

# Numerical Investigations on Heat Transfer Characteristics in Double Tube Heat Exchanger with Variations in Interrupted Fin Height

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The current study investigates heat transfer characteristics in concentric tube heat exchangers, with interrupted fins in annulus side, exploring the use of both active and passive techniques. Interrupted fins are a variation of conventional longitudinal fins featuring interrupted surfaces that create a break in the boundary layer. This design provides a disrupted fluid passage along the length of the flow. The study employs numerical simulations and Taguchi techniques to investigate variation in fin heights in the annular region of a twin-tube heat exchanger and its effect on thermal characteristics under counter flow conditions. Three-dimensional computational fluid dynamics (CFD) simulations were performed under turbulent flow conditions for configurations with a fin length of 100mm; the height varied in steps of 2mm, 6mm, 10mm, and 14mm, with a constant interrupted length of 7mm. After obtaining the best height by simulation, a comparative analysis was done against the reference longitudinal fin of the same height on the same mass flow rate. On the other side, the thickness was kept constant at 4mm for all different heights. This was done because varying thicknesses have minimal impact on the heat transfer rate. The study also compared the Taguchi-derived optimal fin height with the results obtained from computational fluid dynamics, evaluating the accuracy of the analyses using statistical and numerical methods. A comparative analysis was done between the interrupted fin configurations and plain pipe, focusing on the enhancement of heat transfer, Nusselt number, and pressure drop. Heat transfer performance improved with an increase in fin height up to 10mm, after which thermal properties seemed to diminish. The results show that compared to a

plain pipe at the same mass flow rate, heat transfer is 35.39% to 97.56% higher for a fin height of 10mm and thermal enhancement factor of 2.54. The increase in friction factor is 1.34 to 1.56 times the plain pipe.

**Keywords:** Interrupted fins; Heights; Thermal performance factor; Pressure drop; Heat transfer rate.

## 1. Introduction

Heat exchangers are efficient devices for transferring thermal energy between fluids without mixing. They find applications in various industrial and technical processes such as boilers, water heaters, cooling, and heating systems. Despite the existence of various types of heat exchangers, twin-tube heat exchangers provide notable cost savings in terms of installation, maintenance, and design. Haya Hussein et al. [1] studied numerically and experimentally on double pipe heat exchanger with different fin arrangements like split longitudinal fins, staggered and inline fins, as well as split semi-helical fins with angles of 90°, 180°, and 270°. Among this they observed that a semi-helical fin with an angle of 90° is the most effective among the different fin arrangements considered. Huu-Quan, et al. [2] they studied the use of high-efficiency heat exchangers leads to cost and energy savings. Therefore, it is crucial from both scientific and engineering perspectives to continuously improve the performance of double-pipe heat exchangers. Omidi et al. [3] mentioned in their review paper about two main approaches that are typically taken to enhance heat exchanger performance: passive and active methods. The active method involves using external forces to increase the heat transfer rate, such as inducing flow or surface vibrations or creating flow disturbances using a magnetic field. In contrast, the passive method involves utilizing surface or geometry variations without applying external forces, such as incorporating extended surfaces, twisted tapes, turbulators, or wire coils. Morteza Mohammadi [4] employed response surface methods and numerical analysis to investigate a finned double pipe heat exchanger with six variables. He utilized the Face-Centered Central Composite Design (FCCD) technique to create 90 numerical designs, resulting in mathematical regression models of response functions. Iman Bashtani et al. [5] investigated fin turbulators inspired by the dorsal fins of dolphins. They studied various characteristics such as length scale, number of turbulators at different Reynolds numbers, and used an ANN approach to determine the value of  $R^2$ . The results indicate that simplifying the flow reduces friction loss. After conducting experimental and computational studies on the rotating flow within the axial clearance of a double-pipe heat exchanger, Fadime Simsek et al. [6] concluded that inducing rotational flow within the axial clearance is more effective than increasing the flow rate. They observed that rotational flow offers a superior heat transfer coefficient compared to irrotational flow. Lin Liu et al. [7] conducted an investigation into integrated internal longitudinal fin flow and heat parameters using experimental and numerical methods. They concluded that this configuration offers better heat transfer enhancement compared to standard fins. Mohammed Flyyih Hasan et al. [8] utilized the finite volume technique and a semi-implicit approach to solve the governing equations, including pressure-related equations. They found that adding fins to the outer surface of the inner tube improved the heat transfer rate, and the addition of nano particles to the base fluid enhanced the heat transfer coefficient. Abinab Nath et al. [9] found that incorporating a longitudinal fin arrangement in a double-tube heat exchanger

improved the Log Mean Temperature Difference (LMTD), overall efficiency, and heat transfer performance. Abdullah Hamzah et al. [10] investigated the placement of metal foam fins at regular intervals along the axial direction. They found that this configuration improved the heat transfer coefficient by 129% compared to not using metal foam fins. Osama A. Mohsen et al. [11] conducted an experimental investigation using various fin geometries in a double heat exchanger on the inner pipe. They employed interrupted rectangular fins, circular fins, and helical ribs. Their study revealed that rectangular fins maximized heat transfer improvement, while circular fins minimized it. Interestingly, the pressure drop showed the opposite trend under flow conditions. Dinar Susilo Wijayanto et al. [12] investigated the addition of four types of longitudinal fins to a cross-flow heat exchanger. They found that adding fins increased the Nusselt number and effectiveness. Anas El Maakoul et al. [13] investigated the use of split longitudinal fins compared to standard longitudinal fins. They observed that split fins create multiple surface breaks, disrupting the boundary layer and enabling fluid to flow along the entire length. In laminar flow conditions, the heat transfer rate was found to be 31–48% greater than that of longitudinal fins for the same power supply and unit weight. El Maakoul et al. [14] investigated continuous helical baffles with variable spacing ranging from 0.025 to 0.1m using FLUENT software. They examined these baffles in the annulus of a concentric tube heat exchanger with baffle spacings of 100mm, 50mm, 33.3mm, and 25mm. The study revealed that 25mm baffle spacing increased convective heat transfer by 45% but also led to a 21 times increase in pressure drop compared to the conventional one. Mosayebidorcheh et al. [15] considered rectangular, convex, triangular, and concave geometries in their study of transient thermal analysis on a conventional longitudinal fin with different cross-sections; they found that concave profiles were superior for maximizing heat transmission and minimizing fin temperature. Sharifi, K. et al. [16] utilized CFD techniques to investigate the effect of coiled wire inserts in DPHEs. They observed an improvement in the Nusselt number by about 1.77 times and proposed a correlation for the Nusselt number for different insert geometries under laminar flow conditions. Ishaq, M. et al. [17], proposed diamond-shaped augmented fins for heat exchangers, considering both triangular and rectangular designs. The study considers the placement of four and eight diamond-shaped fins to enhance heat transmission, specifically for radii ratios of 0.25. Wang, X. et al. [18], in their study, found that double shell rod baffle heat exchangers outperformed single rod baffle heat exchangers in terms of heat transfer pressure drop. M. Saeedan et al. [19] conducted a numerical investigation on the thermal performance of a 3D finned tube using nano fluids and a helically baffled heat exchanger. They found that as the volume concentration of Cu or CuO increases, the Nusselt number (Nu) rises. Hashemian, M. et al. [20] evaluated various design options for heat exchangers, considering the use of conical tubes instead of cylindrical tubes. At the optimal condition, the results indicate a 55% increase in effectiveness and a 40% improvement in the heat transfer enhancement number.

