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A Comparative CFD Simulation and Hydrothermal Analysis of Double-Pipe Heat Exchangers Enhanced by Fins and Inserts as Passive Methods

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ABSTRACT: The performance of heat exchangers is vital for reducing industrial energy consumption and operational costs, driving significant interest in passive methods to enhance heat transfer. This study conducts a systematic numerical analysis and comparison of two prevalent passive techniques, low-integral circular fins and plain twisted-tape inserts, utilized in a counterflow double-pipe heat exchanger (DPHE). Three models were simulated in ANSYS FLUENT 2022 R1: a smooth DPHE (baseline), a finned DPHE with integrated circular fins on the outside of the inner pipe, and a DPHE with twisted-tape inserts in the inner pipe. Simulations were performed with hot-side Reynolds numbers (Re) varying from 5500 to 30,000 and inlet temperatures between 50°C and 70°C, while the cool side operated at Re from 2000 to 10,000. Critical performance metrics, including temperature changes, heat transfer rate, pressure drop, entropy generation, and the performance evaluation criterion (PEC), were analyzed to assess the balance between thermal improvements and hydraulic penalties. Finned pipes improve the cold-side temperature differential by 17%–19% at low cold-side Re and increase entropy generation by 20%–70%. Their PEC values range from 1.02 to 1.17, suggesting minor enhancements that diminish under fully turbulent conditions. Conversely, twisted tape inserts demonstrate higher performance at elevated Re and higher hot-side temperatures, increasing the cold-side temperature differential by 17%–26%, increasing entropy generation by 20%–82%, and increasing PEC values of 1.12–1.55, indicating a 12%–55% improvement over the baseline. However, this gain is accompanied by a significant pressure drop penalty, reaching up to four times that of the baseline case. Overall, low-integral fins are better suited to low- to moderate-flow regimes. Twisted tapes are more effective under turbulent conditions due to swirl-induced mixing. These findings support the practical selection of passive enhancement techniques in DPHE designs.

KEYWORDS: Double pipe heat exchanger; energy systems; entropy generation; performance evaluation; low integral finned pipe; twisted tape

1 Introduction

Heat exchangers (HEXs) are the key components in energy transfer. They play an important role in various industrial fields, and their thermal transfer efficiency and effectiveness are directly related to energy consumption and financial costs. Improving the overall heat transfer (HT) efficiency of HEXs has been a subject of many studies [1]. A double-pipe heat exchanger (DPHE) is used in various applications such as heating and cooling, the chemical and food industries, and many systems that require heat transfer. The wide range of applications has made it a subject of considerable interest to researchers and developers

aiming to improve its performance. The insertion is considered one of the main passive techniques to improve the thermal performance of DPHE due to its ease of implementation and efficiency [2]. There are two major categories of heat transfer enhancement techniques for heat exchangers: active and passive techniques. The active technique uses external energy sources such as surface vibrations, magnetic fields, and air injection. Passive techniques involve surface modification, displacement inserts, nanofluids or additives, and turbulence generators. Any passive method for improving the thermal characteristics of the HEX increases the pressure drop (ΔP) across the HEX [3].

Finned DPHEs have been the subject of both experimental and numerical investigations and have undergone design improvements that enhanced the system's hydrothermal performance. Saleh et al. [4] conducted an experimental study on the use of helical fins in a DPHE for the preheating of heavy fuel oil. They reported that increasing the Re of the hot air improved the HT rate, the oil outlet temperature, the overall heat transfer coefficient, and the exchanger effectiveness. This system could achieve annual energy savings of about 1.69 MW in a brick factory. Praveenkumara et al. [5] investigated the influence of integrated fins on the thermal performance of DPHE using hot and cold water at varying flow rates. Their experiments showed that finned pipes significantly improved HT compared to smooth pipes, with counterflow operation providing superior performance. The highest efficiency was observed at the maximum flow rate, particularly with fins of 1.25–1.5 mm pitch and 1 mm depth. Hussein and Hameed [6] experimentally studied an air–water DPHE with segmental fins containing semi-circular perforations. They reported that finned HEXs outperformed non-finned ones, with the overall HT coefficient increasing by 29.7%, 62%, and 80.6% for perforation diameters of 30, 25, and 20 mm, respectively. The thermal performance factor remained above unity in all cases, with the best results observed for a 20 mm perforation diameter. Mohsen et al. [7] experimentally investigated HT enhancement in a DPHE using various fin geometries, including interrupted rectangular fins, circular fins, and helical ribs. Their results showed that all fin types improved the HT coefficient, with rectangular fins giving the greatest enhancement of 68% to 168% and circular fins the least of 3% to 47% compared to a smooth pipe. ΔP varied with fin type, being highest for rectangular fins and lowest for circular fins, and the enhancement depended on the Re of both fluids.

Mozafarie et al. [8] conducted a numerical analysis of the thermal and flow behavior of Al_2O_3 nanofluids in a circular-finned DPHE using a 3D CFD model. They examined the influence of fin height, pitch, and Re , on Nu , friction factor, f , and thermal performance for both Newtonian and non-Newtonian fluids. The results showed that fins enhanced the HT up to 36% for Newtonian nanofluids. In contrast, simultaneous use of fins and nanoparticles reduced the performance for non-Newtonian fluids due to increased pressure drop. Therefore, nanoparticles alone are preferable for non-Newtonian fluids without inserts to optimize performance. Mansour et al. [9] conducted a numerical investigation of annular serrated finned pipes to enhance HT performance. The researchers examined the effects of water mass flow rate, number of fin segments (4–14), and twisting angles (0° – 25°) on Nu and f . The results showed that serrated fins significantly increased Nu relative to smooth pipes, with a maximum enhancement of 61.9% recorded for 14 fin segments. Salman et al. [10] conducted an experimental investigation on a stainless-steel DPHE with plain pipes and helical tape inserts within the inner pipe. Air passed through the inner pipe, while heavy fuel oil flowed in the outer pipe. The results indicated that helical tape increased the HT rate by 46%, demonstrating the benefit of using inserts for thermal performance enhancement. Ammar Ali et al. [11] performed a numerical analysis evaluating the impact of two distinct fin-to-fin height ratios, uniform fin height and variable fin height, on heat transfer and pressure drop in a helical micro-finned pipe in comparison to a smooth pipe ($Do = 18.82$ mm, $Di = 15.64$ mm, $L = 2740$ mm). The CFD results indicated that both finned geometries improved heat transfer by approximately 20% compared to smooth pipes, but at the expense of increased pressure drop. The alternating fin design decreased friction losses by approximately 5% while maintaining

nearly equivalent heat transfer, resulting in enhanced thermo-hydraulic performance over an Re range of 12,000–54,000. Kristiawan et al. [12] conducted an experimental study on TiO_2 /water nanofluids flowing through a copper pipe with helical micro fins (height = 0.256 mm, pitch = 0.22 mm, helix angle $\beta = 18.5^\circ$, $\gamma = 15.4^\circ$, 64 fins), showing a significant enhancement in heat transfer compared to smooth pipes. The fins increased the effective heat transfer area by a factor of 2.56, thereby increasing turbulence and secondary flows, thereby enhancing the Nusselt number. Finned pipes demonstrated superior performance compared to smooth pipes, even in the absence of nanofluids, with consistent improvements across turbulent Re.

