_A \rho u dA} \quad (8)$$

All parameters were extracted from the fully developed region of the helical coil to ensure consistency and eliminate entrance effects.

The secondary flow structures induced by the helical geometry resulted in an average enhancement of approximately 5% in the Nusselt number compared to straight pipes across the entire range of Reynolds numbers. As Reynolds number increased, the Nusselt number also increased, showing trends consistent with empirical correlations by Mori and Nakayama, Schmidt, and Xin and Ebadian [1,5,7].

Among the turbulence models, the RSM and k-ω SST models demonstrated strong agreement with the empirical data, with average deviations of less than 3% under most conditions. In contrast, the k-ε model consistently underestimated the convective heat transfer.

Furthermore, the k-ω SST model exhibited superior performance in predicting local convective heat transfer rates, particularly in the midsection and downstream regions of the coil, indicating its suitability for accurately capturing the effects of curvature-induced secondary flows.

4. Conclusion

This study demonstrated the effectiveness of CFD simulations in evaluating the flow and heat transfer characteristics of large-curvature helical coils. The findings confirmed that empirical correlations could reasonably predict performance but may require modifications for large-diameter configurations. The RSM model was found to be the most accurate in capturing both pressure drop and heat transfer characteristics. These results provide valuable insights for optimizing helical coil designs in SMR OTSG applications.

5. Acknowledgement

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