There is very few research work is carried out previously on the interrupted fin, There is no literature related to study of variation of height in split interrupted fin related to heat transfer, nusselt number and pressure drop. This study aims to evaluate heat transfer characteristics by changing the fin heights on the annulus side of a double-pipe heat exchanger. These adjustments affect the pressure and velocity distribution across the annulus

side, impacting both pressure drop and heat transfer rate. Fin heights vary in increments of 2mm, 6mm, 10mm, and 14mm, with a constant 4mm thickness. Taguchi method is used to estimate the optimum height, which is compared to CFD results. Numerical and statistical techniques are employed to assess the accuracy of these analyses.

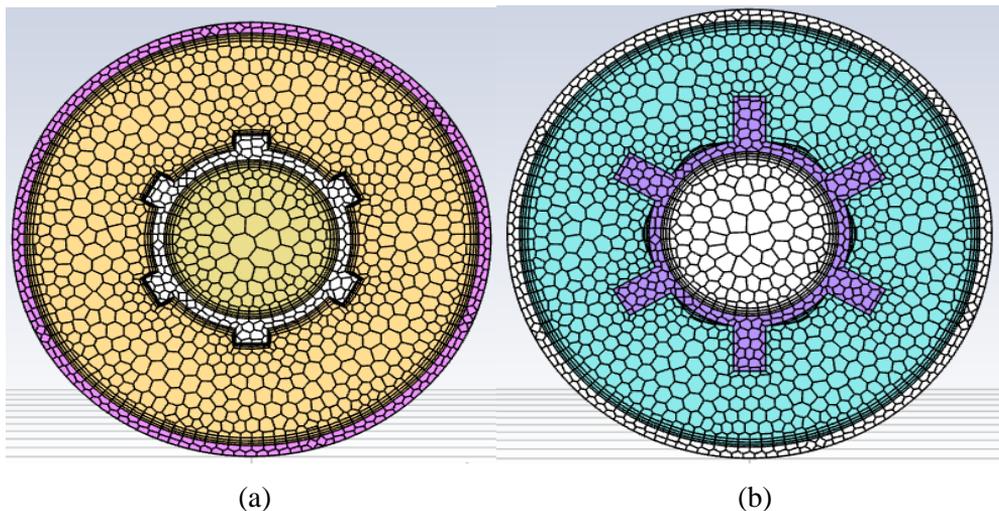
## 2. Numerical model

### 2.1 Physical Model

Figure 1 represents a counter flow double tube heat exchanger with inlet and outlet of cold water and hot water is shown. The simulation was conducted using the Ansys-20 environment to model a tube-in-tube heat exchanger with fins. Hot water flows through the inner tube, while cold water flows through the annulus in a counter-flow configuration. The lengths of both the inner tube and annulus are fixed at 1500 mm. The material selected for both tubes is stainless steel SS304, with a thermal conductivity of 18 W/m K[22]. Figure 2 shows various designs of double-pipe heat exchangers, with different fin heights 2mm, 6mm, 10mm and 14mm. additionally, a longitudinal fin is included, with fin thickness of 4 mm.



Figure 1. Double Tube Heat Exchanger, 1.Hot water inlet 2. Hot water outlet 3. Cold water inlet 4. Cold water outlet



