Paper ID: 362

Enhanced Heat Transfer in Double Pipe Heat Exchanger Using Helical Fins: Simulation and Optimization

Rakibul Islam Kanak[1], Iltimas Wasek[1]
[1]Department of Mechanical Engineering,
Bangladesh University of Engineering and Technology, Dhaka
E-mail: iltiwashek007@gmail.com

Abstract

This study focuses on enhancing the efficiency of a double pipe heat exchanger by incorporating helical fins on the outer surface of the inner tube. The DPHX was designed to facilitate thermal exchange between hot and cold fluids, with the outer tube providing insulation for operational stability. Using the Finite Volume Method (FVM) in ANSYS Fluent, the effects of fin pitch, thickness, and height on heat transfer and pressure drop were analyzed. Initial validation of the simulation model was performed using a finless DPHX configuration, after which simulations with helical fins were conducted. Results showed that, for constant fin thickness and height, a fin pitch of 25 mm was optimal. For constant height and pitch, a fin thickness of 2 mm yielded the best performance. Additionally, increasing fin thickness from 1 mm to 2 mm resulted in an optimal fin height reduction from 6.5 mm to 5 mm, balancing added surface area with minimal flow obstruction. These optimized configurations achieved significant improvements in thermal efficiency for the helical finned DPHX.
Keywords: double pipe, helical fins, heat transfer enhancement, fin geometry optimization

1. Introduction and Literature Review

Research on double-pipe heat exchangers has primarily centered on forced convection, a process where fluid flow is induced by an external force [1], minimizing the effect of buoyancy forces. The double-pipe heat exchanger consists of two distinct regions: the tube side and the annulus side, and is widely used in various industries such as automotive, aerospace, HVAC, and renewable energy systems. Notable early studies include Kakac et al. [2], who used empirical formulas to analyze forced convection in laminar flow. The introduction of helical fins on the annulus side has proven effective in boosting heat transfer rates. Sivalakshmi [3] highlighted that these fins are particularly beneficial at high Reynolds numbers, while Maakoul et al. [4] found through numerical analysis that a pitch of 0.1 m achieved optimal results. A. Mule et al. [5] identified a 20° helix angle as the most efficient for heat transfer enhancement, and Salem et al. [6] noted that a relative fin height of 0.667 and a pitch of 0.333 maximized performance. For non-Newtonian fluids, Mozafarie and Javaherdeh [7] reported that a fin pitch of 0.25 mm provided optimal heat transfer. Despite extensive research, there is still a lack of numerical studies on the impact of fin count and comprehensive fin geometry on annulus-side heat transfer, which this study aims to address. Although the experimental studies mentioned above primarily examined the effects of individual fin parameters, no comprehensive analysis has been conducted on the combined influence of fin height, thickness, and pitch. Testing all possible combinations of these parameters experimentally would be economically unfeasible. To address this gap, this study uses numerical simulations to assess 36 different fin geometry combinations and identify the most optimal design for a helical fin double-pipe heat exchanger.

3. Nomenclature

h_{cold}	Annular side heat transfer coefficient (cold fluid)
Q_{avg}	Average heat transfer rate
Q_{hot}	Heat transfer rate at the pipe side
Q_{cold}	Heat transfer rate at the annulus side
$\dot{m}_{hot}$	Mass flow rate of hot fluid
$\dot{m}_{cold}$	Mass flow rate of cold fluid
$C_{p.hot}$	Specific heat capacity of hot fluid
$C_{p.cold}$	Specific heat capacity of cold fluid
$T_{hot.in}$	Inlet temperature of the hot fluid
$T_{hot.out}$	Outlet temperature of the hot fluid
$T_{cold.in}$	Inlet temperature of the cold fluid
$T_{cold.out}$	Outlet temperature of the hot fluid
T_{cold}	Average temperature of the cold fluid
Nu_{cold}	Nusselt number of the cold fluid at annular side
L_{char}	Characteristic length of the annular space (m)
Gz	Graetz Number
μ_b	Bulk viscosity (Pa-s)
μ_w	Wall viscosity (Pa-s)
f	Friction factor
ξ_{corr}	Correlation factor
De_p	Equivalent Diameter of inner pipe (m)
Dh_p	Hydraulic Diameter of inner pipe (m)
ID_p	Inner Diameter of the inner pipe (m)
OD_p	Outer Diameter of the inner pipe (m)
ΔP	Pressure Drop (Pa)
Dh_a	Hydraulic diameter of the annular pipe
ID_a	Inner diameter of the Annulus pipe

