Thermal–Hydraulic Performance Analysis of Smooth and Corrugated Helical Coil Heat Exchangers Using Mathematical and Numerical Approaches

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Abstract

The growing demand for compact, efficient, and sustainable thermal management systems has led to a growing interest in helical coil heat exchangers due to their superior heat transfer characteristics. The influence of the Dean number and Colburn factor on heat transfer and flow behavior is examined, emphasizing the role of induced secondary flows. A mathematical model was developed to compute Nusselt number, friction factor and pressure drop using modified correlations for corrugated geometries. Numerical simulations using ANSYS Fluent revealed that corrugated tubes enhance Nu by up to 32% compared to smooth tubes. Mathematical correlations are developed based on governing equations of mass momentum and energy equation. the thermal-hydraulic performance factor peaked at 1.38 near De ≈ 734. The coefficient of performance improved by 19–26%, with a 0.12 m coil diameter delivering optimal efficiency. Results showed strong agreement (RMSE < 4%), validating the approach.

Keywords: Corrugated Helical Coil, Dean Number, Thermal-Hydraulic Performance, Vapor Compression Refrigeration System, Colburn Factor

1. Introduction

Cooling represents the most rapidly increasing form of energy consumption in buildings. Vapor compression refrigeration systems, commonly used in applications like air conditioning and cooling, refrigeration contributes significantly to global energy demand. The IEA's 2018 [1] report The Future of Cooling estimates that such systems already account for 10% of global electricity, or nearly 20% of electricity used in buildings. More recent IEA analysis further highlights that cooling contributes about 7% of global greenhouse gas emissions, and in some hot countries, may drive over 70% of peak electricity demand [2]. Protected systems, particularly room air-conditioners, consume over 2,000 TWh of electricity each year about 7% of global generation while emissions from the cooling sector have risen to more than 1 Gt of CO_2 annually [3]. The growing demand for air conditioners is one of the most ignored challenges in today's energy debate. By applying higher efficiency standards for cooling, governments can simultaneously cut emissions, lower costs, and reduce the need for new power plants, making it one of the simplest yet most impactful actions in energy policy [4].

Raised ambient temperatures and extensive use of cooling appliances significantly drive peak electricity demand. Within this framework, water-cooled shell-and-helical coil condensers present a superior and energy-efficient alternative to conventional air-cooled systems, owing to their enhanced heat transfer performance and reduced compressor work requirements [5,6]. These condensers increases the superior heat transfer of water as a coolant and the boosted turbulence created by helical coil geometry to reduce condenser temperature, lower compressor work, and elevate the unit's coefficient of performance (COP) [7]. This efficiency gain is corroborated in studies showing that water-cooled systems, despite higher installation complexity, outperform air-cooled chillers in energy and emissions performance—delivering notably higher seasonal COP (ESEER) under comparable operating conditions [8]. Integrating such high-performance condensers alongside stronger cooling efficiency standards can thus play a potent role in reducing the need for new power plants, curbing emissions, and lowering operational costs—providing governments with one of the most accessible avenues for sustainable energy policy interventions [9,10].

The Dean number describes flow in helically coiled tubes under centrifugal forces. Dean showed that secondary flows, called Dean vortices, enhance heat transfer. Their strength depends on the Reynolds number and the ratio of tube diameter to coil diameter (d/D) [11]. Vishvakarma et al. [12] studied Dean vortices and found that they promote radial mixing with minimal axial back-mixing by moving fluid from the inner to the outer tube wall, thereby enhancing heat and mass transfer rates and reducing residence time variations [13].The transition from laminar to turbulent flow in a helical coil occurs at a higher critical Reynolds number than in a straight tube. The transition from laminar to turbulent flow in a helical coil occurs at a higher critical Reynolds number than in a straight tube [14].

Previous studies have explained two techniques to enhance heat transfer in helically coiled tubes: 1) modifying helically coiled tube surfaces to disrupt the thermal boundary layer [15] and 2) blending nano powders with refrigerants (nanofluids) to enhance fluid mixing, pulsation, and turbulence [16]. Inserting circular grooves [17] or internal corrugations in tubes [18] improves the roughness of the surface flow pattern, causing the heat transfer boundary layer to be destroyed during fluid flow. It strengthens radial fluid mixing for smoother flow and reduces the thermal resistance of the boundary layer [19]. Verma et al. [20] explained the impact of pitch and depth of corrugation on the heat transfer, Nusselt number as a function of Reynolds number and mass flow rate. There were limited studies on internal corrugated or roughness in helically coiled tubes analysing effect of internal corrugation on Dean Number, Colburn factor, inlet pressure, velocity and Nusselt number's heat transfer effects. However, relying solely on experimental investigation or numerical analysis is often inadequate, as each approach can be costly, time-consuming, and challenging [21] particularly when it comes to capturing detailed flow parameters inside the coil. This limitation highlights the need for mathematical modelling as a complementary approach to experimental and numerical analysis. The aim of the research work was to investigate the effect of geometrical shape and corrugation on Dean number, Colburn factor and heat transfer through Mathematical, numerical analysis and compared between them.

2. Mathematical Modelling

2.1 Governing Equations

In general flow in helical coil differs from flow in a conventional pipe due to the curvature of the coil and corrugation inside the tube, which generate radial forces and secondary flows. Both radial (centrifugal) and secondary flows significantly impact pressure drop, velocity distribution and heat transfer characteristics [12]. The elementary / fundamental governing equations for heat transfer and fluid flow in helical coil as expressed through the continuity, momentum and energy equations, describe mass conservation, momentum balance and energy transfer within the system respectively [13].

2.2 Continuity equation: it ensures that mass is conserved as the fluid flows through the helical coil. It is given by:

$$\frac{\partial \rho}{\partial t} + \nabla . (\rho v) = 0 \quad \text{-------(eq.1)}$$

for incompressible fluid $\nabla . (\rho v))=0$

Where ρ is fluid density, t is time, v is velocity vector (u, v, w), $\nabla \cdot v$ = divergence of velocity field.