Conversely, the use of twisted-tape inserts in DPHE has been examined, with numerous researchers reporting hydrothermal enhancement. Abdulhamed et al. [13] conducted experimental investigations on the use of perforated twisted tapes as counter-swirl flow generators in tubular HEXs. Air performed as the working fluid under turbulent flow, with Re ranging from 4800 to 9500 and tapes having hole porosities of 1.30% and 2.70%. The results showed that increasing the number of holes significantly enhanced Nu by 87% to 97%, resulting in a thermal enhancement factor of up to 89%, while reducing the friction factor by 70% to 94% compared with conventional pipes. Sundar [14] conducted an experimental investigation of the thermal and exergy performance of nanodiamond- Fe_3O_4 /water hybrid nanofluids in pipes with twisted tape inserts under turbulent flow conditions (Re = 2000–22,000). At a nanoparticle volume concentration of 0.2% with H/D = 5 inserts, the Nu and exergy efficiency increased by 73% and 161%, respectively, compared with plain water, while thermal entropy generation decreased by around 49%. However, these improvements were accompanied by a friction factor penalty of up to 62%, highlighting the trade-off between heat transfer enhancement and flow resistance.

Kumar et al. [15] conducted an experimental investigation of the thermo-hydraulic performance of twisted-tape inserts, including solid, perforated, and perforated double V-cut designs, over a Re ranging from 2000 to 25,000. The researchers investigated twist ratios (2–6) and V-cut geometries, finding that all twisted tapes showed enhanced thermal performance by improving Nu for the solid twisted tape by 1.33–2.6, while perforated twisted tapes with double V cuts achieved the highest enhancement by 1.64–3.5 compared to the smooth pipe. Yadav et al. [16] conducted a CFD analysis of the thermo-hydraulic performance of circular pipes with twisted tape inserts of varying length ratios. The researchers assessed Nu, f , and the enhanced efficiency and found that twisted tapes with length ratios of 0.29, 0.43, 0.57, and 1.0 showed higher heat transfer enhancements of 1.07, 1.13, 1.17, and 1.25, respectively. Simultaneously, f was increased by a factor of 1.78, 1.90, 2.02, and 2.26 relative to the smooth-pipe value of the identical length ratio. Safitra et al. [17] investigated plain and jagged (triangular, rectangular, and trapezoidal) twisted tape inserts in a DPHE at low Re ranging from 1000 to 5000. The study revealed that the largest HT occurs in a plain twist, followed by triangular, rectangular, and trapezoidal configurations. The usage of plain and jagged twisted tape produces average ΔP values of 573.22 Pa for plain, 465.84 Pa for triangular, 466.42 Pa for rectangular, and 409.196 Pa for trapezoidal. Kumar and Afzal [18] conducted a numerical investigation of multi-channel twisted tapes with plain, cut trilobe, quad lobe, and Penta lobe configurations within a DPHE at turbulent flow conditions (Re = 6000–20,000). The rectangular cut trilobe multi-channel twisted tape achieved the highest Ne, enhancing HT by 105% to 170% compared to plain DPHE and by 62% to 94% relative to plain twisted tapes, with a maximum thermal performance factor of 1.31.

Dhumal and Havaladar [19] perform an experimental study investigating a copper DPHE (L = 800 mm, $d_i = 13.3$ mm, $d_o = 15.8$ mm, outer pipe $D_i = 52.5$ mm, and $D_o = 60.5$ mm) featuring twisted tape inserts (thickness 1 mm, width 13.5 mm, twist ratios 6.77, 4.51, and 3.38) and helical tape inserts (width 18.3 mm, twist ratios 1.0, 0.67, and 0.50). Employing hot water at 70°C and cold water at 30°C in a turbulent counterflow. The results indicated that integrating twisted and helical tapes augmented turbulence, surface area, and mixing, resulting in Nu enhancements of 219%–316% compared to a plain pipe. The friction factors increased by

4.4 to 8.7-fold, while the thermal performance factor (TPF) reached a maximum of 3.06, indicating enhanced overall efficiency. Cabello et al. [20] conducted a numerical analysis of a DPHE using helical screw-tape inserts with different twist ratios of 2.5, 3.0, 3.5, and 4.0 within the inner pipe. The 3D, steady, turbulent hot-water flow was analyzed using a CFD model using RNG $k-\epsilon$ in the inner pipe and cold water in the annulus in a counterflow configuration. The results indicated that reduction in the twist ratio amplified the swirl flow, disrupted the boundary layers, and markedly improved heat transfer. Nu was augmented by 1.5 to 3.5 times compared to plain pipes, with the maximum increase observed at $y = 2.5$. Nonetheless, friction factor penalties increased with tighter twists, resulting in a trade-off between thermal enhancement and pumping power. The helical screw-tape inserts demonstrated a thermal performance factor greater than unity, validating their efficacy in augmenting passive heat transfer.

A wide range of studies have examined the effects of passive techniques, including fins and tape inserts, on the hydrothermal performance of concentric DPHE. However, there is a lag in the literature of compound passive methods in the DPHE. This study aims to conduct a systematic numerical direct comparison study of integrated circular fins (annular region) and plain twisted tape (inner region) as the most common passive HT enhancement techniques in a DPHE of identical dimensions. Unlike previous studies that evaluated these methods independently. The investigation evaluates thermal and hydraulic performance through Nu, overall HT, ΔP , and entropy generation, highlighting the trade-offs between heat transfer augmentation and pumping power to guide optimal passive enhancement design in industrial applications

2 Computational Fluid Dynamics Framework

2.1 Description of Geometry

Fig. 1 illustrates the geometrical diagram of the three counterflow DPHE models that consist of (a) a smooth double-pipe heat exchanger (SDPHE), (b) a low-finned double-pipe heat exchanger (LFDPE), and (c) double-pipe heat exchangers with twisted tape inserts (TDPHE). All models have been numerically simulated and analyzed using ANSYS FLUENT 2022 R1, sharing the same base dimensions. The SDPHE model features an inner pipe made of copper with an inner diameter of 35 mm, a thickness of 1.55 mm, and a length of 1000 mm. The outer pipe is made of polyvinyl chloride, with an inner diameter of 70.26 mm, a thickness of 2.2 mm, and the same length as the inner pipe. The geometrical dimensions mentioned above serve as a benchmark for comparison in the coming enhancement evaluations.