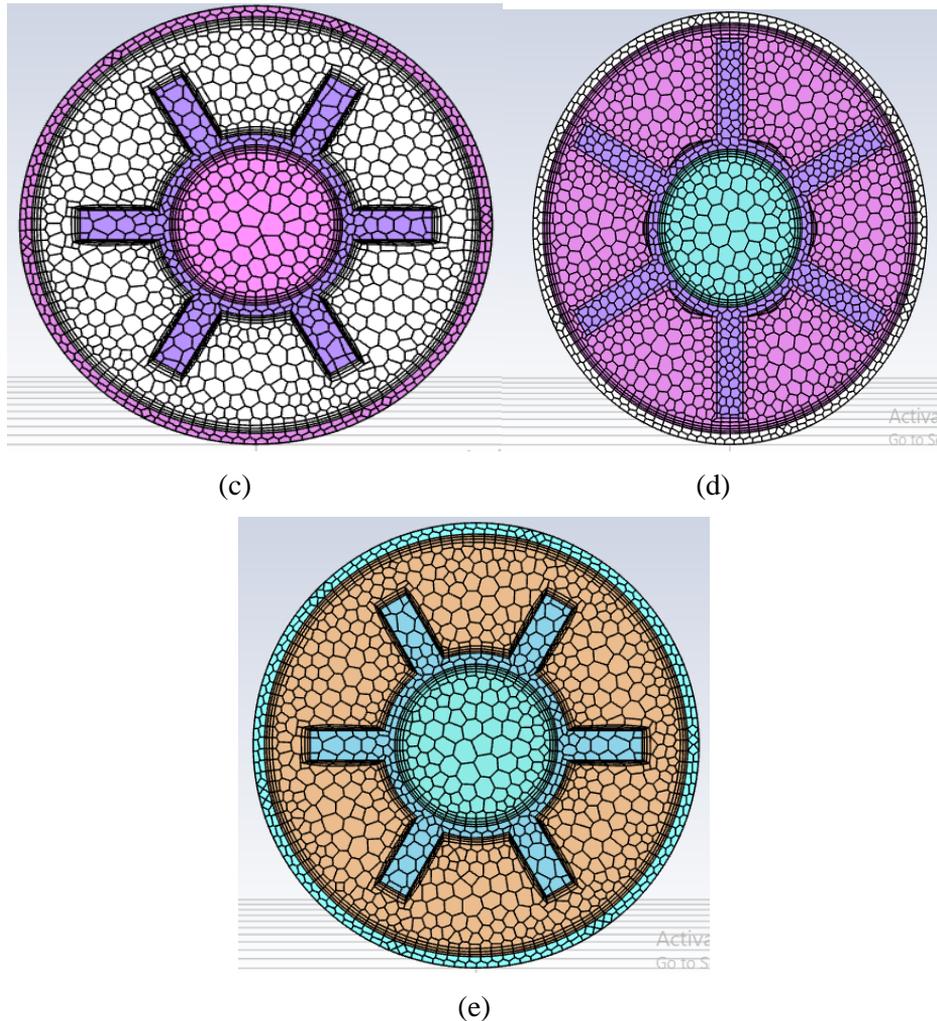


Figure 2. Meshing details of four different fin heights: (a) 2 mm (b) 6 mm (c) 10 mm and (d) 14 mm (e) longitudinal fin

## 2.2 Boundary conditions

The following assumptions are considered in the study of the interrupted fins.

1. Condition of steady state, incompressible, fully turbulent and fully developed flow.
2. Forced convection is used for the heat transport equation.
3. The material of the inner tube is homogeneous and isotropic.
4. The outside tube should be entirely insulated.
5. Radiation plays a very modest role in the heat exchanger, transferring relatively little heat, that is, radiation is negligible.

The equations that govern resolving the turbulence on the heat exchanger's annulus side are listed below [21].

Continuity Equation

$$\frac{\partial u_i}{\partial x_i} = 0 \quad (1)$$

Momentum Equation

$$\frac{\partial u_i u_j}{\partial x} = -\frac{1}{\rho} \frac{\partial \rho}{\partial x_i} + \frac{\partial}{\partial x_j} \left\{ ((v + v_t) \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right)) \right\} \quad (2)$$

Energy Equation

$$\frac{\partial u_i T}{\partial x_i} = \rho \frac{\partial}{\partial x_i} \left\{ \left( \frac{v}{Pr} + \frac{v_t}{Pr_t} \right) \frac{\partial T}{\partial x_i} \right\} \quad (3)$$

Figure3. Represents a grid independence test graph drawn between the number of elements and Nusselt number. The study involved in the modeling four double-pipe heat exchangers with different height configurations and a longitudinal fin. Mesh sensitivity analysis was done on the validation setup for a 2mm height fin at 20lpm to ensure the numerical results were accurate. Four distinct grids, labeled G1, G2, G3, and G4 with 467,311, 806,727, 1,153,826, and 1,321,855 elements respectively, were computed, as shown in Figure 3. The grid independence study revealed that the Nusselt number of G4 were insignificant compared to G3, as the increase in number of elements 168029 would show an improvement in Nusselt number. Hence G3 was selected considering computational elements and convergence time.

Fluid flow and heat transfer in this computational technique are computed using the FLUENT program. The governing problems are solved using the finite volume approach and the SIMPLE algorithm. The first-order upwind method is applied for turbulent kinetic energy and turbulent dissipation rate, while the second-order upwind method is used for energy and momentum. A second-order approach is employed for the pressure term. The convergence criterion for energy is set to  $10^{-6}$ , and for continuity, it is set to  $10^{-3}$ . Table.1 shows the number of elements and its relevant Nusselt number at a fin height of 2 mm and 20lpm mass flow rate.

Table 1. Quality of mesh

S. No	No.of elements	Nusselt number
1	467311	78.51
2	806727	79.4
3	1153826	79.75
4	1321855	79.63

Hexahedral elements are used to mesh computing domains. The momentum boundary condition of no slip and no penetration is applied to every solid wall. Zero heat flux is set for the outer pipe to prevent heat transport from the annular side. The inlets of the tube's interior and annulus sides are set as mass-inlet boundary conditions, while the outlets are set as pressure outlets. The entrance pressure equals the pressure drop on the inner tube and

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annulus sides, assuming that the exits have zero pressure [14].

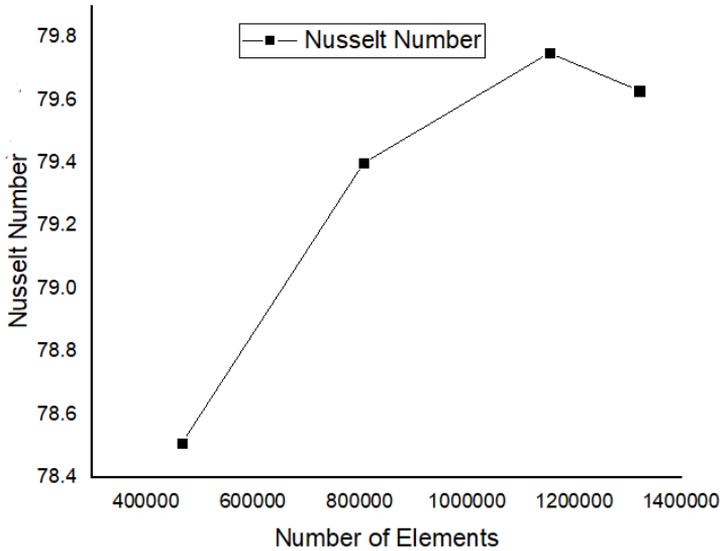


Figure 3. Grid independence study

### 2.3 Data Reduction

Simulated parameters of the heat exchanger with varying heights on the annulus side are used to determine the output temperature of both the cold and hot fluids. The following formulas are employed to calculate the Nusselt number (Nu) using these simulated values.

The rate of heat transfer from the hot fluid

$$Q_h = m_h C_{ph} (T_{h1} - T_{h2}) \tag{4}$$

Where  $Q_h$  is the heat transfer rate of is hot fluid,  $m_h$  is the mass flow rate of hot fluid and  $C_{ph}$  is the specific heat of hot fluid.  $T_{h1}$  is the inlet temperature of hot fluid.  $T_{h2}$  is the outlet temperature of hot fluid.