2.3 Momentum equation (Navier-Stokes equations): It considers the forces acting on the fluid elements, which includes internal, pressure, viscous and body forces (gravity) forces.

$$\rho \frac{\partial v}{\partial t} + v\,\nabla v = -\nabla P + \mu \nabla^2 v + \rho g \quad \text{------------(eq.2)}$$

Where P is pressure (Pa), μ is dynamic viscosity $\nabla^2$is Laplacian of velocity (viscous diffusion) and g is gravitational acceleration vector. In a helical coil, the Dean number considered for the secondary flow due to coil curvature and which is given by:

$De = \sqrt{\frac{R_e}{D_c}}$--(eq.3)

Where$Re = \frac{\rho v d}{\mu}$ is the Reynolds number, dh = 4A/P , d is the pipe diameter, and Dc is the coil diameter, A is cross-sectional flow area and P= wetted perimeter [14].

2.4 Colburn Factor (j)

The Colburn factor is a dimensionless parameter used to describe convective heat transfer in relation to pressure drop: where $j = \frac{Nu}{pr^{\frac{2}{3}}}$ [15]

Where is the $st = \frac{Nu}{Re\ Pr}$ is the Stanton number

$pr = \frac{\mu Cp}{k}$ is the Prandtl number

$Nu = \frac{hd}{k}$is the Nusselt number

The relationship between j and De provides insight into the exchanger's thermal efficiency and pressure losses [16].

For turbulent flow in helically coiled tubes, the Dittus-Boelter correlation is modified to account for secondary flow effects

$Nu = CR_e^m P_r^n$ ---(Eq.4)

Where C, m, n and P are Empirical constants based on experimental data. For corrugated tubes, enhanced turbulence due to surface roughness modifies the exponent values in the correlation, leading to increased heat transfer rates [17,18].

2.5 Pressure Drop Analysis

The pressure drop in a helical coil heat exchanger can be estimated using the Darcy-Weisbach equation $\nabla P = f\ \frac{L}{D}\ \frac{\rho v^2}{2}$ where f is the friction factor. Empirical correlations account for curvature effects and secondary flow influences on pressure drop.

2.6 Corrugation Effect on Nusselt Number:

Use a modified Dittus-Boelter correlation:

$Nu = C\ Re^m\ Pr^n \left(\frac{d}{D}\right)^{\gamma} \left(\frac{e}{d}\right)^{\beta}$---(Eq. 6) where e is the corrugated height and γ and β is an empirical exponent determined by experimentally. Corrugation effect on Friction Factor (f): Modify the Darcy-Weisbach equation using an empirical relation

$$f = 0.079Re^{0.0250}\left(\frac{e}{d}\right)^{\lambda} \text{ ---(Eq.5)}$$

Where λ accounts for roughness effects. The Python program was developed to analyze the performance of both corrugated and plain helically coiled tubes with varying inner and outer diameter Fluid density ρ = 1000 (kg/m^3) ,Velocity v = 2 (m/s) , Dynamic viscosity μ= 0.001 (Pa.s), Specific heat capacity Cp = 4180 (J/kg.K), Thermal conductivity k = 0.6 (W/m.K), Empirical constant C = 0.023 , Pipe length L = 10 (m), Roughness height e = 0.0005(m), Empirical exponent for corrugation β = 0.1, Roughness effect exponent λ= 0.2.

3. Numerical Studies

To complement the experimental work, numerical simulations were carried out using Computational Fluid Dynamics (CFD) in ANSYS Fluent based on the Finite Volume Method (FVM). The 3D geometry of helically coiled tubes, both with and without corrugations, was modelled using CATIA V5 and then imported into the ANSYS Workbench for pre-processing, meshing, and simulation, as shown in Fig. 1.

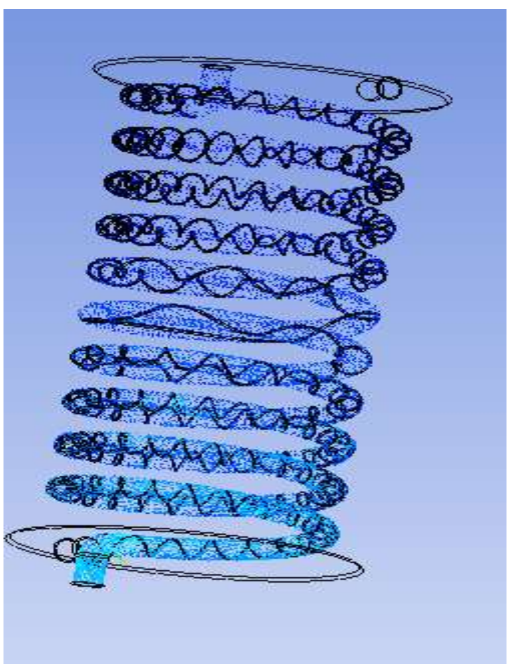

Fig. 1. 3D Geometric Models of Helically Coiled Tubes with and without Corrugation

The helically coiled tube had an inner diameter of 9.5 mm and a wall thickness of 1.5 mm. The coil diameter (D) varied as 0.1 m, 0.12 m, 0.14 m, and 0.16 m, while the total tube length (L) was kept constant at 0.71 m. The pitch (Pcor = 5mm) and corrugation depth (dcor = 5mm) were maintained constant for all models. This parametric variation was aimed at evaluating the impact of the Dean number on thermal and hydraulic performance. The inlet and outlet sections were designed as straight, non-corrugated extensions, to allow for fully developed flow conditions and eliminate entrance/exit effects.

3.1 Mesh Generation and Grid Independence

An unstructured tetrahedral mesh was generated for all geometries using ANSYS Meshing. Mesh refinement was applied near the wall regions and corrugation surfaces to accurately capture boundary layer behavior. An aspect ratio of approximately 1:1.1 was maintained near curved regions to ensure mesh quality.

