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Numerical Investigation of the Effect of Pitch Variation on the Flow Field, Pressure Distribution, and Heat Transfer Coefficient in a Vertical Double-Pipe Helical Heat Exchanger

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Abstract

Helical double-pipe heat exchangers have been widely investigated for their compact design, small volume, high heat-exchange area, and the potential to generate secondary flows due to helical curvature. However, changes in the heat exchanger geometry, including the helical pitch, might simultaneously affect the flow structure, pressure distribution, and heat transfer. In this study, a three-dimensional Computational Fluid Dynamics (CFD) analysis is performed to investigate the impact of helical pitch on the hydrodynamic and heat-transfer characteristics of a vertical double-pipe helical heat exchanger. The equations of continuity, momentum, and energy for a Newtonian incompressible fluid are solved numerically using a finite-volume approach implemented in ANSYS FLUENT. The effect of three different helical pitches of 151, 191, and 231 mm is analyzed. At the same time, all the other geometric and operational parameters are kept constant in order to focus on the effect of pitch modification. It is found that a decrease in the helical pitch increases the heat transfer coefficient along the helical flow channel. The main cause of this is the higher number of turns within the same axial distance and the enhanced curvature-driven secondary flow, which facilitate fluid mixing and reduce the thermal boundary layer thickness.

On the other hand, a decrease in pitch affects the pressure distribution, increasing hydrodynamic resistance due to more intense secondary flow. Hence, although a smaller pitch is preferable for improved heat transfer, the pitch cannot be selected solely on this basis. It is necessary to conduct a combined thermo-hydraulic analysis of heat-transfer improvement, pressure drop, and pump power demand to determine the most appropriate pitch value.

Keywords: Double-pipe helical heat exchanger, Helical pitch, Heat-transfer coefficient, Pressure distribution, Secondary flow, CFD, Thermo-hydraulic performance.

1 | Introduction

The growing worldwide demand for energy sources, the scarcity of fossil fuels, and the pressing need to reduce energy consumption and greenhouse gas emissions have underscored the importance of research on highly efficient, compact heat exchangers. As an important component of energy conversion processes and

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industry, heat exchangers facilitate the recovery and transfer of thermal energy. Helical heat exchangers have attracted considerable interest from investigators due to their compact design, large specific heat-transfer surface area, and ability to enhance fluid mixing. This explains the application of helical heat exchangers in various fields, including the chemical and petrochemical industries, power plants, heating, ventilation, and air conditioning systems, food and pharmaceutical industries, nuclear systems, and waste-heat recovery processes [1–4].

Secondary flow is a key feature of flow in helical tube bundles, arising from the geometry of the flow passage. Flow in a curved channel generates centrifugal forces that alter the velocity profile in the transverse direction and create secondary vortices. Secondary flows affect hydrodynamic and thermal boundary layers and can enhance mixing and heat transfer rate. At the same time, the intensification of secondary motion usually increases the pressure drop and power input to the system. Thus, when assessing the performance of a helical heat exchanger, the balance between heat-transfer enhancement and the corresponding hydrodynamic penalty needs to be considered. The initial work on the subject belongs to Eustice, Dean, White, and others, who provided basic physical insight into flow in curved tubes and its dependence on secondary motion [5–7]. In particular, Dean developed the famous Dean number, which characterizes the effect of tube curvature and flow regime on the intensity of secondary flow.

In double-pipe helical heat exchangers, several geometrical parameters, such as tube diameter, coil diameter, curvature ratio, coil pitch, and number of turns, could significantly influence the flow structure and heat transfer processes. Among all the listed parameters, coil pitch plays a crucial role, as variations in it affect the number of coil turns per unit length, the spacing between consecutive turns, and the flow structure within and around the helical channel. As a result, variations in pitch can modify the velocity, pressure, intensity of secondary flows, and temperature fields, leading to changes in the heat transfer coefficient and pressure drop.