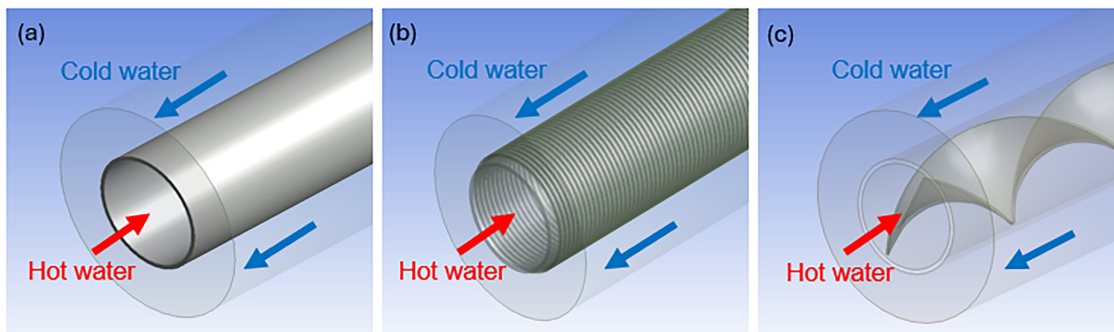


Figure 1: Simulation models and geometrical outlines for (a) SDPHE, (b) LFDPE and (c) TDPHE.

The second model (LFDPE) targets the annular region enhancement of the DPHE, where circular low-integral fins are located on the outer surface of the copper pipe. The fins are 1.5 mm high with a spacing of 1 mm between them. The fin height was selected to ensure relative roughness (e/d) remains ≤ 0.04 [21].

In the TDPHE model, a plain twisted tape (TT) is inserted inside the inner pipe. The width, thickness, and length of TT are 33, 1, and 1000 mm, respectively. Stainless steel was selected as the material for TT with a twist ratio of 2.0. Twisting ratio, thickness, and width with respect to inner diameter were taken from Alsally et al. [22]. All dimensions and material specifications are illustrated in Tables 1 and 2.

Table 1: Geometrical parameters of DPHEs.

Configuration	Parameters (mm)	Length (mm)	Material
Inner Pipe	Inner/outer diameter 35.0/38.1	1000	Copper
Outer Pipe	Inner/outer diameter 70.66/74.66	1000	Polyvinyl chloride
Circular Fins	Height/spacing 1.5/1.0	1.0 (Thickness)	Copper
Twisted tape	Width/Thickness 33/1.0	1000	Stainless steel

Table 2: Material specification.

Material	Density (kg/m ³)	Specific Heat (c_p) (J/kg·K)	Thermal Conductivity (k) (W/m·K)
Copper	8978	381	387.6
Polyvinyl chloride	1388	1050	0.13
Stainless steel	8030	502.48	16.27

Hot water flows in the inner pipe with Re_h ranging from 5000 to 30,000, and the temperature is 50°C to 70°C. In contrast, cold water flows in the annular region with Re_c ranging from 2000 to 10,000, maintaining a constant temperature of 25°C. The selection of piping materials and operating Re ranges followed the methodology outlined by Kumar et al. [23], while the specific pipe dimensions were determined based on their practical availability in the commercial market.

2.2 Mesh Independence Analysis

Mesh independence analysis validates that numerical predictions remain stable regardless of grid refinement. It involves testing multiple mesh sizes and observing fluctuations in output parameters, such as the Nu and outlet temperatures. The analysis ensures computational efficiency while maintaining solution accuracy and physical reliability. Fig. 2 illustrates how mesh sizing affects the output parameters.

An unstructured tetrahedral mesh network was chosen for all models, as illustrated in Fig. 3. Based on mesh tests, the element counts for SDPHE, LFDPE, and TDPHE are 3,246,739, 3,677,164, and 3,443,435, respectively. Further refinement of the grid will not affect simulation outputs. Ten inflation layers were chosen with a controlled first-layer height and a proper growth rate to keep the Y^+ close to 1 for accurate heat transfer prediction.

The numerical uncertainty was estimated from the mesh independence analysis by comparing the selected mesh with the next finer mesh. The maximum deviations in Nu, pressure drop, and outlet temperature were 0.89%, 1.40%, and 0.02%, respectively. Since the PEC and entropy generation were derived from mesh-independent quantities such as Nusselt number, pressure drop, and temperature fields, their associated numerical uncertainties are expected to be within the same range (approximately $\pm 1.5\%$).

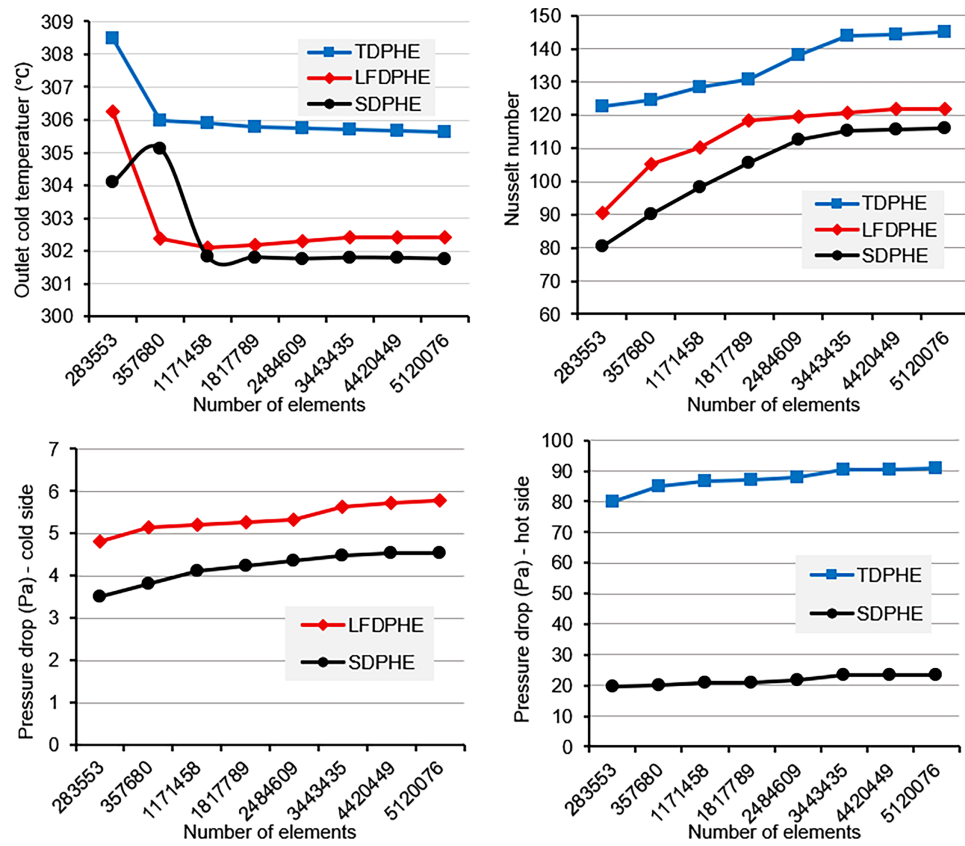


Figure 2: Mesh independence analysis for SDPHE, LFD PHE and TDPHE.