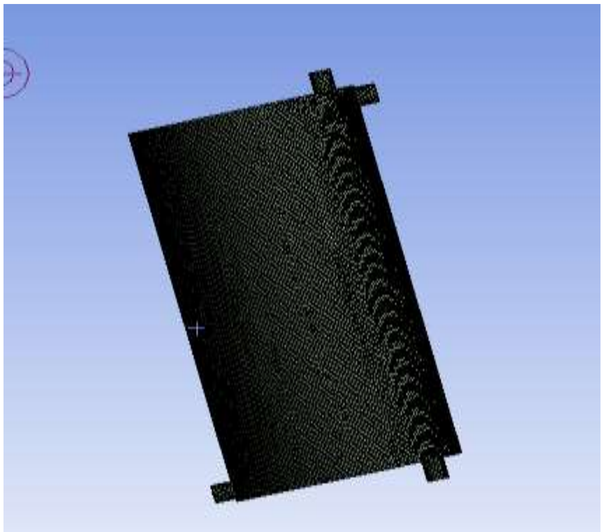

Fig. 2. Unstructured Tetrahedral Mesh of Corrugated Helical Coil Geometry

A grid independence study was performed by varying the total number of mesh elements and comparing key output parameters such as temperature, velocity components (x, y, z), thermal conductivity, and energy. As illustrated in Fig. 2 and Fig. 3, the results showed a variation of less than 5% beyond a critical mesh size, confirming mesh independence.

3.2 Boundary Conditions and Solver Settings

The boundary conditions were defined as follows:

Inlet: Uniform velocity profile applied in all three directions (x, y, z), with constant inlet temperature and a turbulence intensity of 4%.

Outlet: Zero-gauge pressure was imposed; all other variables were set to outflow conditions.

Tube Inner Wall: No-slip condition applied; constant heat flux of 10 kW/m^2 was applied on the inner surface.

Tube Outer Wall: Treated as adiabatic, implying zero temperature gradient.

Turbulence Model: Standard k–ε model was employed with standard wall functions to resolve near-wall behavior.

3.3 Convergence study:

Convergence history validation Fig. 3 ensure the reliability of numerical results. The residuals of governing equations (continuity, momentum, energy, and turbulence) were monitored during iterations and were reduced by at least three orders of magnitude before accepting a solution as converged. In addition to residuals, global parameters such as mass flow balance, pressure drop, and heat transfer rate were also tracked until they attained steady values with negligible fluctuations.

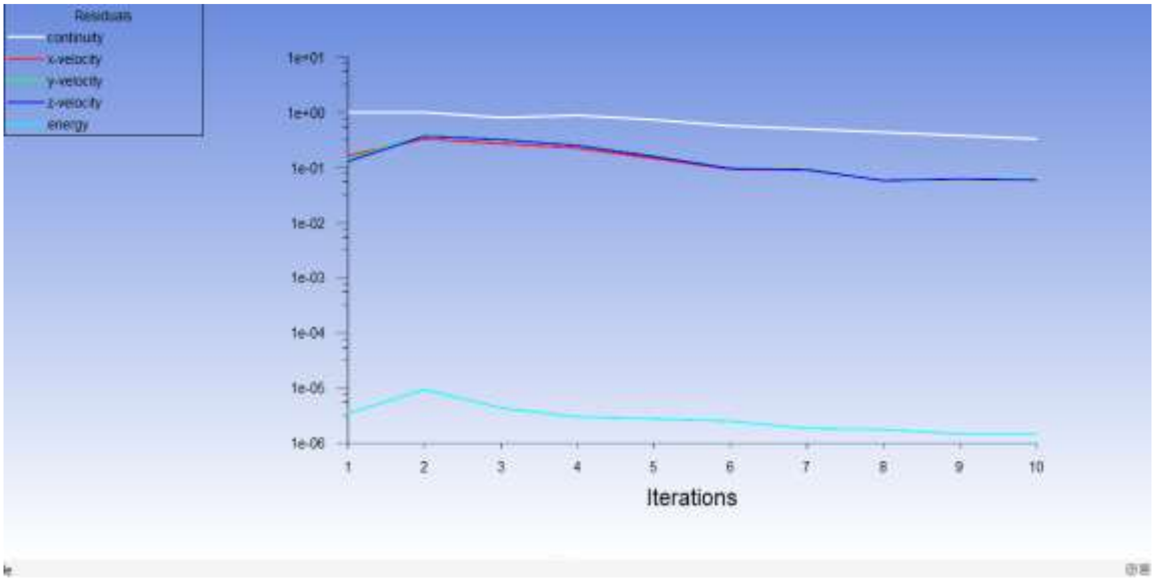

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Fig. 3. Residual Convergence History for Grid Independence Validation in Corrugated and Non-Corrugated 3D Helical Coil Models

4. Result and Discussion

This section presents the findings from the mathematical, numerical investigations aimed at understanding the effect of varying coil diameter on the thermal and hydraulic performance of corrugated helical coil heat exchangers. While the corrugation geometry specifically the pitch and depth were maintained constant throughout all configurations, four different coil diameters were considered to examine their influence on the Dean number and Colburn factor. The analysis includes comparative plots and data showcasing trends in heat transfer coefficient, pressure drop, and overall thermal performance. The results highlight how increasing coil diameter affects secondary flow development, leading to significant variations in heat transfer enhancement and pressure drop characteristics.

4.1 Mathematical results

The mathematical model, coded in Python, generated the thermal-hydraulic performance graphs presented in Fig. 4. The study investigated helically coiled tube heat exchangers with varying coil geometries specifically tube diameters of D = 0.10 m, 0.12 m, 0.14 m, and 0.16 m—under both smooth and corrugated surface conditions. As shown in Fig. 4 (a), the Reynolds number increases with tube diameter due to the larger cross-sectional area and increased flow velocity at a constant mass flow rate. Fig. 4 (b) depicts the De, which exhibits a near-linear increase with tube diameter, reflecting stronger curvature-induced secondary flow effects in larger coils. The Prandtl number, a thermophysical property of the working fluid (R134a), remains nearly constant across all cases, as shown in Fig. 4 (c), indicating that the fluid maintains stable viscous and thermal behavior throughout.