The heat transfer rate to the cold fluid

$$Q_c = m_c C_{pc} (T_{c2} - T_{c1}) \tag{5}$$

Where  $Q_c$  is the rate of heat transfer rate to the cold fluid (Fe3O4 –water nanofluid),  $m_c$  is the rate of mass flow of cold fluid and  $C_{pc}$  is the specific heat of the cold fluid.  $T_{c1}$  is the inlet temperature of cold fluid,  $T_{c2}$  is the outlet temperature of cold fluid.

The average heat transfer is given by

$$Q_{avg} = (Q_h + Q_c) / 2 \tag{6}$$

Nusselt number can be calculated using

$$Nu = h_o Dh / k_o$$

For a plain pipe, the hydraulic diameter  $D_h = D_i - d_o$ , Finned pipe,  $Dh = 4A_f/P$ ,  $A_f$  area of finned tube, P is the perimeter.

### 3. Results and Discussion

The numerical simulation results are used to calculate heat transfer rate and Nusselt number for different height configurations. The boundary conditions in FLUENT specify zero pressure at both pipe sides' outlets. The pressure drops in the inner tube and annulus are set equal to the corresponding intake pressures on either side. Throughout the simulations with varying height configurations, the inlet mass flow rate and temperature are held constant.

#### 3.1 Validation of the model

Throughout the model validation process, a double-pipe exchanger with a plain pipe is considered. The model selection includes activating the energy equation and opting for the feasible and improved wall treatment (k-epsilon). A realizable K-turbulence model is utilized in this work because it can provide more accurate results for flows involving rotation, boundary layer effects under unfavorable pressure gradients, and recirculation. All Reynolds numbers considered in this study exceed 3000, accounting for fluid flow and heat transfer processes in the steady-state scenario and turbulent regime.

The current numerical study compares various models with a plain tube heat exchanger, focusing on evaluating heat transfer and pressure drop in four geometrically discontinuous finned double heat exchangers with an additional longitudinal fin. Hot water is introduced into the tube side with a constant mass flow rate of 20 lpm at an inlet temperature of 80°C. In the annulus side, cold water is introduced at different mass flow rates ranging from 8 lpm to 20 lpm at a constant inlet temperature of 30°C. Based on the average temperatures of the inlet and outlet of the annulus side, the Nusselt number, pressure drop, and average heat flow rate are calculated

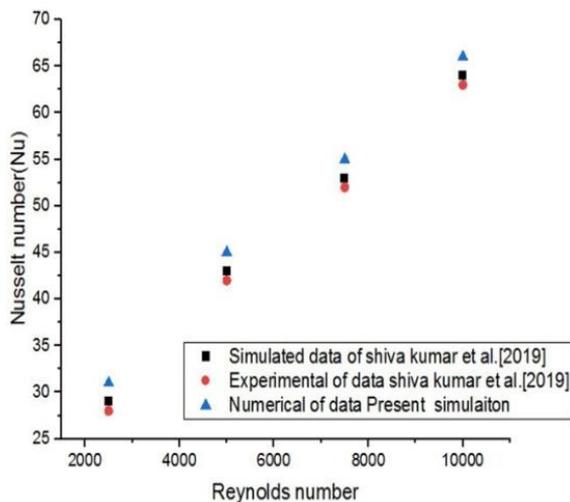


Figure 4. variation of Nusselt number with Reynolds number

Figure 4. Shows the validation of the Nusselt number for the conventional model plain pipe across various Reynolds numbers. To validate the numerical findings of this study, a comparison was made between the experimental and numerical results of those obtained from reference [23], The Nusselt number trends consistently across all outcomes. The maximum error deviation between the current numerical study and numerical, experimental results is 6.89% and 10.71%. The discrepancies in results stem from several factors such as heat losses, friction losses, measurement accuracy of instruments, and numerical assumptions [1]. Consequently, it can be concluded that the established numerical approach is sufficiently accurate and can be utilized to analyze the parameters considered in this study, as detailed in the following section.

### 3.2 Numerical Analysis

Figure 5 shows the velocity contours of the different heights that were studied under the same mass flow rate. From the velocity contours it is visible that as height is increasing velocity of flow increases due to a decrease in hydraulic diameter. With an increase of velocity, heat transfer rate as well pressure drop increase. Figure 6 represents different fin height models

Figure 7 shows static temperature pictures of a heat exchanger with different fin height with a constant interrupted length of 7mm. behind the two fins, there is a recirculation movement of water, creating a vortex motion due to the interrupted length. This enhances local turbulence intensity, leading to the disruption of the thermal boundary layer on the fins and increasing heat transfer.

As the fin height increases from 2mm to 6mm, the temperature rises due to the increased fin area, and the depth of the interrupted length also increases. This results in higher secondary flow and turbulence, causing a higher temperature than the 2mm height configuration. Further increasing the height from 6mm to 10mm intensifies the secondary flow and vortex motion, reducing the thermal boundary layer and increasing turbulence intensity. This leads to a higher temperature compared to the 6mm height configuration.

Increasing the height from 10mm to 14mm results in a decrease in the water temperature, despite the increased fin area and interrupted length of depth. This temperature decrease may be attributed to the reduced flow rate, which prevents the creation of vortex motion in the interrupted length, thus decreasing turbulence. Consequently, despite the increased fin area, the water temperature decreases. With a higher Reynolds number, the velocity and mass flow rate of cold water increase, as does the height of the fin. In finned heat exchangers, the fins create pressure obstructions that increase pressure drop.

As the fin height increases from 2mm to 6mm, the area and depth of the interrupted length increase, leading to a higher pressure drop. Similarly, increasing the height from 6mm to 10mm further increases the area and depth, causing more obstruction to the flow and increasing the pressure drop. When the height of the fin increases from 10 mm to 14 mm, the depth of the interrupted length also increases. This results in greater resistance to flow and consequently a further increase in pressure drop.