Numerical and experimental investigations revealed the role of the coil pitch in determining the thermal performance of helical systems. Shin et al. [8] experimentally studied the natural convection heat transfer from a helical coil with varying coil pitch and number of turns. Their results showed that the effect of pitch on heat transfer is not necessarily monotonic and depends on the interaction among pitch, number of turns, and flow structure. Optimal ranges for the pitch-to-coil-diameter ratio were identified, highlighting the importance of pitch choice in coil design. Recently, investigations of the thermal-hydraulic performance of helical heat exchangers have been widely conducted numerically. Ghazanfari et al. [9] numerically analyzed the thermal-hydraulic performance of a helical coil heat exchanger with twisted-tube geometries. The effects of Reynolds number, pitch length, and tube orientation on heat transfer and pressure drop were studied. The results of this work revealed that pitch length simultaneously influences the Nusselt number and hydraulic resistance, indicating that the choice of an adequate pitch must be based on thermal-hydraulic performance criteria. In another recent study, Kirkar et al. [10] investigated the thermal and flow characteristics of helically coiled tubes with modified surfaces. Numerical results demonstrated that the curvature of tubes and surface modifications intensify the secondary flows and, hence, the heat transfer performance. At the same time, the intensification of heat transfer was accompanied by an increase in frictional resistance and pressure drop. The above findings demonstrate the need to evaluate helical heat exchangers using thermal-hydraulic performance indicators.

A recent review of heat-transfer augmentation methods for helically coiled tube heat exchangers was carried out by Rahman [4]. It was shown that geometrical modifications, internal inserts, fins, and baffles are some of the methods used to enhance heat transfer in these devices. However, one of the major problems in the design of helical heat exchangers is the accompanying rise in pressure drop due to increased fluid mixing and secondary flow. Thus, the design of configurations that strike a balance between heat-transfer improvement and hydraulic penalty remains a crucial issue.

The optimization of geometrical parameters in helical heat exchangers has also been discussed recently. Huang et al. [11] used response surface methodology in combination with a multi-objective genetic algorithm to optimize the performance of a double-helical coil heat exchanger. Several geometrical parameters, such as inner and outer coil pitches, coil diameter, and tube diameter, were chosen as design variables, and the overall heat transfer coefficient and pressure drop were used as performance parameters. The results show that the simultaneous optimization of geometrical parameters yields better overall heat exchanger performance with a lower hydraulic penalty.

In a more recent study, Alklaibi et al. [12] examined the effects of geometry and nanofluids on the performance of double-pipe helical-coil heat exchangers. Several designs were examined, varying pitch ratio and coil curvature. It was found that geometric modifications can substantially increase the heat transfer coefficient; however, this increase can come at the expense of increased pumping power. These findings demonstrate the importance of accounting for the flow field, heat transfer, and pressure drop in double-pipe helical heat exchangers.

In addition to the above-mentioned work, other recent studies have shown that the thermal-hydraulic performance of helical heat exchangers is not dependent on a single geometric parameter. For example, a recent paper on a helically coiled tube heat exchanger for underground applications demonstrated that coil diameter can influence thermal performance and that variations in geometrical parameters can change the Nusselt number and thermal-hydraulic performance index [13]. Also, some papers on multi-helical coil heat exchangers have shown that the arrangement and geometry of the coils can significantly influence flow distribution and thermal-hydraulic performance [14].

Despite significant achievements in recent years, several issues related to flow and heat transfer in double-pipe helical heat exchangers remain. Most of the literature on this subject focuses on average values such as the Nusselt number, heat transfer coefficient, and pressure drop. However, Analysis of the pressure field, velocity field, and secondary-flow structure would provide more insight into how coil pitch influences heat-transfer performance. In addition, recent research shows that the influence of coil pitch on heat transfer performance is not always monotonic and depends on several factors, such as geometrical configuration, curvature ratio, flow regime, and thermal boundary conditions [9-13]. Thus, assuming a universal correlation between heat transfer performance and coil pitch, without considering other parameters, may lead to misinterpretation of the results.

Moreover, most of the previously published papers were devoted to shell-and-coil heat exchangers or to modified geometrical configurations, and studies dedicated to the influence of coil pitch in double-pipe helical heat exchangers and to the Analysis of the pressure field, velocity field, and heat transfer characteristics are scarce. Such a gap in research is essential, as the modification of coil pitch influences not only the heat-transfer surface area but also the secondary-flow structure and pressure distribution. Thus, detailed investigation of this problem will provide valuable information in the future.