4. Modelling

4.1. Geometry

This research aims to analyze the impact of varying helical fin geometries on heat transfer and pressure drop in the annulus of a double-pipe heat exchanger. A 3D-CFD model, using Ansys Fluent, simulates fluid flow under steady-state laminar conditions. Water is used as both the hot and cold fluid, with temperature-dependent thermal properties.

Key dimensions of the heat exchanger include:

- Inner pipe diameter: 28 mm
- Outer pipe diameter: 30 mm
- Annulus inner diameter: 50 mm
- Annulus outer diameter: 52 mm

Table 1. Range of Values of Geometric Parameters

Geometric Parameters	Values
Fin Thickness (mm)	1, 1.5, 2
Fin Height (mm)	5, 6.5, 8
Fin Pitch (mm)	25, 50, 75, 100

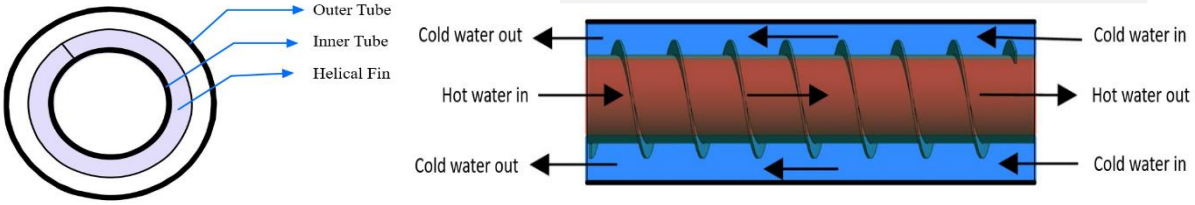


Figure 1: Cross section of the helical fin DPHX (left) and schematic diagram of the CFD model (right).

A total of 36 configurations were analyzed by varying these parameters. The helical fins induce rotational flow, enhancing heat transfer by modifying the velocity and pressure fields. Figure 2 provides an illustration of the fin geometry. All components, including the fins, are made of copper (k = 398 W/m·K).

4.2 Meshing

A tetrahedral mesh was applied to both the hot and cold fluids, while a cubic mesh was utilized for the outer pipe. The element size was set to 2e-3 m, with a total of 1,902,587 elements. Inflation layers were introduced along the hot and cold fluids to accurately capture heat transfer and fluid flow near the inner pipe's surface. The

mesh quality index and other error criteria are detailed in the following table, indicating that the overall mesh quality is sufficient to reliably model the CFD simulation.

Error Check	Quality Criterion	Warning (Target) Limit	Error (Failure) Limit	Worst
☐	Min Element Quality	Default (0.05)	Default (5e-04)	0.147
☐	Max Aspect Ratio	Default (5)	Default (1000)	14.893
☐	Min Orthogonal Quality	Default (0.05)	Default (5e-03)	0.155
☑	**Max Skewness**	**Default (0.9)**	Default (0.999)	0.845
☐	Min Tet Collapse	Default (0.1)	Default (1e-03)	0.195

Fig.2. Mesh Quality index

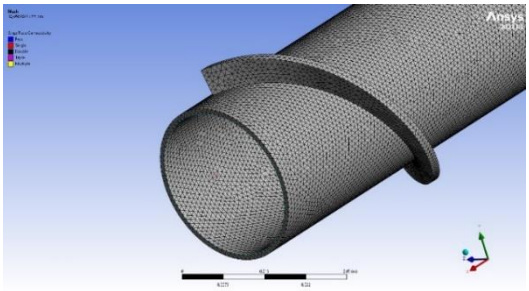


Fig.3. Mesh cross-section

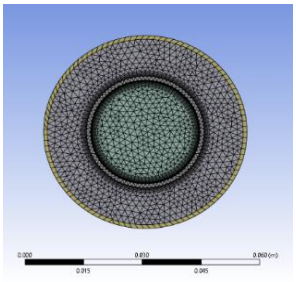


Fig.4. Inner pipe and fin mesh

4.3 Numerical Model:

To simulate the steady, incompressible and laminar flow of water in the annulus and inner tube of the double pipe heat exchanger studied in the present study, the conservation of mass, momentum and energy equations has been solved. These equations are as follows:

1. Continuity Equation: $$\frac{\partial \rho}{\partial t} + \nabla.(\rho \vec{v}) = 0 \quad (1)$$
2. Momentum Equation: $$\frac{\partial}{\partial t}(\rho \vec{v}) + \nabla.(\rho \vec{v}\vec{v}) = -\nabla p + \rho \vec{g} + \nabla.\hat{\tau} \quad (2)$$
3. Energy Equation: $$\frac{\partial}{\partial t}(\rho h) + \nabla.(\rho \vec{v} h) = \nabla.(k \nabla T) \quad (3)$$

These calculations were made subject to the boundary conditions shown in the following subsection.

4.4 Simulation Setup and Boundary Conditions:

The double-pipe heat exchanger features specified mass flow rates at the inlets (0.0022 kg/s at 80°C for the inner pipe and 0.0628 kg/s at 30°C for the outer pipe), outlet pressures, and no-slip conditions on the walls, with an adiabatic outer annular surface. This configuration yields Reynolds numbers of 100 for the hot fluid and 2600 for the cold fluid, optimizing thermal performance by balancing low and high Reynolds numbers to enhance efficiency. The simulation employed a steady-state, pressure-based solver with absolute velocity formulation, assuming laminar flow and ignoring gravity. Pressure-velocity coupling was achieved using the Rhie-Chow momentum-based flux. Spatial discretization utilized least squares cell-based gradients, second-order for pressure, and second-order upwind for momentum and energy. Residuals were set at 0.001 for velocity components and 1e-06 for energy, with 200 iterations and automatic time stepping using a time scale factor of 1.

4.5 Heat Transfer Coefficient, Heat Transfer Rate, Nusselt Number and Pressure Drop

To evaluate the thermal performance of the helical finned double-pipe heat exchanger and gain insights into the heat transfer mechanisms, critical parameters such as the heat transfer coefficient, heat transfer rate, and Nusselt number must be analyzed. The cold side (annular region) heat transfer coefficient is formulated as:

$$h_{cold} = \frac{Q_{avg}}{A_{total}(T_{wall} - T_{cold})} \quad (4)$$

Where, A_{total} is the total surface area covered by the helical fins and the unfinned portion of the outer surface of the inner pipe and Q_{avg} is the average heat transfer rate between hot water and cold water, which can be defined as follows:

$$Q_{avg} = \frac{Q_{hot} + Q_{cold}}{2} \tag{5}$$

The heat transfer rates from the heating fluid into cold fluid can be calculated by the following equations:

$$Q_{hot} = \dot{m}_{hot} C_{p.hot} (T_{hot.in} - T_{hot.out}) \tag{6}$$

$$Q_{cold} = \dot{m}_{cold} C_{p.cold} (T_{cold.out} - T_{cold.in}) \tag{7}$$

Once the average heat transfer rate, Q_{avg} , wall temperature T_{wall} and the cold fluid bulk temperature T_{cold} are defined:

$$T_{cold} = \frac{T_{cold.in} + T_{cold.out}}{2} \tag{8}$$

The convective heat transfer coefficient of the cold fluid at the annular region can be calculated as well as the Nusselt number (Nu) using the following equation:

$$Nu_{cold} = \frac{h_{cold} L_{char}}{k_{cold}} \tag{9}$$

4.6 Numerical Model Validation

To assess the thermal performance of the helical finned double-pipe heat exchanger, a CFD model for a conventional double-pipe heat exchanger was initially developed, adhering to the specified pipe dimensions. This foundational model, featuring a smooth configuration, was evaluated by calculating the Nusselt number and pressure drop across the annular region [8]. These values were then compared against established correlations typically used for smooth double-pipe heat exchangers. Given the relatively short length of the pipes (0.2 m), Nusselt number correlations and other relevant equations appropriate for developing flow in annular spaces were applied.