In Fig. 4 (d), the Nu is markedly higher for corrugated tubes particularly at D = 0.10 m due to intensified turbulence and increased heat transfer surface area, which significantly enhances convective heat transfer. This trend is further supported by the Colburn factor, presented in Fig. 4 (e), where corrugated geometries consistently outperform smooth ones by approximately 20–35%. Similarly, the Stanton number, shown in Fig. 4 (f), follows the same enhancement pattern, reinforcing the overall heat transfer augmentation. The heat transfer coefficient, as illustrated in Fig. 4(g), is highest for corrugated tubes at smaller diameters, underscoring the impact of surface modification on thermal performance.

However, these thermal benefits are accompanied by increased pressure drops, particularly in smaller diameter coils. Although the D = 0.10 m corrugated tube delivers superior heat transfer, the D = 0.12 m corrugated configuration offers a more favourable balance between enhanced thermal performance and manageable pressure loss, making it a more viable option for practical heat exchanger applications.

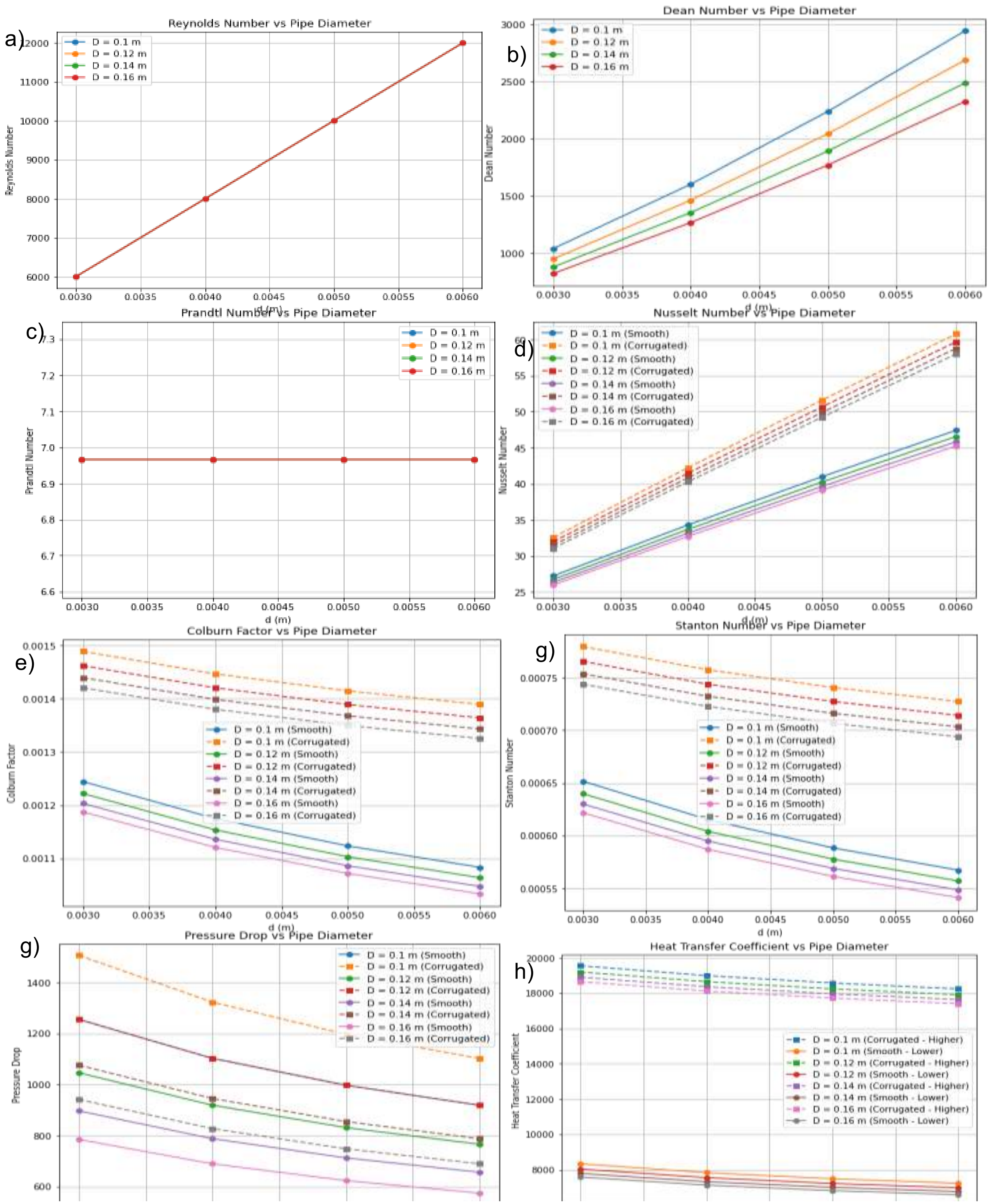

Fig. 4: Influence of Corrugation, Tube Diameter, and Coil Diameter on (a) Reynolds Number, (b) Dean Number, (c) Prandtl Number, (d) Nusselt Number, (e) Colburn Factor, (f) Stanton Number, (g) Pressure Drop, and (h) Heat Transfer Coefficient in a Helically Coiled Tube Heat Exchanger.

4.2 Numerical Results

The CFD simulation image illustrates a comparative thermal performance analysis between **smooth and corrugated helical tubes**, a key area of interest in enhancing heat exchanger efficiency. These simulations were specifically generated to assess the impact of surface geometry on fluid dynamics and heat transfer behaviour within helically coiled systems, which are widely used in various thermal and energy systems.