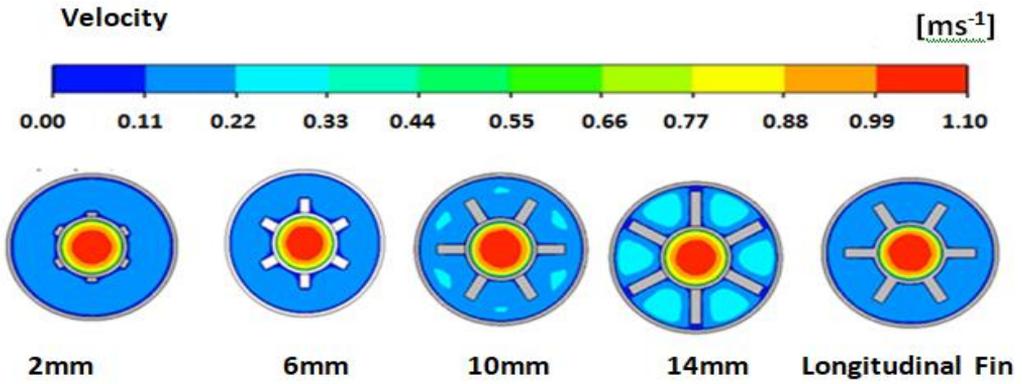
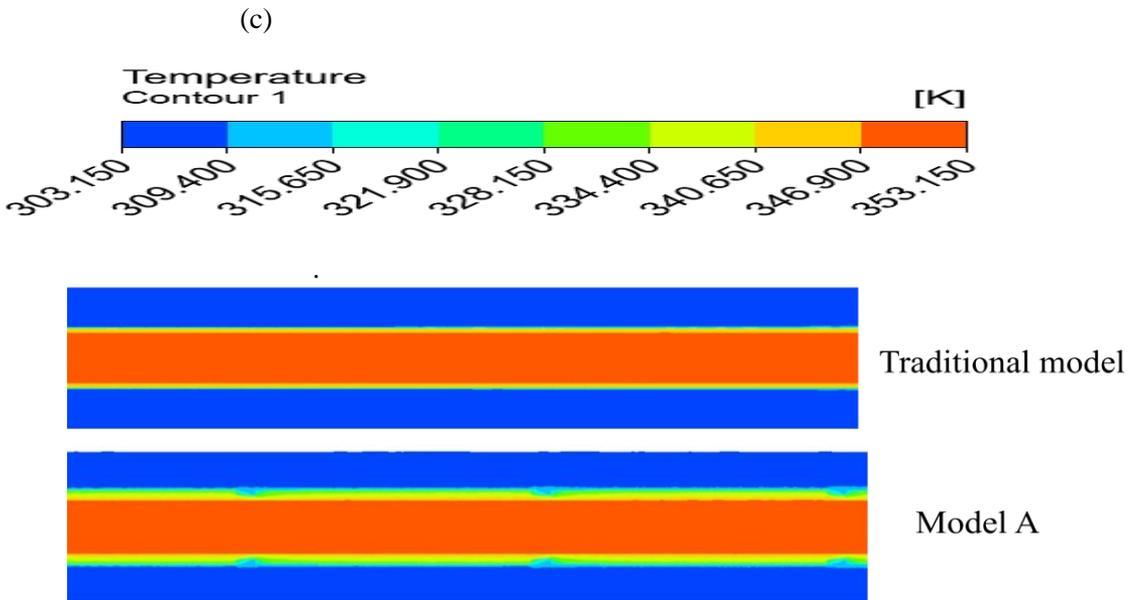


Figure 5. The velocity contours and vectors of fluid flow for different heights



Figure 6 all fin models with 2mm,6mm,10mm and 14mm height



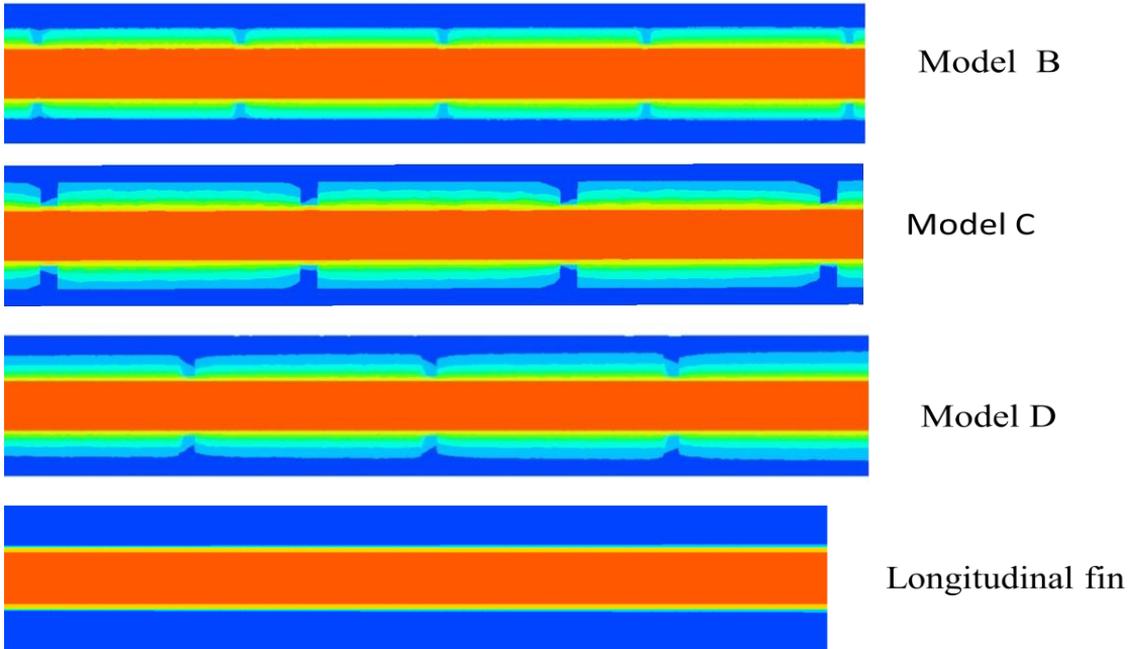


Figure 7. Temperature contours of plain pipe, 2mm, 6mm, 10mm, 14mm fin heights and longitudinal fin

As the fin height increases, the surface area available for heat transfers increases, which leads to a positive correlation between fin height and heat transfer up to a certain point. Higher fins typically enhance heat transfer more effectively than shorter fins due to the increased surface area. However, beyond a certain height, the improvement in heat transfer begins to diminish. This occurs because taller fins become less effective as additional surface area contributes less, primarily due to boundary layer effects and increased thermal resistance. Additionally, taller fins can result in higher pressure drops due to the added resistance they create in the fluid flow.