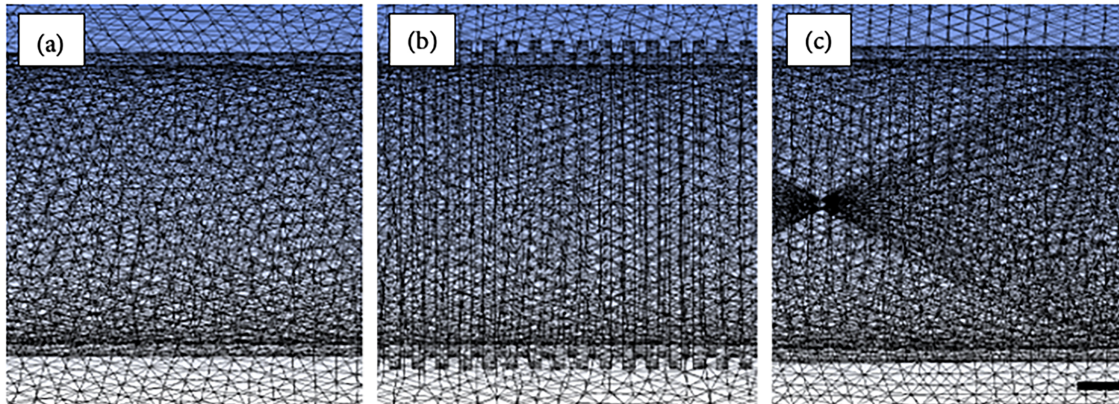


Figure 3: Mesh network for (a) SDPHE, (b) LFD PHE and (c) TDPHE.

2.3 Mesh Quality

Mesh quality is an important part of modeling. To ensure that CFD simulations are accurate and reliable, ANSYS must maintain a high-quality mesh and ensure metrics such as skewness, orthogonality, and aspect ratio. A suitable mesh minimizes numerical errors, accelerates convergence, and accurately describes the flow's physical behavior. The worst elements will have an orthogonality value closer to 0, while the best elements are closer to 1. The minimum value of orthogonality for all types of elements should be higher than 0.01, with an average value considerably higher. For aspect ratio, it is recommended to avoid sudden changes in its

value at locations where large or strong gradients appear; otherwise, for highly skewed elements can reduce accuracy and destabilize the solution. The skewness ranged from zero (best) to one (worst). The maximum skewness for a triangular/tetrahedral mesh in most flows should be kept below 0.95, with a low average value. A maximum value exceeding 0.95 may result in convergence issues and may require adjustment to ANSYS solver parameters. The average mesh quality for the models is shown in Table 3 and agrees with the ANSYS Fluent user guide [24].

Table 3: Average mesh metrics for SDPHE, LFDPE, and TDPHE.

Model	Orthogonal Min./aver.	Skewness Max./aver.	Aspect Ratio Max./aver.	Number of Elements
SDPHE	0.19/0.79	0.79/0.20	9.4/2.23	3,246,739
LFDPE	0.27/0.72	0.72/0.27	14.49/2.29	3,677,164
TDPHE	0.21/0.79	0.78/0.18	20.6/9.56	3,443,435

2.4 Boundary Conditions

The boundary condition used in the numerical simulation is listed in Table 4 below. Those boundary conditions have been applied to all simulation cases for the SDPHE, LFDPE, and TDPHE. The inner and outer pipes are identified by the inlet mass flow rates of hot and cold water, respectively, while the pressure identifies the conditions at the outlets.

Table 4: Boundary conditions for SDPHE, LFDPE, and TDPHE.

Boundary Condition	Parameters
Inner pipe—inlet hot water	Mass flow rate ($\dot{m}_h$) = 0.085–0.426 kg/s Temperatures: 50°C, 60°C and 70°C
Outer pipe—inlet cold water	Mass flow rate ($\dot{m}_c$) = 0.0448–0.224 kg/s Constant temperature 25°C
Outlets (for all models)	Pressure outlet
Walls (for all models)	All walls via system coupling Outer surface of outer pipe heat flux 0 W/m ² (Adiabatic)

2.5 Numerical Assumptions, Setup, and Convergence Criteria

ANSYS FLUENT 2022 R1 was used to simulate the hydrothermal process in a smooth DPHE and a DPHE with built-in circular fins and plain twisted-tape inserts. The assumptions and convergence criteria are set up to reliably measure flow behavior, heat transfer, and entropy generation, enabling a fair comparison of the two passive enhancement strategies. All models are based on several assumptions: an incompressible Newtonian fluid, steady-state single-phase flow, heat radiation is ignored, an adiabatic wall condition for the outer surface of the outer pipe (valid for theoretical study only), and consistent physical parameters with temperature.

To ensure a fair comparison, all models shared similar setup choices. The double-precision Fluent solver was selected as pressure-based, with the energy model activated to calculate the temperature distribution and energy transport. The selected viscous model was the realizable k - ε with enhanced wall treatment, and

the curvature correction factor was set to unity. The realizable k - ε model was selected due to its suitability for a wide range of flows, including rotational/swirl flows, channel flow with pressure variation, and curved flow lines [25]. A coupled method was selected for the pressure and velocity calculations with second-order upwind for pressure, momentum, turbulent kinetic energy, turbulent dissipation rate, and energy. Relaxation factors were set to the program default values as follows: pressure = 0.5, momentum = 0.5, density = 1, body forces = 1, turbulent kinetic energy = 0.75, turbulent dissipation rate = 0.75, turbulent viscosity = 1, and energy = 0.75. The governing equations for this article are explained as follows [26]:

Continuity equation:

$$\frac{\partial \bar{u}}{\partial x} + \frac{\partial \bar{v}}{\partial y} + \frac{\partial \bar{w}}{\partial z} = 0 \quad (1)$$