The top two panels (Fig. 5 (a and b) and the bottom two panels (Fig. 5(c and d)) present the CFD results from different viewing angles, offering a three-dimensional visualization of temperature distribution and fluid flow behaviour. The smooth helical tube, depicted in panels Fig.5(a) and Fig. 5(c), demonstrates relatively **laminar flow characteristics**. The coloured contours represent temperature gradients, where red indicates high temperatures and blue represents cooler regions. The **blue streamlines** represent the fluid velocity profiles inside the tube. In the case of the smooth helical tube, the streamlines are more orderly, indicating less turbulence and slower mixing. The thermal boundary layer is noticeably thicker, and the temperature gradients develop more gradually along the tube's surface. This relatively stable flow limits the convective heat transfer rate, resulting in **moderate thermal performance**.

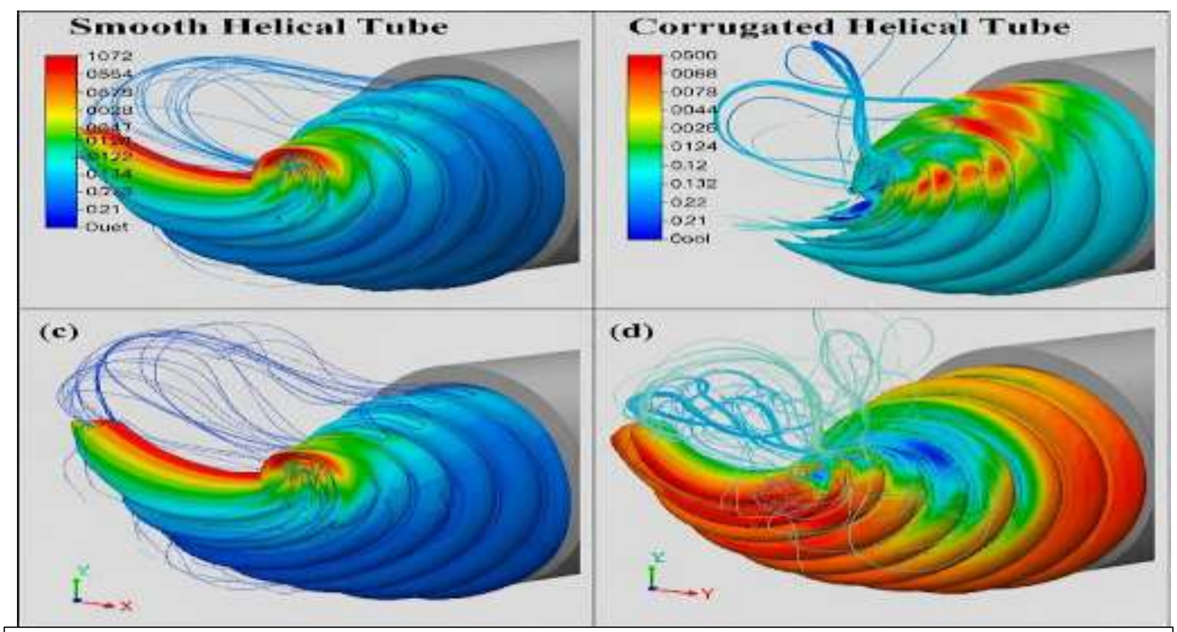

Fig. 5 CFD simulation comparing smooth and corrugated helical tubes. Panels (a, c) show smoother flow and thicker thermal boundary layers, while panels (b, d) display enhanced turbulence due to corrugations. Increased mixing in corrugated tubes leads to improved heat transfer performance and more uniform temperature distribution.

Conversely, the panels representing the **corrugated helical tube** (Fig.5 (b) and Fig.5(d)) reveal a contrasting flow behaviour. The introduction of surface corrugations leads to more **complex and chaotic streamline patterns**, indicative of **enhanced turbulence** within the tube. These corrugations act as flow disruptors, breaking the boundary layer and promoting **secondary flows and vortices**. This dynamic mixing intensifies the interaction between the hot and cold fluid layers, significantly reducing the thermal boundary layer's thickness and enabling faster heat dissipation. As a result, the temperature distribution across the tube wall becomes more uniform, indicating superior heat transfer effectiveness.

From a thermal-hydraulic perspective, the corrugated geometry demonstrates significant advantages. Despite a potential rise in pressure drop due to increased friction, the corresponding **increase in heat transfer coefficient** and **Nusselt number** outweighs this penalty. This makes corrugated helical tubes an optimal choice for compact and high-performance heat exchangers, especially where space constraints and thermal efficiency are critical. The **corrugated helical tube** clearly outperforms the smooth tube in terms of both **temperature uniformity** and **fluid mixing**, which are essential for maximizing the effectiveness of heat exchangers in applications such as HVAC systems, refrigeration units, automotive thermal management, and energy recovery systems. This insight is crucial for the design of next-generation heat exchanger components that demand **high efficiency, compactness, and reliability** in various industrial sectors.

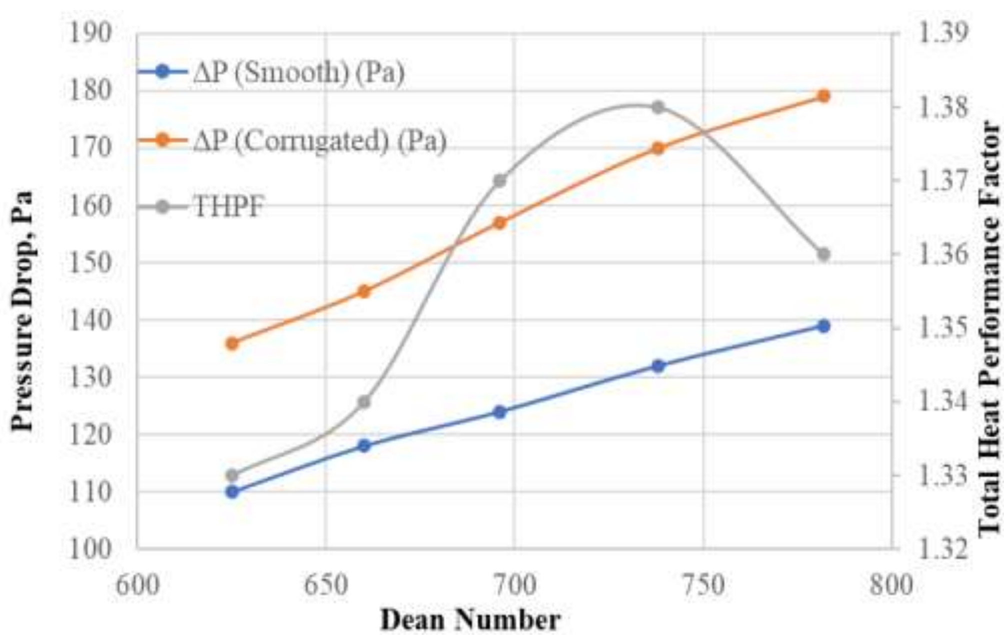

Fig. 6 Variation of Pressure Drop and Total Heat Performance Factor with Dean Number for Smooth and Corrugated Helical Coils