Thus, the key aim of the present work is to conduct a numerical investigation into the effects of coil pitch on the velocity and pressure fields, and the heat transfer properties, of a vertical double-pipe helical heat exchanger. The Analysis will consider different coil pitch configurations, and the results will be compared with existing benchmark data. At the same time, because the current work concerns both the velocity and pressure fields and the heat transfer coefficient, it enables a more comprehensive analysis of the physical processes underlying the effect of coil pitch on heat-exchanger operation. The results will help define appropriate pitch values and provide recommendations for the thermal-hydraulic design and optimization of double-pipe helical heat exchangers.

2 | Geometry and Flow Characterization

Fig. 1 presents a schematic representation of the double-pipe helical heat exchanger investigated in the present study. The main geometrical parameters of the heat exchanger include the inner diameter of the inner tube ($\mathrm{D_i}$), outer diameter of the inner tube ($\mathrm{D_o}$), mean coil diameter ($\mathrm{D_C}$), and helix pitch (p). The helix pitch is defined as the axial distance between two consecutive turns of the helical coil. The geometrical configuration of the heat exchanger is designed to investigate the influence of helix pitch on the thermal and hydrodynamic characteristics of the flow.

The inner and outer diameters of the inner tube were maintained at 87 and 151 mm, respectively, while the mean coil diameter was fixed at 471.8 mm. Three different helix pitches, namely 151, 191, and 231 mm, were considered. Except for the helix pitch, all other geometric and operating parameters were kept constant to isolate its effect on heat transfer and flow characteristics. It should be noted that the helix pitch and helix angle are distinct geometrical parameters, although they are directly related. A change in pitch modifies the helix angle and the number of turns over a given axial length, thereby affecting the curvature-induced centrifugal forces and secondary flow structures.

Contrary to the straight tube flow case, the flow in the helically coiled passage is significantly affected by the centrifugal forces arising due to the curvatures of the flow passage. Thus, apart from the Reynolds number, the Dean number is introduced to describe the flow characteristics in the helical passage. The Reynolds number is expressed through the relation.

$$\mathrm{Re} = \frac{\mathrm{VD}}{\nu}, \tag{1}$$

where (V) stands for the mean fluid velocity; (D) is the characteristic diameter of the flow passage; and (ν) denotes the kinematic viscosity. The Reynolds numbers for the inner tube and annular passage are respectively calculated according to the equations

$$\mathrm{Re_i} = \frac{\mathrm{V_i D_i}}{\nu_\mathrm{i}}, \tag{2}$$

$$\mathrm{Re_a} = \frac{\mathrm{V_a D_{h,a}}}{\nu_\mathrm{a}}, \tag{3}$$

where the subscript (i) corresponds to the inner tube, whereas the subscript (a) indicates the annular passage; and ($\mathrm{D_{h,a}}$) is the hydraulic diameter of the annular passage. The Dean number quantifies the effect of the centrifugal forces that appear owing to the curvature of the helical flow passage. The Dean number for the inner tube is expressed through the relation

$$\mathrm{De_i} = \mathrm{Re_i}\sqrt{\frac{\mathrm{D_i}}{\mathrm{D_c}}}, \tag{4}$$

While for the annular passage it takes the form

$$\mathrm{De_a} = \mathrm{Re_a}\sqrt{\frac{\mathrm{D_{h,a}}}{\mathrm{D_c}}}. \tag{5}$$

An increasing Dean number tends to intensify the secondary flow due to the centrifugal force. As a result, changes in the velocity and temperature fields could improve convective heat transfer, but an increase in the pressure drop accompanies this improvement. Hence, both thermal and hydraulic parameters should be considered when estimating the effect of the helix pitch.

The volumetric flow rates of the working fluids have been set to 0.006 m^3/s for all configurations considered. The mean velocities for the flow in each passage were estimated using the specified flow rates and the cross-sectional areas of each passage. The obtained Reynolds and Dean numbers have been used to identify the flow regime for each configuration.