### 3.3 Heat transfer rate

Figure 8 illustrates the variation in heat transfer rate with mass flow rate for various fin heights in a double-tube heat exchanger along with plain pipe. It is observed that the heat transfer rate increases with the mass flow rate; the analysis includes fin heights of 2mm, 6mm, 10mm, and 14mm, heat transfer rate increases with fin height up to 10mm, later it decreases. As fin height is increased from 2mm to 10mm, the area of fin increases and turbulence is created in interrupted length, which enhances the heat transfer rate. Further, as the height of the fin increases to 14mm and so does the area of the fin, the heat transfer rate decreases due to reduction in flow area and velocity of cold fluid on the side of the annulus, which causes reduction in heat transfer coefficient.

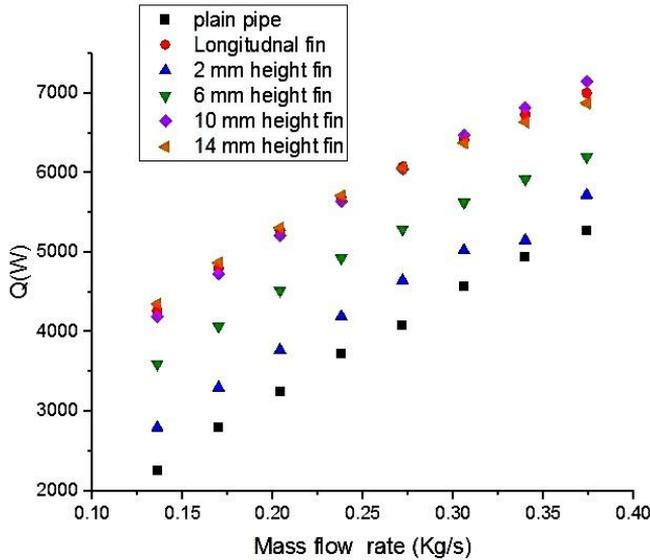


Figure 8. Heat transfer variation with mass flow rate

### 3.4 Nusselt number (Nu)

Figure 9 shows the variation of Nusselt number with mass flow rates; it is observed that as mass flow rate increases Nusselt number also increases. The convective-to-conductive heat transfer at a fluid boundary is quantified by the Nusselt number (Nu). The addition of fins is found to increase Nusselt number in the annulus area. Nusselt number was estimated for various fin heights of 2mm, 6mm, 10mm, and 14mm, longitudinal, plain pipe.

It is observed that Nusselt number increases with fin height; an interruption at regular intervals on the fins causes a disruption in the flow, breaking the thermal boundary at repeated intervals. In turbulent flow, energy transfer via molecular diffusion is faster than in laminar flow. The heat transfer coefficient increases with height up to 10mm and sufficient flow area appears along the finned tube. Due to this, the velocity flow of increases, which enhances heat transfer coefficient; turbulence also occurs in interrupted depth. As the fin height is increased to 14mm, the flow area is reduced and the velocity of flow decreases, resulting in poor velocity of flow and heat transfer coefficient. Therefore maximum Nusselt number was obtained at 10mm fin height, further increase in fin height, the value of Nusselt number gradually decreases.

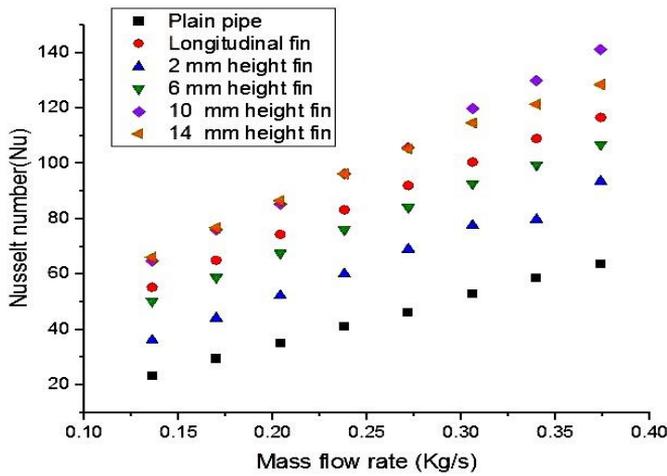


Figure 9. Nusselt number variation with mass flow rate

### 3.5 Pressure drop

Figure 10. indicates that the pressure drop increases proportionally with the fin height. Observations indicate that as mass flow rate increases, there is a corresponding increase in pressure drop. The velocity of cold water also increases in the annulus side; it offers greater resistance to the flow. The pressure drop is smaller when fins are absent compared to when fins are employed. As the fin height increases from 10mm to 14mm, the area of the fin increases and the depth of the interrupted length also increase, and the pressure drops further.

With an increase of fin height from 2mm to 14mm, the height area of fin increases and the area of flow decreases causing reduction in the hydraulic diameter. It enhances the turbulence motion in the interrupted length leading to an increase in heat transfer; as the height is increased from 10 to 14mm, the depth of interrupted length increases, which leads to greater resistance to flow.

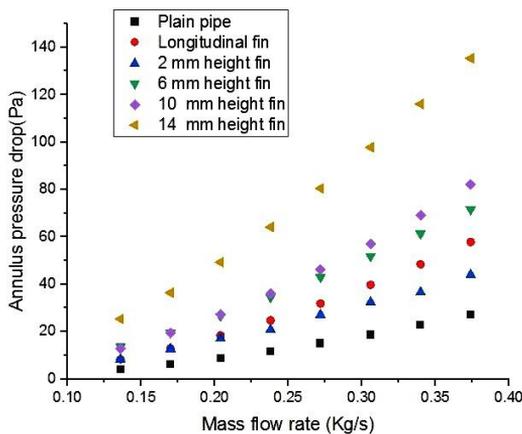


Figure 10. Changes in pressure drop in relation to mass flow rate

### 3.6 Thermal enhancement factor

The thermal enhancement factor, a dimensionless parameter, is established to evaluate the viability of employing a double-pipe heat exchanger. Introducing fins into this setup enhances both the heat transfer rate and the Nusselt number (Nu). However, this improvement also triggers an unwanted escalation in the pressure drop. Consequently, this heightened pressure drop correlates with an increased friction factor.