The variation of pressure drop (ΔP) and THPF with respect to the De is illustrated in **Fig. 6**. As expected, the pressure drop increases with increasing De for both smooth and corrugated coils due to higher flow velocity and the intensification of centrifugal forces in the helical geometry.

At **De = 625**, the pressure drop in the smooth coil was approximately **110 Pa**, while the corrugated coil showed a higher-pressure drop of around **135 Pa**, indicating an increase of about **22.7%.** This trend continued with increasing De, and at **De = 782**, the pressure drops were observed to be **138 Pa** and **179 Pa** for smooth and corrugated coils, respectively.

This higher pressure drop in the corrugated coil is attributed to enhanced surface roughness and flow disruption caused by the corrugation, which introduces additional frictional resistance and vortex formation. Despite this increase in pressure loss, the **THPF**, which balances both heat transfer enhancement and hydraulic penalty—remained consistently above **1.34** throughout the De range, indicating net thermal performance gains.

The THPF increased with De, peaking at around **1.38** near **De = 734**, before slightly declining. This suggests an **optimal De** where the heat transfer gain is maximized relative to the pressure penalty. Beyond this point, the rise in pressure drop may outweigh the incremental benefits in heat transfer. Thus, corrugated helical coils, despite incurring higher pressure losses, deliver superior thermal performance overall, especially at intermediate De where the THPF reaches its maximum.

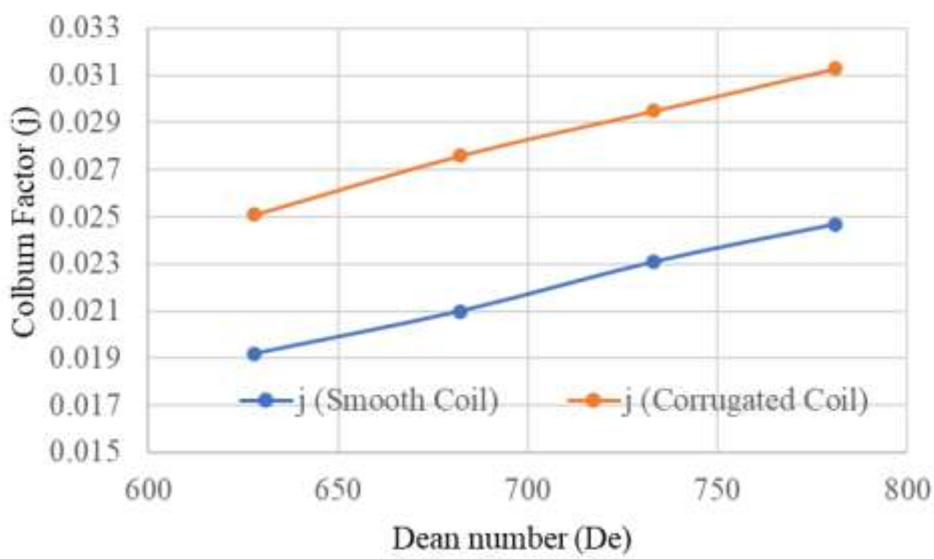

Fig. 7 Variation of Colburn Factor (j) with Dean Number (De) for Smooth and Corrugated Coils

The Fig. 7 shows the variation of the Colburn factor (j) with De for both smooth and corrugated helical coils. The De number ranged from approximately 620 to 790, covering moderate flow regimes. It is observed that the Colburn factor increases with an increase in De for both smooth and corrugated coils. This is due to the enhanced centrifugal effects and secondary flow induced by the coil curvature, which promote better mixing and heat transfer. The corrugated coil consistently exhibits higher Colburn factor values than the smooth coil at all De numbers. This clearly shows that the corrugation enhances the heat transfer performance by disturbing the thermal boundary layer and promoting turbulence at lower Reynolds numbers. Hence, the experimental findings of this study align well with the existing literature and further validate the advantages of corrugated geometries in compact heat exchanger design. Shokouhamand and Kashani reported that corrugated tubes enhance the Colburn *j*-factor by approximately 20–35%[27]. Similarly, studies by Lei Sun *et al.* [28]and Jafar *et al.[29]* also demonstrated that the Colburn *j*-factor increases with both De and surface corrugation.