Conservation of mass

$$\left(\bar{u} \frac{\partial \bar{u}}{\partial x} + \bar{v} \frac{\partial \bar{u}}{\partial y} + \bar{w} \frac{\partial \bar{u}}{\partial z} \right) + \left(\frac{\partial}{\partial x} (\overline{u'^2}) + \frac{\partial}{\partial y} (\overline{u'v'}) + \frac{\partial}{\partial z} (\overline{u'w'}) \right) = -\frac{1}{\rho} \frac{\partial p}{\partial x} + \frac{\mu}{\rho} \left(\frac{\partial^2 \bar{u}}{\partial x^2} + \frac{\partial^2 \bar{u}}{\partial y^2} + \frac{\partial^2 \bar{u}}{\partial z^2} \right) \quad (2)$$

$$\left(\bar{u} \frac{\partial \bar{v}}{\partial x} + \bar{v} \frac{\partial \bar{v}}{\partial y} + \bar{w} \frac{\partial \bar{v}}{\partial z} \right) + \left(\frac{\partial}{\partial x} (\overline{u'v'}) + \frac{\partial}{\partial y} (\overline{v'^2}) + \frac{\partial}{\partial z} (\overline{v'w'}) \right) = -\frac{1}{\rho} \frac{\partial p}{\partial y} + \frac{\mu}{\rho} \left(\frac{\partial^2 \bar{v}}{\partial x^2} + \frac{\partial^2 \bar{v}}{\partial y^2} + \frac{\partial^2 \bar{v}}{\partial z^2} \right) \quad (3)$$

$$\left(\bar{u} \frac{\partial \bar{w}}{\partial x} + \bar{v} \frac{\partial \bar{w}}{\partial y} + \bar{w} \frac{\partial \bar{w}}{\partial z} \right) + \left(\frac{\partial}{\partial x} (\overline{u'w'}) + \frac{\partial}{\partial y} (\overline{v'w'}) + \frac{\partial}{\partial z} (\overline{w'^2}) \right) = -\frac{1}{\rho} \frac{\partial p}{\partial z} + \frac{\mu}{\rho} \left(\frac{\partial^2 \bar{w}}{\partial x^2} + \frac{\partial^2 \bar{w}}{\partial y^2} + \frac{\partial^2 \bar{w}}{\partial z^2} \right) \quad (4)$$

Conservation of energy

$$\left(\bar{u} \frac{\partial \bar{T}}{\partial x} + \bar{v} \frac{\partial \bar{T}}{\partial y} + \bar{w} \frac{\partial \bar{T}}{\partial z} \right) + \left(\frac{\partial}{\partial x} (\overline{u'T'}) + \frac{\partial}{\partial y} (\overline{v'T'}) + \frac{\partial}{\partial z} (\overline{w'T'}) \right) = \alpha \left(\frac{\partial^2 \bar{T}}{\partial x^2} + \frac{\partial^2 \bar{T}}{\partial y^2} + \frac{\partial^2 \bar{T}}{\partial z^2} \right) \quad (5)$$

The residual convergence behavior of the CFD solution for the three cases was generated. A sample of the results is shown in Fig. 4. The most complicated geometry, and hence the most complex flow, is selected. The necessary iterations to achieve convergence ranged from 200 to 220 for the SDPHE, 200–260 for the LFDPE, and 280–300 for the TDPHE. The required duration varies from one to two hours, depending on the case. Solution convergence criteria were set to 1×10^{-4} for continuity, x-velocity, y-velocity, z-velocity, and k - ε . At the same time, the energy convergence criteria were kept as 1×10^{-6} .

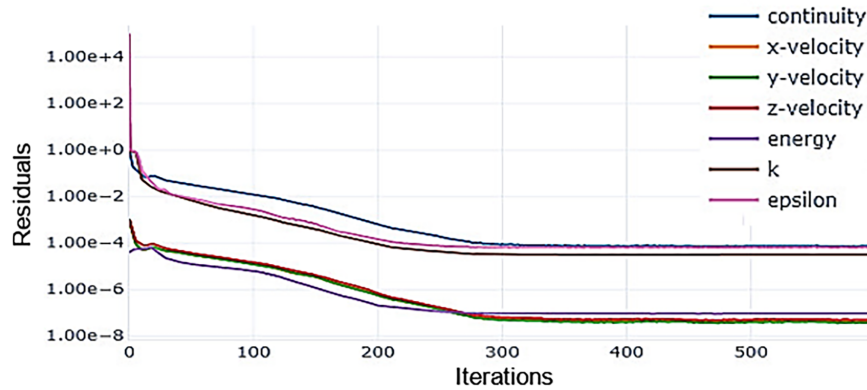


Figure 4: Residual convergence behavior of CFD solution for the TDPHE.

3 Data Reduction Equations

To evaluate the thermal–hydraulic performance of the DPHE, the measured and simulated parameters must be converted to dimensionless, performance-based quantities. The following data reduction equations are used to calculate the Re, Nusselt number, friction factor, and thermal effectiveness, enabling consistent comparison across different passive enhancement techniques. These correlations form the basis for analyzing heat transfer, pressure drop, and entropy generation.

Heat transferred from hot water (Q_h) defined by Eq. (6) and heat transferred to cold water (Q_c) defined by Eq. (7) can be calculated based on the energy balance of a fluid stream [27]:

$$Q_h = \dot{m}_h C_{p,h} (T_{h,i} - T_{h,o}) \quad (6)$$

$$Q_c = \dot{m}_c C_{p,c} (T_{c,o} - T_{c,i}) \quad (7)$$

where $\dot{m}_h C_{p,h}$, $\dot{m}_c C_{p,c}$ are fluid properties (mass flow rate and specific heat) were calculated based on the average temperature (T_{avg}) given by Eq. (8) as follows:

$$T_{avg} = (T_i + T_o)/2 \quad (8)$$

where T_i and T_o represent inlet and outlet temperatures, respectively.

In addition, from the energy balance, it could be stated that the heat transferred from the hot water is equal to the heat transferred to the cold water ($Q_h = Q_c$).

The Nusselt number is a dimensionless quantity defined by Eq. (9) as the ratio of convective heat transfer to purely conductive heat transfer across a fluid layer as follows:

$$Nu = (h \times D_h)/k \quad (9)$$

where h is the convective heat transfer coefficient, D_h is the hydraulic diameter, and k is the conductive heat transfer coefficient.

The Reynolds number, Re, is a dimensionless quantity given by Eq. (10) as the ratio between inertial forces and viscous forces in a fluid flow as follows:

$$Re_D = \rho u D_h / \mu \quad (10)$$

The performance evaluation criterion is defined by Eq. (11) as the ratio of the heat transfer enhancement to the frictional penalty, expressed as:

$$PEC = \left(\frac{Nu}{Nu_{smooth}} \right) / \left(\frac{f}{f_{smooth}} \right)^{1/3} \quad (11)$$

where f is the friction factor defined by Eq. (12) for laminar flow and Eq. (13) for turbulent flow as follows:

$$f = 64/Re \quad (12)$$

$$f = 0.316/Re^{0.25} \quad (13)$$

Total entropy generation ($S_{gen,total}$) defined by Eq. (14) as the measure of irreversibility caused by two main effects: heat transfer across a finite temperature difference ($S_{gen,ht}$) given by Eq. (15) and fluid friction leading to pressure drops and viscous dissipation ($S_{gen,f}$) given by Eq. (16). It quantifies the thermodynamic losses that reduce the exchanger's efficiency [28,29] as follows:

$$S_{gen,total} = S_{gen,ht} + S_{gen,f} \quad (14)$$

$$S_{gen,ht} = m_h C_{p,h} \ln \left(\frac{T_{h,o}}{T_{h,i}} \right) + m_c C_{p,c} \ln \left(\frac{T_{c,o}}{T_{c,i}} \right) \quad (15)$$

$$S_{gen,f} = \frac{m_h \Delta P_{h,i}}{\rho_h T_{avg,h}} + \frac{m_c \Delta P_{c,o}}{\rho_c T_{avg,c}}, \quad (16)$$

where $\Delta P_{h,i}$ and $\Delta P_{c,o}$ are the pressure differences across the hot and cold pipes, respectively.