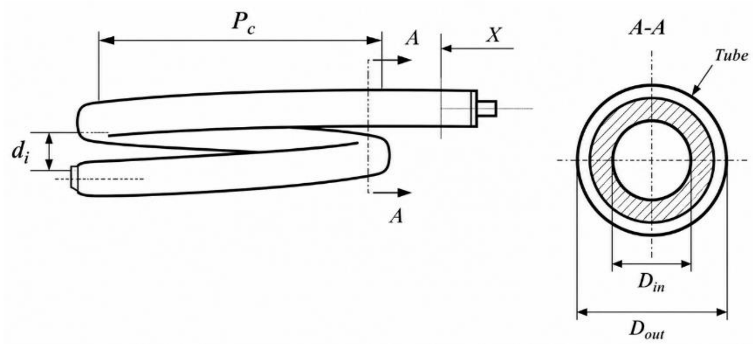


Fig. 1. Schematic representation of the double-pipe helical heat exchanger and definition of the main geometrical parameters.

3 | Computational Fluid Dynamics Modeling

A three-dimensional Computational Fluid Dynamics (CFD) modeling approach was used to study the thermal and hydrodynamic behavior of the double-pipe helical heat exchanger. The computational geometry was developed from the geometric parameters outlined in Section 2 and implemented in ANSYS FLUENT. Computational domains for the inner tube and the annular passage were created separately, followed by the creation of the computational mesh. Geometries and grids used for different helix pitches are shown in *Fig. 2* below.

Due to the curvature of the helical flow passages, a sufficiently fine mesh was used to accurately predict the velocity and temperature gradients, especially near the walls. Grid independence analysis was carried out on several computational meshes with different numbers of cells under the same operating conditions. Heat transfer rate, pressure drop, and average heat transfer coefficient were chosen as parameters for the grid sensitivity analysis. When additional grid refinement yielded negligible changes in the above parameters, the mesh was considered grid-independent and used in all subsequent simulations to ensure computational accuracy without unnecessary computational effort.

The governing equations for fluid flow and heat transfer were solved using the finite-volume approach in ANSYS FLUENT. Conservation of mass, momentum, and energy was considered in the computational domain. For an incompressible Newtonian fluid, the continuity equation can be written as

$$\nabla \cdot \mathrm{u} = 0, \tag{6}$$

and the momentum equation as

$$\rho(\mathrm{u} \cdot \nabla)\mathrm{u} - \nabla \mathrm{p} + \mu \nabla^2 \mathrm{u}. \tag{7}$$

Here, ϱ, p, and μ are the density, pressure, and dynamic viscosity of the fluid. The energy equation is given as:

$$\rho c_p(\mathrm{u} \cdot \nabla \mathrm{T}) \nabla \cdot (\mathrm{k} \nabla \mathrm{T}). \tag{8}$$

The simulations were performed under steady-state flow conditions. A volumetric flow rate of 0.006 m^3/s was assumed at the inlet of each fluid path. The inlet temperatures were assigned based on the heat exchanger's operating conditions, whereas outlet conditions were considered at the respective outlets. The no-slip condition was considered at all solid-fluid interfaces. Heat transfer through the dividing wall between the two working fluids was accounted for by introducing a thermally coupled interface; thus, the energy flux from the hot fluid to the wall could be consistently transmitted to the cold fluid.

The solution to the mathematical problem was obtained via pressure-velocity coupling and spatial discretization schemes in ANSYS FLUENT. The [SIMPLE/SIMPLEC/PISO] method was used for pressure-velocity coupling, while the second-order upwind scheme was used for discretization of the convective terms. The convergence of the numerical simulation was analyzed using residual values of the differential equations and the stability of key engineering parameters, such as outlet temperatures, heat transfer rate, and pressure drop. The same numerical schemes, criteria of convergence, and operating conditions were used for all three helix-pitch arrangements.

To assess the reliability of the numerical model, the current CFD results were compared with existing experimental and/or numerical data reported in the literature under similar geometric and operating conditions. The validation was performed for typical thermal and hydraulic characteristics, such as Nusselt number, heat transfer coefficient, and pressure drop. The relative deviation between the current results and reference data was evaluated as

$$\mathrm{Error}(\%) = \left|\frac{X_{present}-X_{ref}}{X_{ref}}\right| \times 100, \quad (9)$$

where ($X_{present}$) and (X_{ref}) are the current numerical result and the corresponding reference value, respectively. The satisfactory correlation of the current predictions with the reference data demonstrates the reliability of the developed CFD model.

After the numerical model has been verified and validated, the influence of the helix pitch on flow and heat-transfer characteristics has been systematically studied. For this purpose, the velocity profile, temperature profile, pressure drop, heat transfer rate, average heat transfer coefficient, and Nusselt number have been considered. Because all geometric and operational parameters, except for the helix pitch, remained constant throughout the study, the variations observed across all three geometries can be attributed mainly to the influence of the helix pitch on secondary flow and heat transfer.