Pressure Drop Characteristics

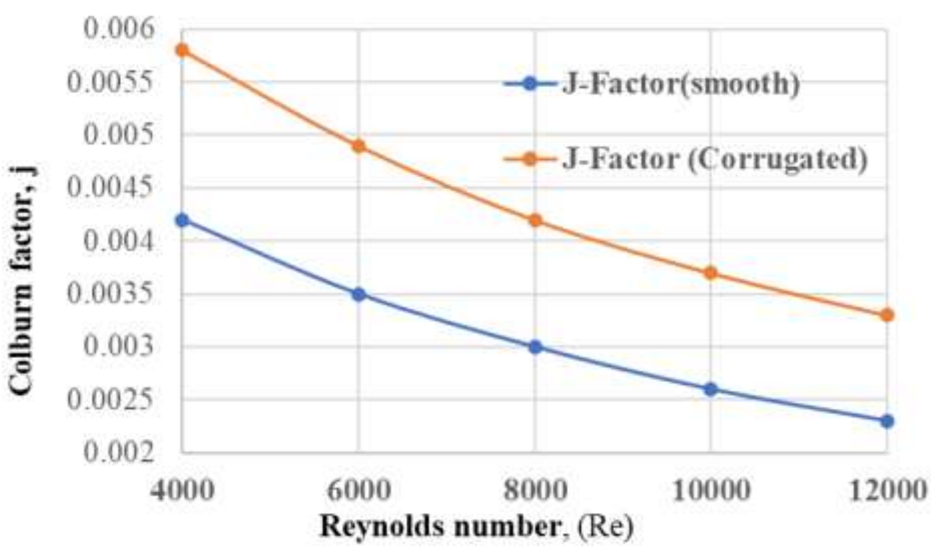

Fig. 8 Variation of Colburn factor (j) with Reynolds number for smooth and corrugated condenser tubes.

Fig. 8 presents the variation of the Colburn factor (j) with Re for both smooth and corrugated condenser tubes. The Colburn factor is a key dimensionless parameter that characterizes convective heat transfer performance, and its trend helps evaluate the thermal effectiveness of the tube surfaces. It can be observed that for both smooth and corrugated tubes, the Colburn factor decreases with increasing Reynolds number. This trend is consistent with conventional heat transfer theory, where the decrease in the Colburn factor is attributed to the dominance of inertial forces at higher Reynolds numbers, reducing the relative enhancement due to surface geometry. However, the Colburn factor for corrugated tubes consistently remains higher than that of smooth tubes across all Reynolds numbers considered (4000 to 12000). This indicates a superior heat transfer performance of the corrugated tube, which can be attributed to increased surface area and the presence of secondary flow effects induced by the corrugations. These geometrical modifications enhance fluid mixing and disrupt the thermal boundary layer, thereby improving convective heat transfer.

At Re = 4000, the Colburn factor for the smooth tube is approximately 0.0041, while for the corrugated tube, it reaches around 0.0058 showing an enhancement of nearly 41%. As Reynolds number increases to 12000, the values drop to approximately 0.0023 for the smooth tube and 0.0032 for the corrugated tube, but the enhancement remains evident. This sustained higher performance by the corrugated design implies that the geometrical perturbations maintain their heat transfer advantage even at higher flow rates. The results clearly demonstrate that incorporating corrugations into tube surfaces significantly enhances thermal performance, especially beneficial in applications where compactness and heat transfer efficiency are critical, such as in condensers of vapor compression refrigeration systems.

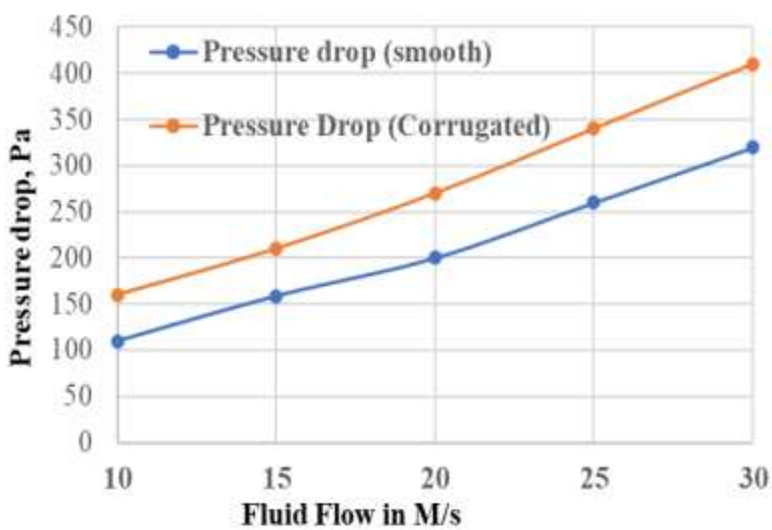

Fig. 9 Variation of Pressure Drop with Fluid Flow Rate for Smooth and Corrugated Tubes

Fig. 9 illustrates the relationship between pressure drop and fluid flow rate for both smooth and corrugated tubes. The pressure drop is plotted on the vertical axis (in Pascals), while the fluid flow velocity is represented on the horizontal axis (in meters per second). The experimental data shows a clear trend: as the fluid flow rate increases from 10 m/s to 30 m/s, the pressure drop also increases in both tube configurations. However, the rate of increase is more significant in the case of the corrugated tube compared to the smooth tube.

CoP Comparison

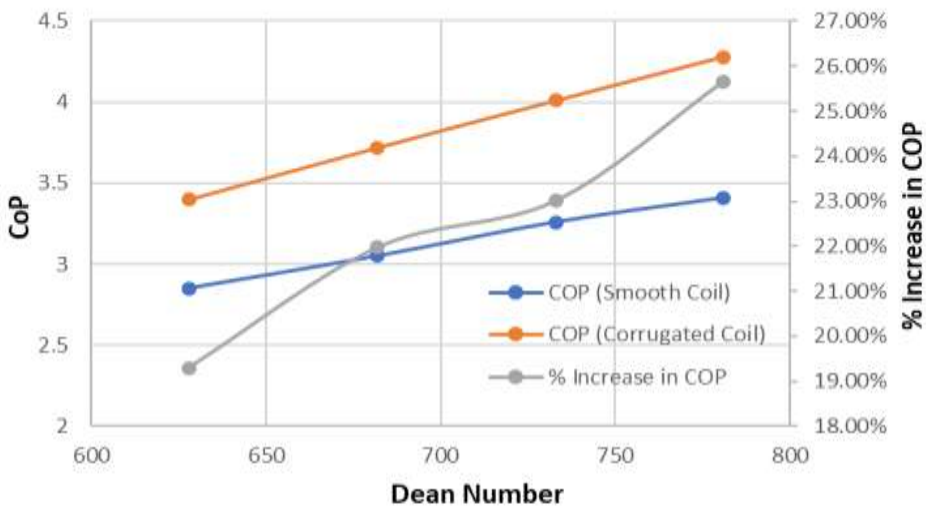

Fig. 10. Variation of Coefficient of Performance (COP) with Dean Number for Smooth and Corrugated Coils and Percentage Increase in

Fig. 10 illustrates the variation in the COP of a VCRS with respect to De for both smooth and corrugated coil evaporators. Additionally, the graph depicts the percentage increase in COP when a corrugated coil is used in place of a smooth coil. The De, which accounts for both the Re and curvature effects, is a critical non-dimensional parameter in curved tube flow and significantly influences the heat transfer characteristics in the evaporator coils.