For calculating the Bejan number (Be) can be used, and it's given by:

$$Be = \frac{S_{gen,ht}}{S_{gen,ht} + S_{gen,f}} \quad (17)$$

4 Results and Discussion

4.1 Validation of the Computational Simulation Procedure

To verify ANSYS FLUENT 2022 R1, results were compared with those of Kumar et al. [23], featuring flow rates between 0.02 and 0.1 kg/s. The temperatures at the hot water inlet were 332 and 303 K for the cold water. Pressure and velocity equations were resolved using SIMPLEC with a skewness factor of 1. The selected viscous model was the realizable $k-\epsilon$ with enhanced wall treatment, and the curvature correction factor was set to unity. Figs. 5 and 6 show that the relative deviation was less than 10%, which proves the software's accuracy.

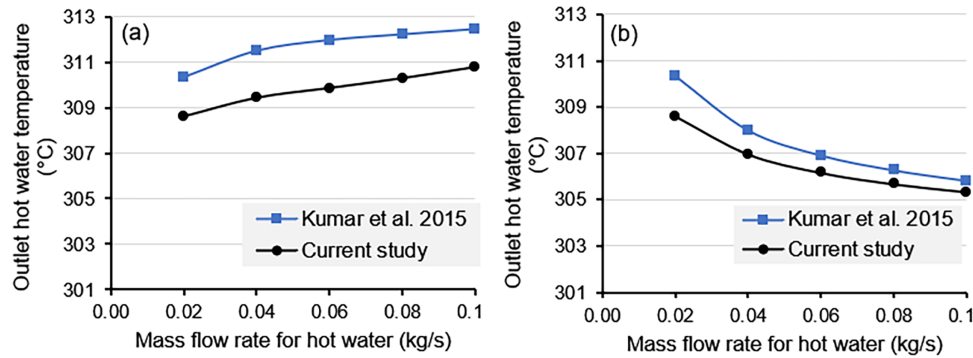


Figure 5: Validation of the current study with Kumar et al. [23] results. (a) at a cold-water flow rate of 0.02 kg/s. (b) at hot-water flow rate 0.02 kg/s.

4.2 Effect of Fins and Twisted Tape on Cold Side Temperature Difference Profiles

Fig. 7 shows the effect of hot-side Re_h on the cold-side temperature difference, ΔT_c , for the three models at two cold-side temperatures ($Re_c = 2000$ and $10,000$) and varying hot-water inlet temperatures of 50°C – 70°C . At low $Re_c = 2000$, as shown in Fig. 7a, ΔT_c increases with increasing Re_h for all models. As Re_h increases, higher velocity and turbulence are introduced into the HEX, which, in turn, leads to an increase in HT. This finding is consistent with the results reported by Nwoko et al. [30], which makes the hot water carry more heat energy to the DPHEs since heat capacity ($\dot{m}_h C_{p,h}$) increased as stated in Eq. (6). The LDPHE exhibits the highest ΔT_c enhancement, outperforming the SDPHE by 17%–19% and the TDPHE by 6%–8%. Low fins outperformed the twisted tape at low Re_c because, in this region, weak convection on the cold side mostly stops heat transfer. Low-finned pipe works better because it increases the surface area and breaks the thermal boundary layer. These results agree well with those of Ammar Ali et al. [11].

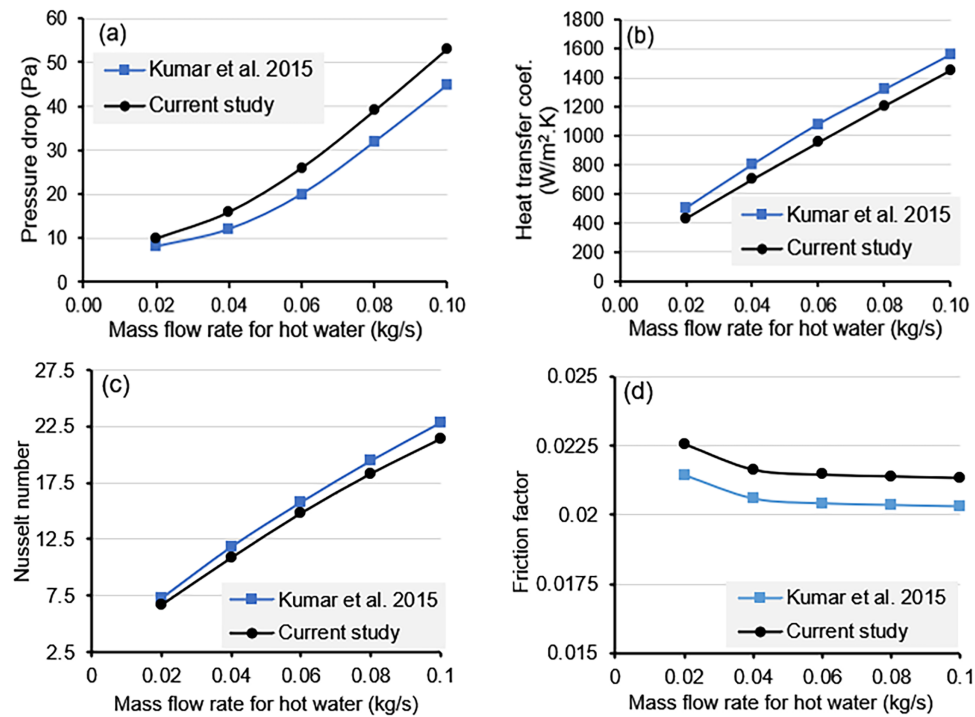


Figure 6: Validation of the current study with Kumar et al. [23] at a hot-water flow rate of 0.02 kg/s (a) pressure drop validation, (b) annulus heat transfer coefficient validation, (c) Nusselt number-hot side, and (d) friction factor-hot side.