The correlations used in validating the smooth pipe CFD model are as follows:

- $Nu = 1.86(Gz)^{1/3} \left(\frac{\mu_b}{\mu_w} \right)^{0.14}$ (10)
- $Gz = \frac{\mathrm{Re}\,\mathrm{Pr}}{L/D}$ (For developing laminar flow inside annular space/tube) (11)
- Friction factor, $f = \xi_{corr} \left(\frac{64}{\mathrm{Re}} \right)$ (Laminar flow) (12)
- Pipe area
  - $De_p = Dh_p = ID_p$ (13)
  - $\Delta P_p = f_p \frac{L}{ID_p} \left(\frac{1}{2} \rho_p V_p^{\,2} \right), \xi_{corr} = 1$ (14)
- Annular area
  - $Dh_a = ID_a - OD_p$ (15)
  - $De_a = \frac{ID_a^{\,2} - OD_p^{\,2}}{OD_p}$ (16)

- $$\frac{1}{\xi_{corr}} = \frac{1+\kappa^2}{(1-\kappa)^2} + \frac{1+\kappa}{(1-\kappa)\ln(\kappa)}; \kappa = \frac{OD_p}{ID_a} \quad (17)$$
- $$\Delta P_a = (f_a \frac{L}{Dh_a} + 1)\left(\frac{1}{2}\rho_a V_a^2\right) \quad (18)$$

Using these equations and correlations the calculated Reynolds number (Re) is 2600, and the Nusselt number (Nu) for the annular cold fluid is approximately 223. In comparison, the CFD simulation results yield a Nusselt number of 263.15. Similarly, the pressure drop calculated via empirical correlations is 1.7 Pa, while the simulation gives a value of about 2 Pa. The minor deviation can be attributed to the use of bulk temperature in the hand calculations, which may fluctuate with iteration. Nevertheless, since both the Nusselt number and pressure drop align closely between the two methods, the results are considered accurate.

Further analysis compared the thermal performance and pressure drop of a smooth pipe with that of a pipe fitted with helical fins. For the smooth pipe, the outlet temperature of the hot fluid was 59.3°C. In contrast, using helical fins (e.g., 2 mm pitch, 5 mm fin height, and 1 mm fin thickness), the outlet temperature decreased to 57.68°C. However, the pressure drop with the helical fins was significantly higher due to the increased friction caused by the fins. These findings validate the accuracy of the CFD model in predicting both thermal performance and pressure drop for helical fin configurations.

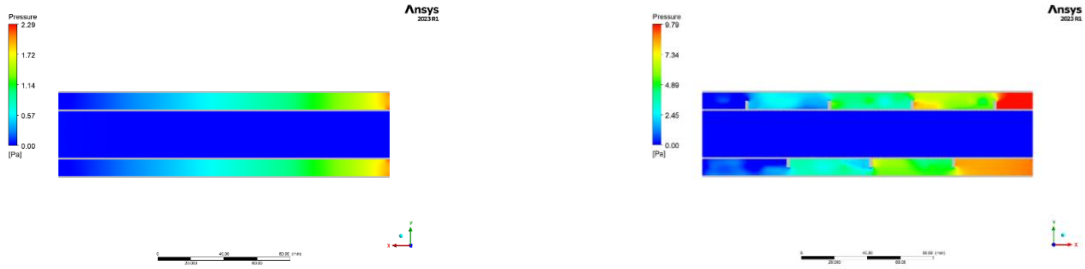


Fig 5: Pressure drop contour for smooth pipe (left) and 50mm pitch helical fin (right)