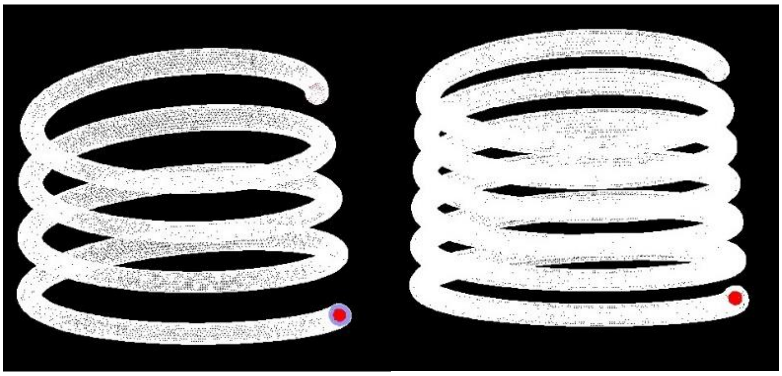


Fig. 2. Computational geometry and mesh of the double-pipe helical heat exchanger for different helix pitches.

4 | Results and Discussion

The accuracy and reliability of the numerical model are validated by comparing the numerical results with the experimental data provided by Prabhanjan et al. [15]. The heat transfer coefficient, one of the most important parameters for the thermal performance of the heat exchanger, is shown in *Fig. 3*. The figure shows that the numerical results match the experimental data well. Thus, the good agreement between the results confirms the ability of the present numerical model to predict the thermal performance of the fluid flow in the helical heat exchanger and provides a basis for studying the influence of geometric parameters on the results.

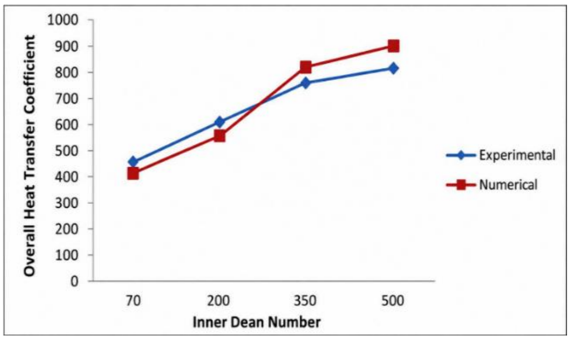


Fig. 3. Comparison of the numerical and experimental results.

The effect of the helical pitch on the pressure distribution and heat transfer coefficient along the tube was examined further. Three values of the helical pitch of 231, 191, and 151 mm were used in this study. The pressure contour for each case is shown in *Fig. 6*. From *Fig. 4*, the pressure distribution along the helical tube is nonuniform and changes significantly as the fluid moves through the curved flow channel. This non-uniformity is caused by the combined effects of wall friction and the centrifugal force arising from the curvature of the flow channel. In the helical tube, the curvature of the flow generates a radial pressure gradient, increasing the pressure on the outer side of the coil relative to the inner side. With the decrease of the helical pitch, the number of turns per unit length increases.

Variation of the wall heat transfer coefficient along the helical tube for three different pitch values is shown in *Fig. 5*. The heat transfer coefficient varies spatially along the flow path due to flow development, changes in the thermal boundary layer, and secondary flow caused by the tube curvature. In all three cases, there is a noticeable local increase of the heat transfer coefficient near the outlet section. However, one should be careful when interpreting such an increase, as it may be influenced by the outlet boundary conditions and the flow's development in the vicinity of the outlet. Hence, the local maximum observed near the outlet section cannot be directly interpreted as an increase in the overall heat transfer performance of the heat exchanger (*Fig. 6*).

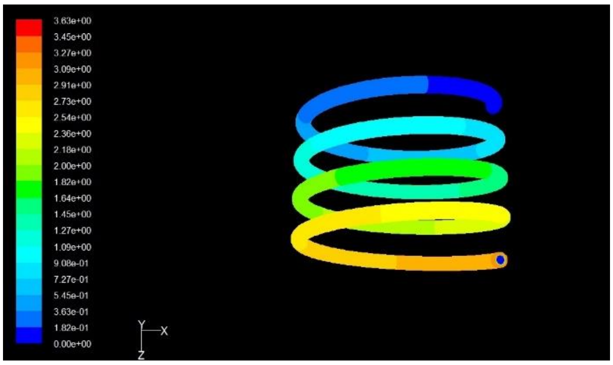


Fig. 4. Pressure contour along the tube for a pitch of 231 mm.