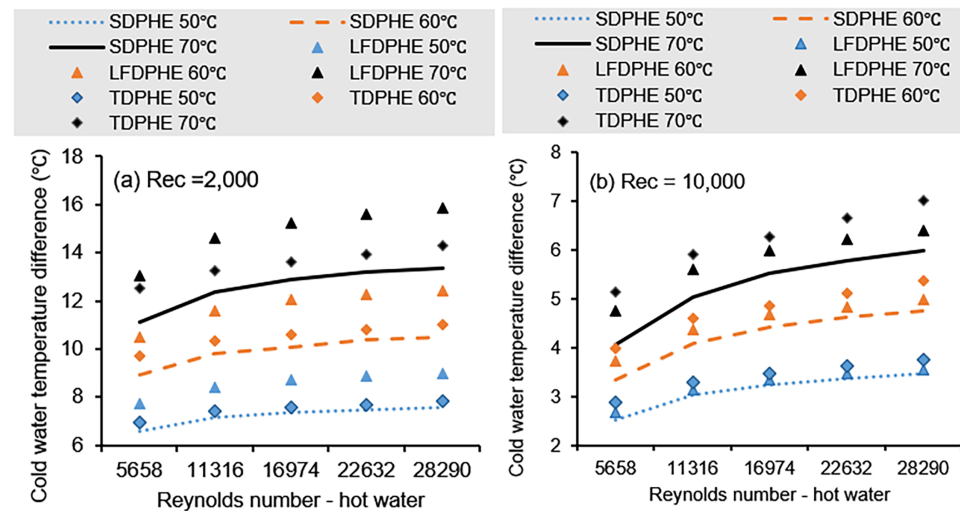


Figure 7: Temperature difference profiles for LFDPE, TDPHE, and SDPHE. (a) at $Re_c = 2000$ and (b) at $Re_c = 10,000$.

At high $Re_c = 10,000$ as shown in Fig. 7b, the performance trend shifts. Although ΔT_c continues to rise with Re_h , the twisted tape demonstrates the strongest enhancement, surpassing the smooth pipe by 17%–26% and outperforming the low fins by nearly 10%. Under these conditions, the cold side is already fully turbulent, and the heat transmission is no longer limited by cold-side convection. Instead, the hot-side resistance becomes dominant, and the twisted tape provides significant improvements by generating strong swirl and secondary flows that promote fluid mixing and reduce the hot-side boundary-layer thickness [31].

In both cases, the smooth pipe consistently shows the lowest ΔT_c , confirming that passive techniques provide measurable thermal benefits.

The results in Fig. 8 highlight a regime-dependent enhancement mechanism: low-integral fins are more effective at low Re , while twisted tape is more effective at high Re , where hot-side resistance controls heat transfer.

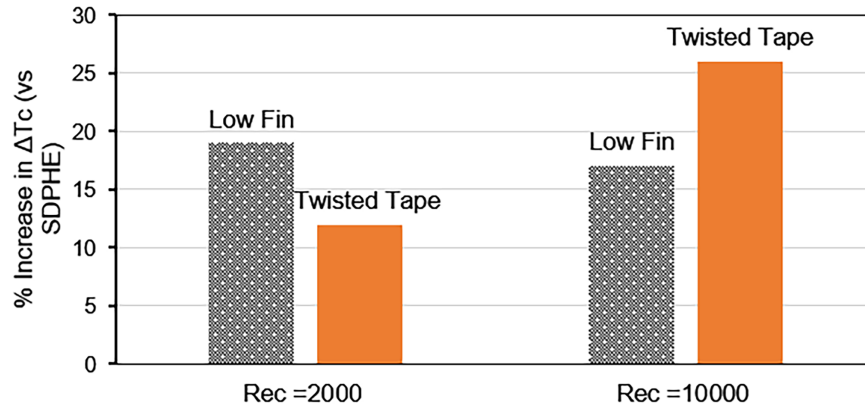


Figure 8: The enhancement percentage in cold-side temperature difference at $Re_c = 2000$ and at $Re_c = 10,000$ using low fins and twisted tape in the DPHE.

4.3 Effect of Reynolds Number, Fins, and Twisted Tape on Heat Transfer Rate

Fig. 9 shows the effect of Re_h on the heat transfer rate for the three models at Re of 2000 and 10,000, with hot-water inlet temperatures of 50°C, 60°C, and 70°C. The heat transfer behavior is similar to the temperature difference of cold water mentioned in Section 4.2. At low $Re_c = 2000$, as shown in Fig. 9a, the LFDPE demonstrates the highest performance, achieving HT enhancements of approximately 9%–11% over the SDPHE, whereas the TDPHE provides smaller improvements of 6%–7.7%. This behavior is consistent with the earlier observation that low fins are more effective under weak cold-side convection, as the added surface area and fin-induced disruption of the boundary layer mitigate cold-side resistance. At high $Re_c = 10,000$, as shown in Fig. 9b, the performance trend reverses; the TDPHE exhibits substantial gains, with HT enhancements ranging from 19% to 31% relative to SDPHE, while LFDPE improvements remain modest at only 3%–9%. This transition occurs when the cold side is already turbulent, with the primary resistance on the hot side, where twisted tape significantly improves mixing and reduces thermal resistance. The findings validate a regime-dependent enhancement mechanism, wherein low fins are beneficial at low Re , whereas twisted tape prevails at high Re , as stated and cited in Section 4.2. Fig. 10 illustrates the above results by comparing the enhancement percentage in the cold-side temperature difference for LFDPE, TDPHE and SDPHE.

4.4 Entropy Generation

Entropy generation reflects exergy degradation and represents thermodynamic irreversibility. It helps designers make the best choices about system characteristics by measuring how much energy is lost due to inefficiencies [32].

Fig. 11 presents the variation of entropy generation due to heat transfer for the three configurations. The TDPHE exhibits the highest entropy generation in all tested cases, exceeding that of the SDPHE by approximately 20%–82%, while the LFDPE shows intermediate values with increases of 15%–70% relative to the baseline. These elevated values arise from enhanced fluid mixing, swirl, and boundary-layer disruption,

which intensify heat transfer but simultaneously increase thermal irreversibility. Entropy generation peaks at an inlet temperature of 70°C and increases with Re_h , as higher flow rates enhance convective transfer and amplify entropy production.

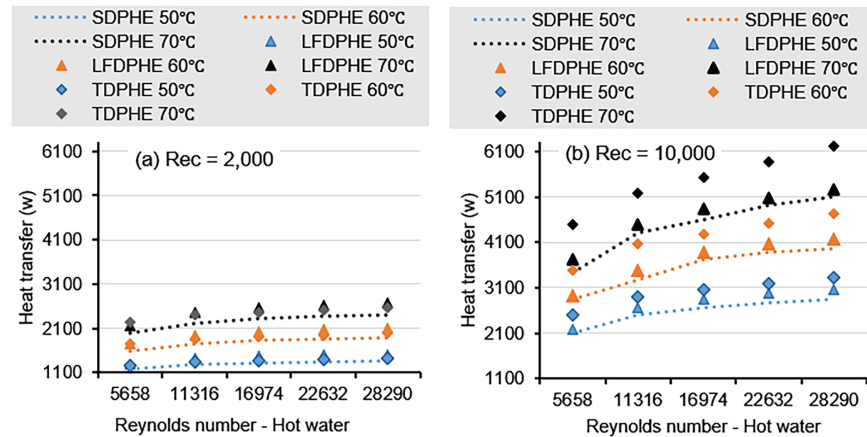


Figure 9: Heat transfer between the fluids, (a) at $Re_c = 2000$ and (b) at $Re_c = 10,000$.