5. Results and Discussion

5.1 Fin parameters optimization through variations in fin height, fin thickness, and pitch length.

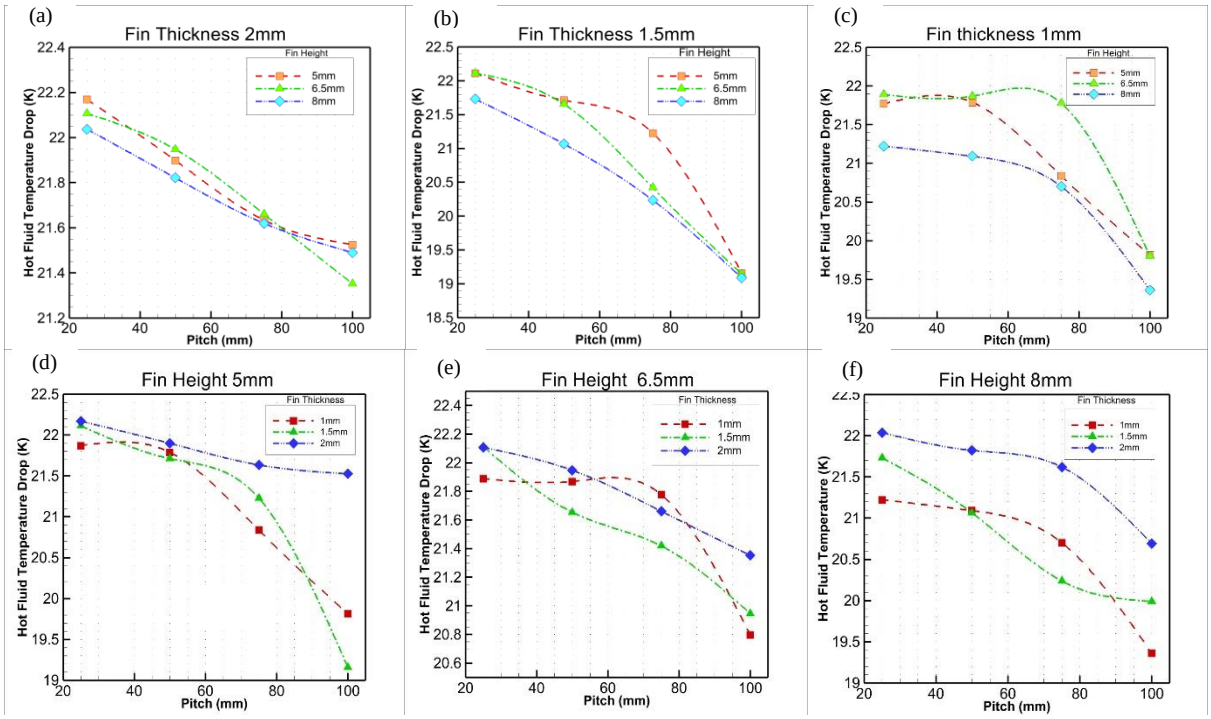


Fig.6. Variation of hot fluid temperature drop with pitch length for different fin heights, with fin thickness kept constant (a), (b), (c) and for different thickness, with fin height kept constant (d), (e), (f).

The simulation results show that decreasing the pitch of helical fins for a given thickness increases the simulation results indicate that while increasing fin height generally enhances the temperature drop, the optimal height for balanced performance is 6.5 mm among the tested heights of 5, 6.5, and 8 mm. At this height, the fins offer an effective balance between surface area and flow disruption, maximizing heat transfer without excessively hindering fluid flow. Shorter fins (less than 6.5 mm) lack sufficient surface area for optimal heat transfer, while taller fins (over 6.5 mm) can disrupt flow, induce stagnation, and decrease overall efficiency. The 6.5 mm fins generate moderate turbulence, which boosts convective heat transfer without causing flow separation or significant pressure drops. In contrast, taller fins may obstruct flow, and shorter fins may not generate sufficient turbulence for effective heat exchange.

Further analysis shows that, at a constant pitch of 25 mm, increasing fin thickness consistently results in higher temperature drops, with variations depending on fin height. For a 1 mm thickness, the highest temperature drop (21.89°C) is at a 6.5 mm fin height. A 1.5 mm thickness yields a maximum temperature drop of 22.11°C at the same height, while at 2 mm thickness, the greatest temperature drop (22.16°C) occurs at a 5 mm height. This trend suggests that increased fin thickness enhances heat transfer, even at shorter heights, by providing a larger conduction surface. The 25 mm pitch is thus optimal, and the interaction between fin height and thickness confirms that thicker fins maintain strong heat transfer performance at shorter heights.