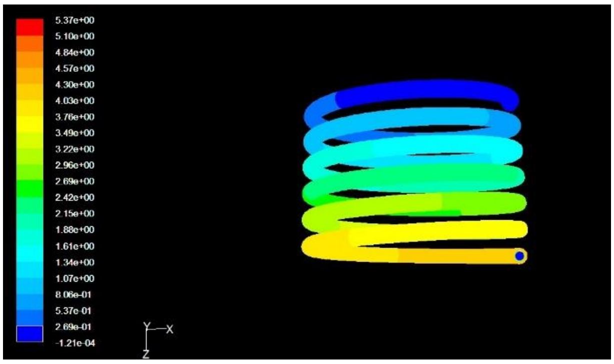


Fig. 5. Pressure contour along the tube for a pitch of 191 mm.

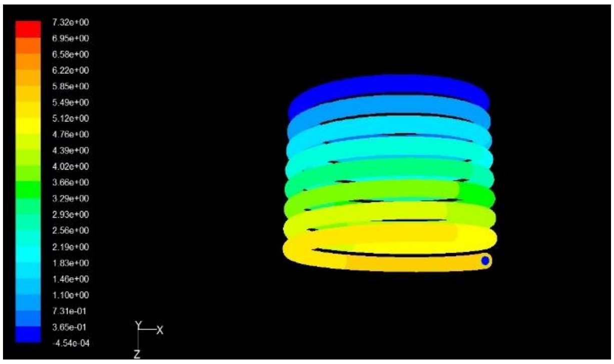
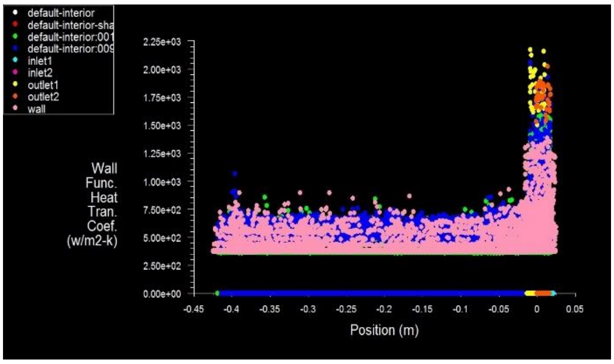


Fig. 6. Pressure contour along the tube for a pitch of 151 mm.

A comparison of the three cases suggests that decreasing the helical pitch from 231 mm to 191 mm, then to 151 mm, increases the heat transfer coefficient along the helical tube. Reduction of the pitch increases the number of coil turns within a certain length and, therefore, increases the intensity of the secondary flow caused by tube curvature. Enhanced mixing of the flow reduces the thickness of the thermal boundary layer near the wall, thereby improving heat transfer.

Thus, a reduction in the helical pitch can be considered a positive factor for the heat exchanger's heat transfer performance. Nevertheless, optimization of the helical pitch should take into account both the heat transfer coefficient and the related pressure drop and pumping power. Consequently, the optimum helical pitch should be evaluated using comparative Analysis of the average heat transfer coefficient, total pressure drop, and, preferably, the thermo-hydraulic performance criterion for the three considered geometries (*Figs. 7-9*).

Fig. 7. Variation of the heat transfer coefficient along



the tube for a pitch of 231 mm.

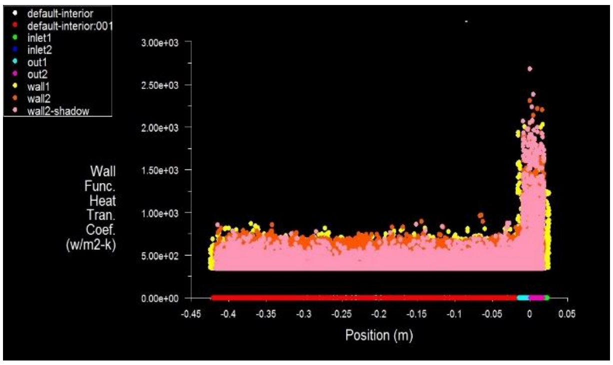


Fig. 8. Variation of the heat transfer coefficient along the tube for a pitch of 191 mm.