The thermal enhancement factor is instrumental in evaluating whether the double-pipe heat exchanger is feasible and effective. It can be represented mathematically as [23] :

$$\text{Thermal Enhancement Factor (TEF)} = (\text{Nu}_f / \text{Nu}_p) / (\Delta P_f / \Delta P_p)^{1/3} \text{ -----(7)}$$

Here,  $\text{Nu}_f$  and  $\text{Nu}_p$  represent the Nusselt numbers for the heat exchanger of the finned tube and the plain pipe, respectively, while  $\Delta P_f$  and  $\Delta P_p$  denote the pressure drop for the finned tube and plain pipe.

In Figure 11, the graph displays the relationship between thermal enhancement factor and mass flow rate. Heat transfer rate increases with fin height due to the increased area and turbulence effects. An increase in heat transfer rate leads to an increase in Nusselt number. However, this increase in Nusselt number also results in a higher pressure drop. Initially, it was evident that for all fin configurations, thermal enhancement factor remains above 1, validating the utilization of fins in double-pipe heat exchangers within the scope of the work. Observations indicate that in the case of low mass flow rates, the thermal enhancement factor tends to be higher, gradually decreasing as the mass flow rate increases for finned heat exchangers. The average thermal enhancement factor is 1.28, 1.63, 2.14, and 2.12 for 2mm, 6mm, 10mm, and 14mm heights respectively. The highest thermal enhancement factor is obtained at 10mm height, despite further increase in surface area of fin height. This is because of the thermal enhancement factor decreases, as the flow area gets reduced due to increase in fin height. And also the depth of interruption increases due to this high pumping power required. It is observed from the Figure.10 that thermal enhancement factor was improved for the interrupted fin than longitudinal fin.

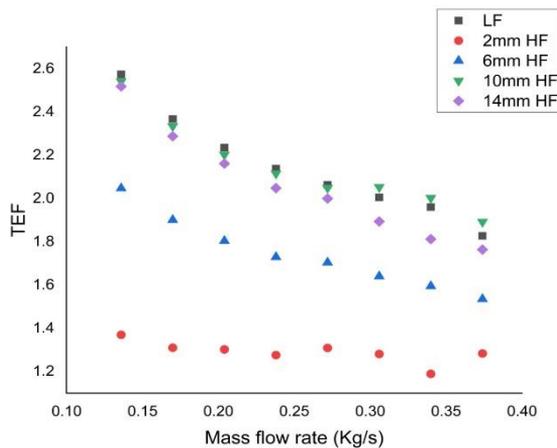


Figure 11. Thermal enhancement factor versus mass flow rate

#### 4. Statistical approach: Taguchi method theory

Various approaches are employed to reduce the cost of experiments, with the Taguchi method being a frequently used technique in the production industry. It serves as a crucial tool for enhancing quality while minimizing costs by identifying control factors that significantly impact the final results [24]. This study focuses on determining the optimal height for maximal heat transmission. Signal-to-noise (S/N) ratio analysis using the Taguchi technique involves designing tests to explore the influence of different control parameters on the S/N ratio.

In this study, four different heights and four different mass flow rates are considered, with the inlet temperature of cold water at 30°C and hot water at 80°C. The Taguchi L16 orthogonal array is utilized for the control factors, and the levels of these variables are shown in Table 2. Orthogonal arrays (OA) are employed in data analysis to estimate optimal results with minimal trials and log functions of the desired output or the ratio of signal to noise (S/N). Table 3 illustrates the combinations of factors and their corresponding responses.

##### 4.1 Signal to noise ratio

The S/N ratio is determined based on the quality of the attributes. The objective of this method is to enhance the heat transfer rate and Nusselt number in an exchanger by efficiently transferring heat from the hot fluid to the cold fluid, which ultimately identifies the optimal height.

Table 2. Control factors and their corresponding levels

Factors	Level 1	Level 2	Level 3	Level 4
Mass flow rate (Kg/s)	0.17	0.238	0.306	0.374
Height(mm)	2	6	10	14

Table 3 Factor level combinations and responses

MFR	Height	Total Heat Transfer	Nusselt Number
0.17	2	3298.56	44.08
0.17	6	4070.64	58.805
0.17	10	4839.33	75.998
0.17	14	4867.59	76.773
0.238	2	4194.33	60.059
0.238	6	4928.78	76.098
0.238	10	5716.24	96.31
0.238	14	5710.75	96.243
0.306	2	5025.31	77.655
0.306	6	5625.82	92.744
0.306	10	6534.55	119.859
0.306	14	6374.3	114.646
0.374	2	5722.8	93.582
0.374	6	6198.14	106.862
0.374	10	7224.16	141.155
0.374	14	6876.18	128.521

The study employs three different types of performance features: “larger-is-better,” “smaller-is-better,” and “nominal-is-better.” For the current investigation, the “larger-is-better” approach is selected for the output variables. After running the simulations, the outcomes are analyzed using the signal-to-noise ratio, where “*y<sub>i</sub>*” represents the measured quantity’s grade, and “*m*” is the total number of trials.