5.2 Fin Surface Area: Pressure & Temperature Effects

The effective fin surface area significantly impacts the combined effects of fin geometry, as geometric changes directly influence it. Fig. 6 illustrates that as the heat transfer area increases, the cold fluid pressure drop rises sharply, indicating greater flow resistance, while the hot fluid's temperature drop increases more gradually. Enhancing the heat transfer area improves thermal performance but also results in higher pressure drops, which can reduce system efficiency. This underscores the need to balance heat transfer and pressure loss in fin design. Although the cold fluid's viscosity decreases with heat, larger heat transfer areas increase pressure drop due to heightened turbulence, friction, and tighter flow paths. These factors create greater flow resistance, emphasizing the trade-off between better heat transfer and increased flow resistance.

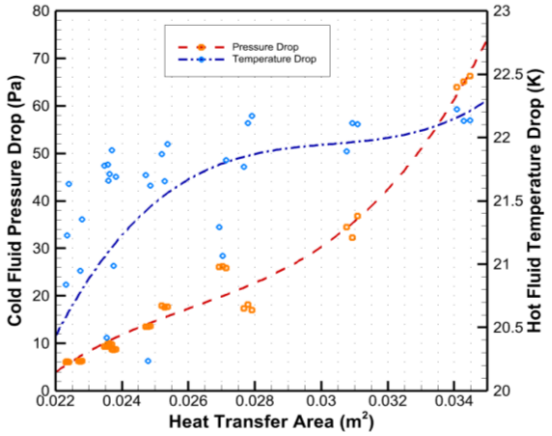


Fig.7. Effect of fin heat transfer area on Cold fluid pressure drop and hot fluid temperature drop

6. References

[1] S. S. Mousavi Ajarostaghi, A. Basem, K. W. Al-Mansoori, A. J. Sultan, M. Al-Yasiri, S. H. Hashemi Karouei, and H. Ahangaran, "Numerical evaluation of the impact of using spiral innovative turbulator on improving the thermal performance of a helical double-pipe heat exchanger," *International Journal of Thermofluids*, vol. 24, p. 100830, Aug. 2024.

[2] S. Kakac, H. Liu, and A. Pramuanjaroenkij, *Heat Exchangers: Selection, Rating, and Thermal Design*, 3rd ed. Boca Raton, FL, USA: CRC Press, 2012.

[3] S. Sivalakshmi, M. Raja, and G. Gowtham, "Effect of helical fins on the performance of a double pipe heat exchanger," *Materials Today: Proceedings*, Sep. 2020, doi: https://doi.org/10.1016/j.matpr.2020.08.563.

[4] M. Maakoul, A. Metoui, S. Abdellah, M. Saadeddine, and M. Meziane, "Numerical investigation of performance of thermohydraulic air to water double-pipe heat exchanger with helical fins," *Appl. Therm. Eng.*,vol. 127, pp. 127–139, 2017.

[5] B. A. Mule, D. N. Hatkar, and M. S. Bembde, "Analysis of bundle pipe heat exchanger with helical fins," *Int. Res. J. Eng. Technol. (IRJET)*, vol. 4, no. 8, Aug. 2017.

[6] M. R. Salem, M. B. Eltoukhey, R. K. Ali, and K. M. Elshazly, "Experimental investigation on the hydrothermal performance of a double-pipe heat exchanger using helical tape insert," *International Journal of Thermal Sciences*, vol. 124, pp. 496–507, Feb. 2018, doi: https://doi.org/10.1016/j.ijthermalsci.2017.10.040.

[7] S. S. Mozafarie and K. Javaherdeh, "Numerical design and heat transfer analysis of a non-Newtonian fluid flow for annulus with helical fins," *Engineering Science and Technology, an International Journal*, vol. 22, no. 4, pp. 1107–1115, Aug. 2019, doi: https://doi.org/10.1016/j.jestch.2019.03.001.

[8] ME307 charts equations