From the graph, it is evident that the COP increases with an increase in De for both types of coils. However, the rate of increase is more pronounced in the case of corrugated coils. For instance, at a De of approximately 620, the COP for the smooth coil is around 2.85, while it reaches about 3.4 for the corrugated coil. As the De number increases to 780, the COP values rise to about 3.4 and 4.3 for smooth and corrugated coils, respectively. This consistent enhancement in performance with increasing De number can be attributed to improved secondary flow and turbulence induced by the corrugated geometry, which enhances the convective heat transfer rate within the evaporator.

The percentage increase in COP, shown by the grey line, also shows an increasing trend with De number, starting at about 19% and reaching approximately 26%. This clearly demonstrates the thermal performance advantage of corrugated coils over smooth coils. The higher turbulence generated in corrugated coils leads to better refrigerant-side heat transfer, reduced thermal resistance, and improved refrigerant distribution, all contributing to enhanced system efficiency. Thus, the adoption of corrugated evaporator coils, particularly in systems operating at higher Dean numbers, proves to be a viable approach for increasing the thermal performance and energy efficiency of refrigeration systems.

Validation of Models

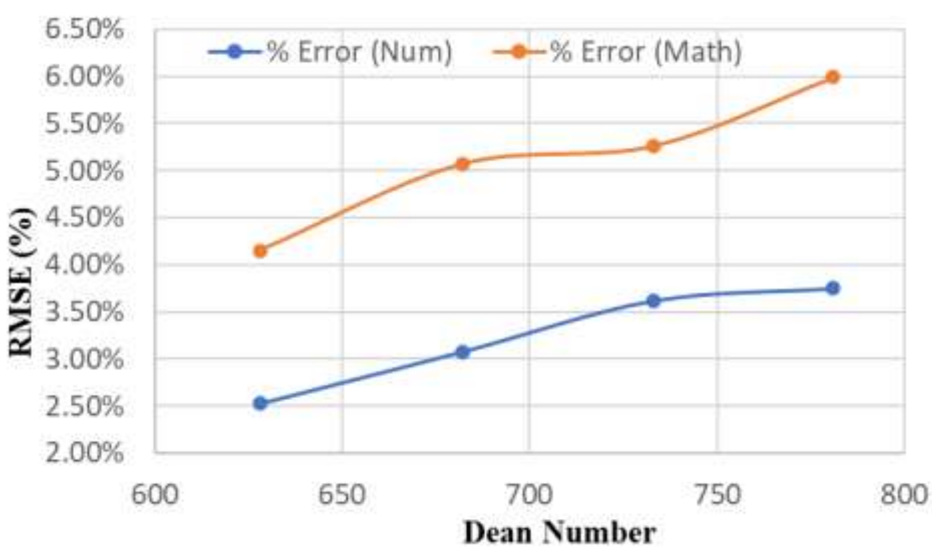

Fig. 11 : Variation of Root Mean Square Error (RMSE) percentage for numerical and mathematical models with respect to Dean Number.

Fig. 11 shows the comparative analysis of the RMSE percentage between the numerical and mathematical models over a range of De numbers, specifically from 625 to 785. The De number, a dimensionless quantity characterizing flow in curved tubes, influences the secondary flow intensity and consequently the convective heat transfer characteristics. As observed from the figure, both numerical and mathematical models show an increasing trend in RMSE with rising

De number. For the numerical model, the RMSE begins at approximately 2.5% at De number 625 and increases steadily, reaching around 3.75% at Dean number 785. In contrast, the mathematical model exhibits a higher initial RMSE of approximately 4.15%, which continues to rise to nearly 6.0% at the highest De number studied.

This consistent increase in RMSE indicates that both models encounter increasing predictive error as the De number grows. However, the numerical model maintains significantly lower RMSE values throughout the entire range, suggesting higher accuracy and better alignment with reference or experimental data compared to the mathematical model. The higher RMSE in the mathematical model could be attributed to simplifications or assumptions made in its derivation, which may not adequately capture the complex flow behaviours at higher De numbers. The graph validates the superiority of the numerical model in terms of prediction accuracy, especially at higher De numbers, and highlights the importance of robust computational modelling in capturing flow characteristics in curved geometries.

5. Conclusion

Helical coil heat exchangers are widely recognized for their compact design and superior heat transfer characteristics compared to straight-tube configurations. Their inherent curvature induces secondary flows, which enhance fluid mixing and improve heat transfer performance. In recent years, researchers have explored various geometric modifications to further improve the thermal efficiency of helical coils, with corrugations emerging as a particularly effective strategy. The integration of corrugations into helical coils intensifies turbulence and secondary flow structures, thereby significantly enhancing heat transfer capability. Experimental and numerical analyses have revealed a strong dependence of the Nusselt number and Colburn factor on the Dean number, highlighting the critical role of curvature-induced flow instability in dictating thermal behavior. The study demonstrated that the Thermal Hydraulic Performance Factor (THPF) reached a maximum of 1.38 at a Dean number of approximately 734, indicating an optimal operating range for corrugated coils. Although the adoption of corrugations leads to a 22–27% increase in pressure drop, the overall heat transfer gain substantially outweighs the frictional penalties. These findings suggest that corrugated helical coils represent a promising advancement in heat exchanger design, offering a balanced trade-off between pressure drop and enhanced thermal efficiency, thereby making them highly suitable for modern energy-intensive applications.

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