$$S/N = -10 \log_{10} \left( \frac{1}{m} \sum_{i=0}^m \frac{1}{y^2} \right)$$

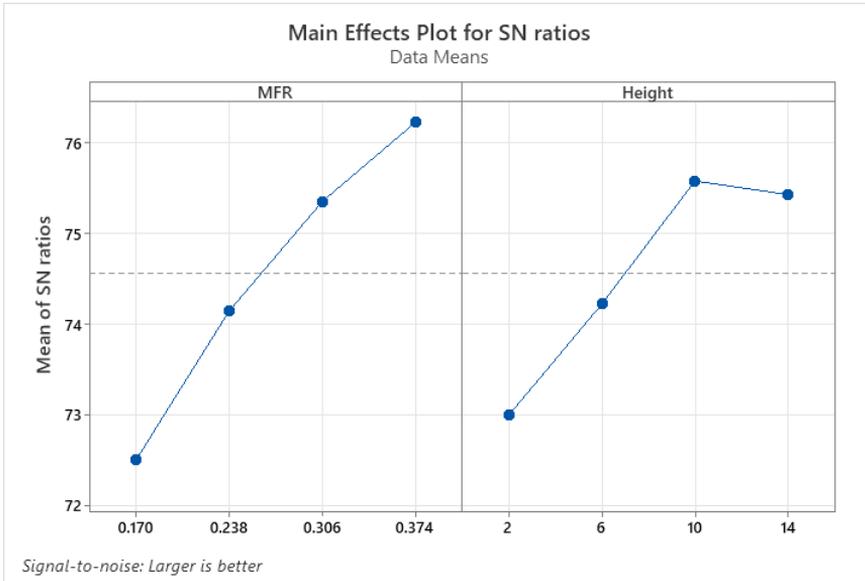


Figure 12. Effect of control factors on total heat transfer

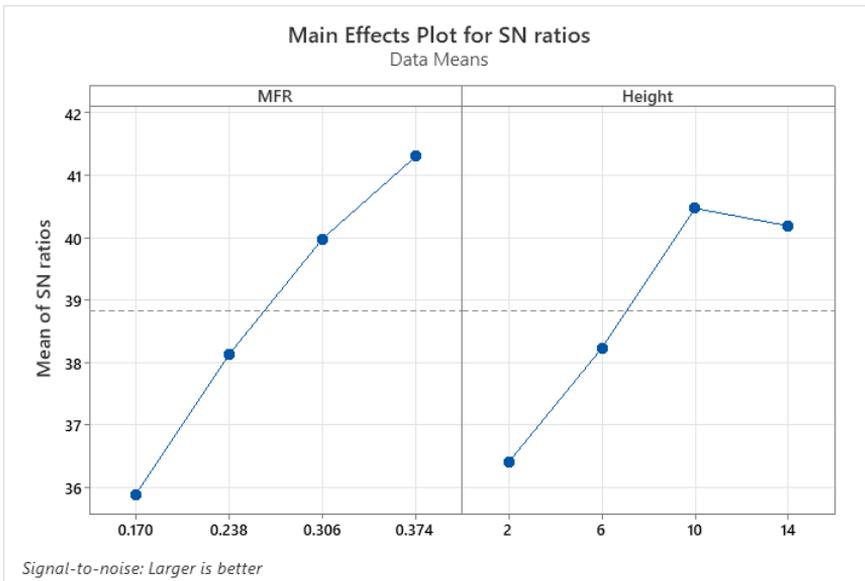


Figure 13. Effect of control factors on Nusselts number

Figures 12–13 depict the main effect plots of S/N ratios, illustrating the main effects for means. By observing the residual plots, the best combination of height, mass flow rate, and Nusselt number (Nu) has been identified. The optimum height for better heat transfer rate and Nusselt number is found to be 10mm, and the heat transfer rate and Nusselt number increase with the mass flow rate.

4.2 ANOVA of the responses

Tables 4 and 5 present the ANOVA results for total heat transfer and Nusselt number, respectively. It is evident from these tables that both factors, MFR and height, significantly affect the responses. MFR contributes more significantly with a contribution percentage of 65%, while height contributes 35% to total heat transfer. For the Nusselt number, MFR has a higher contribution percentage of 62% compared to height, which contributes 38%. The developed model is considered accurate as both R-squared values are over 95%.

Table 4. ANOVA of the total heat transfer

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Mass Flow Rate	3	11198483	3732828	446.03	0.000
Height	3	6014510	2004837	239.56	0.000
Error	9	75320	8369		
Total	15	17288313			
Model Summary					
S	R-sq	R-sq(adj)	R-sq(pred)		
91.4819	99.56%	99.27%	98.62%		

Regression Equation

$$\text{Total Heat Transfer} = 1453 + 10973 \text{ MFR} + 126.6 \text{ Height} \text{-----}(8)$$

Table 5. ANOVA of the Nusselt number

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Mass Flow Rate	3	6478.8	2159.60	171.56	0.000
Height	3	4062.4	1354.12	107.57	0.000
Error	9	113.3	12.59		
Total	15	10654.5			
Model Summary					
S	R-sq	R-sq(adj)	R-sq(pred)		
3.54793	98.94%	98.23%	96.64%		

Regression Equation

$$\text{Nusselt Number} = -6.81 + 264.6 \text{ MFR} + 3.258 \text{ Height} \text{-----}(9)$$

5. Conclusions

In this study, numerical simulations were employed to evaluate the thermal enhancement factor of double-pipe heat exchangers with different heights and constant interrupted length of 7mm. Configurations with different height fins and conventional plain pipe are compared in terms of heat transfer capacity, pressure loss, and comprehensive performance. The effects of the height and mass flow rate on the thermo-fluid performance were also examined. The following conclusions were drawn from the obtained result.

- It is observed that with an increase in the height of fin, heat transfer increases up to a height of 10mm and later decreases due to obstruction in the flow.
- Highest heat transfer of 97.56% is obtained for height of 10mm height compared to a plain pipe with the same mass flow rate.

- Pressure drop increased 2.84 times more than the conventional plain pipe due to obstruction in the flow due to fins and interrupted length.
- The maximum value of thermal enhancement factor was arrived as at 2.54 for the fin height of 10mm under constant interrupted length of 7mm.
- Regression equations have been developed for heat transfer as  $1453 + 10973 \text{ MFR} + 126.6 \text{ height}$ .
- Regression equations have been developed for Nusselt number as  $-6.81 + 264.6 \text{ MFR} + 3.258 \text{ height}$ .

Nomenclature:

Qh	Rate of heat transfer from hot water(W)
QC	Rate of heat transfer from cold water(W)
mh	Mass flow rate of hot water(kg/s)
mc	Mass flow rate of cold water(kg/s)
Cph	Specific heat of hot water (kj/kgK)
Cpc	Specific heat of cold water(kj/kgK)
Qavg	Average heat transfer rate(W)
Th1	Hot water inlet temperature (K)
Th2	Hot water outlet temperature (K)
Tc1	Cold water inlet temperature (K)
Tc2	Cold water outlet temperature(K)
Dh	Hydraulic diameter (m)
Re	Reynolds number
MFR	Mass flow rate(kg/s)
TEF	Thermal enhancement factor
S/N	Signal-to-noise ratio

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