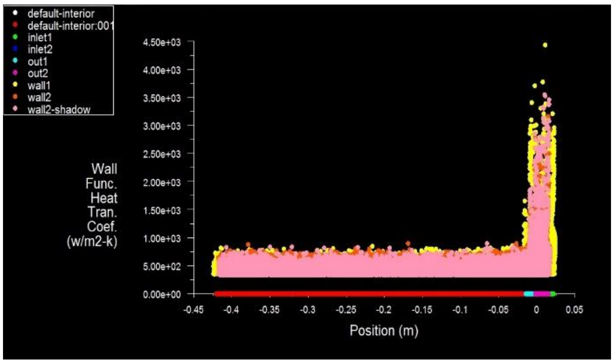


Fig. 9. Variation of the heat transfer coefficient along the tube for a pitch of 151 mm.

5 | Conclusion

A three-dimensional CFD analysis has been carried out to investigate the effects of helical pitch on the flow structure, pressure distribution, and heat transfer properties of a vertical double-pipe helical heat exchanger. In this study, three pitches of 151, 191, and 231 mm have been considered for investigation, while the remaining geometric and operational parameters are held fixed. From the numerical results, it can be concluded that:

I. Pitch significantly affects the flow structure and secondary motion. A reduction in the helical pitch increases the number of turns along a certain axial distance and therefore the effects of centrifugal forces. As a result, secondary flow becomes more pronounced, leading to enhanced transverse mixing.

II. The curvature of the helical passage affects the pressure field. The pressure distribution in the helical passage is spatially nonuniform due to the effects of friction at the wall and centrifugal forces caused by the curvature. In particular, a radial pressure gradient is created, resulting in higher pressure on the outer side of the passage than on the inner side. Therefore, it is not sufficient to characterize the hydrodynamics of a helical heat exchanger by means of only the axial frictional losses.

III. A decrease in the pitch leads to better heat transfer. The results of this work show that reducing the pitch from 231 to 191 mm, and further to 151 mm, increases the heat transfer coefficient. The mechanism behind this improvement lies in a stronger secondary flow and better mixing, which promote the redistribution of momentum and thermal energy and, therefore, a reduction in the effective thermal boundary-layer thickness near the heat-transfer surface.

IV. Better heat transfer goes hand in hand with an increased hydraulic penalty. While a reduction in pitch positively influences heat transfer, the enhanced secondary flow and curvature effects modify the pressure distribution and increase hydraulic resistance. Therefore, maximizing the heat transfer coefficient does not necessarily imply an optimal heat exchanger configuration.

V. Thermo-hydraulic considerations should be taken into account when selecting the pitch. The results presented in this study clearly show that pitch selection should be based on the average heat transfer coefficient, Nusselt number, pressure drop, and pumping power requirements.

VI. The proposed numerical approach enables a prediction of the heat transfer characteristics. The comparison between the predicted heat transfer coefficient and the experimentally obtained results by Prabhanjan et al. indicates satisfactory agreement, confirming the ability of the developed CFD model to predict the basic thermal features of the helical heat exchanger and to analyze the influence of geometric changes.

To summarize, the current findings reveal that the helical pitch plays a significant role in the heat and mass transfer behavior of vertical double-pipe helical heat exchangers. Smaller pitch leads to better heat transfer performance due to secondary flows, which, however, increase resistance and pumping power requirements. Therefore, the optimal design of the heat exchanger is to be found through a compromise between thermal performance and the penalties associated with hydraulic resistance and pumping power, rather than by seeking purely maximal heat transfer. Future studies should be conducted based on the presented Analysis with respect to various ratios of pitch to coil diameter, Reynolds and Dean numbers, and operating conditions, including a thermo-hydraulic performance indicator.

Conflict of Interest

The authors certify that they have no affiliations with or involvement in any organization or entity with any financial or non-financial interest in the subject matter discussed in this manuscript.

Data Availability

All data relevant to this study are contained within the manuscript. Additional information is available from the corresponding author upon reasonable request.

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