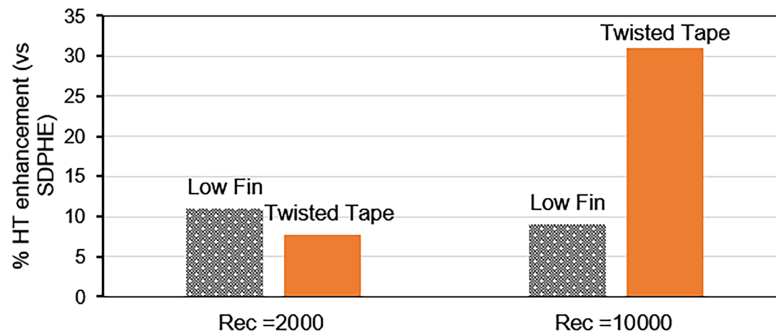


Figure 10: The enhancement percentage in the heat transfer for LFD PHE and TDPHE compared to SDPHE.

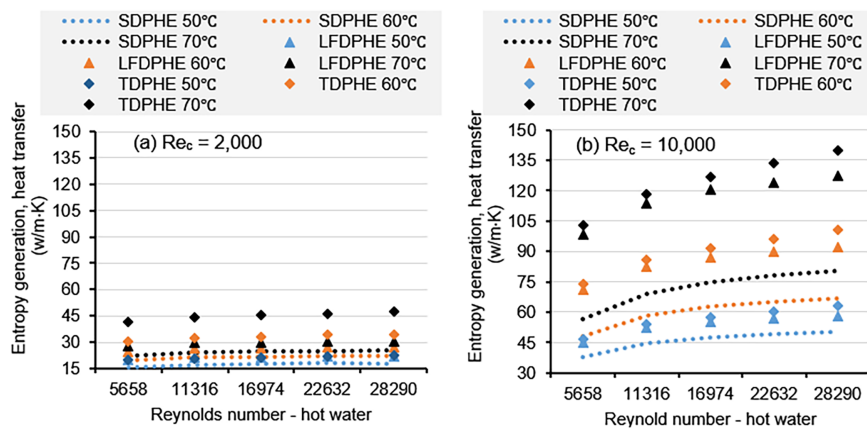


Figure 11: Entropy generation due to heat transfer for SDPHE, LFD PHE, and TDPHE (a) at $Re_c = 2000$ and (b) at $Re_c = 10,000$.

Fig. 12 illustrates the entropy generation due to frictional irreversibility. The results confirm that increasing Re_h significantly raises frictional entropy due to higher turbulence intensity and elevated wall shear stress. The TDPHE again produces the greatest penalty, with frictional entropy generation rising nearly sixfold compared to its initial value, while the LFPHE shows a threefold increase relative to the SDPHE. This behavior is attributed to the swirling motion induced by twisted tape, which raises local wall shear stress and elongates the fluid path length through spiral motion rather than straight flow, thereby increasing flow resistance and dissipation losses.

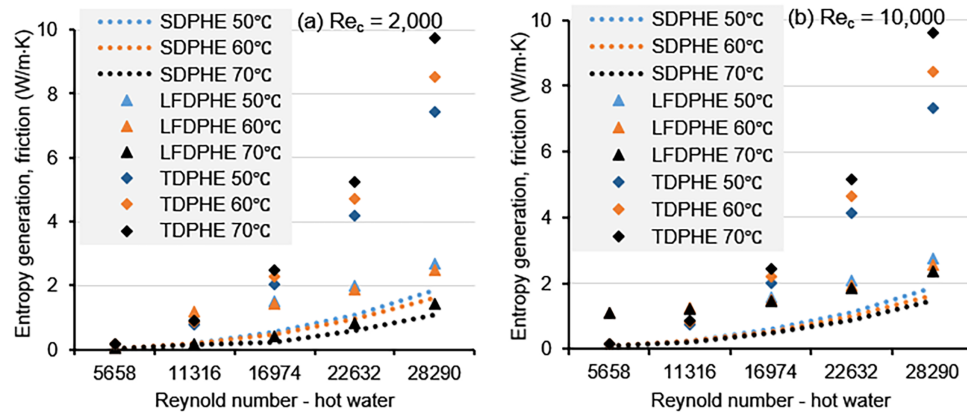


Figure 12: Entropy generation due to friction for SDPHE, LFPHE, and TDPHE (a) at $Re_c = 2000$ and (b) at $Re_c = 10,000$.

In general, the results in Figs. 12 and 13 showed that entropy generation, whether from heat transfer or frictional irreversibility, increases with Re ; a similar behavior was reported by Mertaslan and Keklikcioglu [33], and Cao et al. [34]. As shown in Fig. 14, the highest value of total entropy generation calculated using Eq. (14) was achieved by the twisted tape insert, followed by the low-finned pipe.

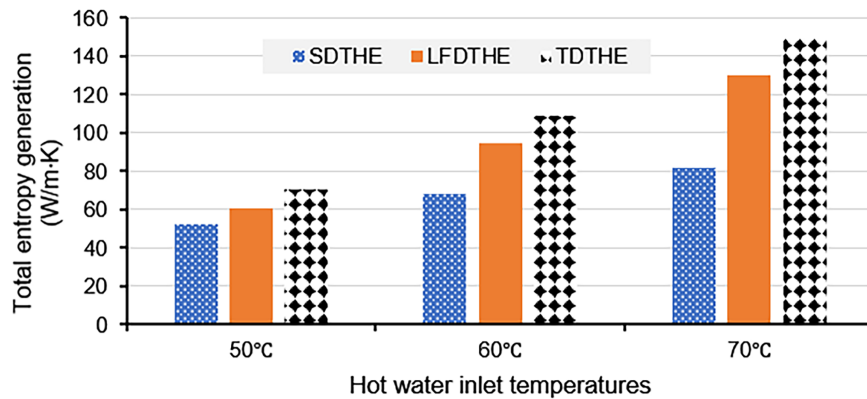


Figure 13: The total entropy generation for SDPHE, LFPHE, and TDPHE at different inlet hot water temperatures and the highest Re in both cold and hot sides of $Re_c = 10,000$ and $Re_h = 30,000$.

Figs. 14 and 15 illustrate the variation of the Bejan number across the Re range studied. The Bejan number remains close to 1.0 for all models, indicating that thermal transfer irreversibility dominates the total entropy generation over fluid friction losses. As the hot water Re increases, the Bejan number decreases slightly across all models, with the TDPHE consistently showing lower values than the SDPHE and LFPHE.

This decreasing trend and the difference between the two models occur because higher fluid velocities and the presence of twisted tape physically enhance the turbulent mixing and swirl intensity, which increases fluid friction and ΔP penalties relative to the overall Q .

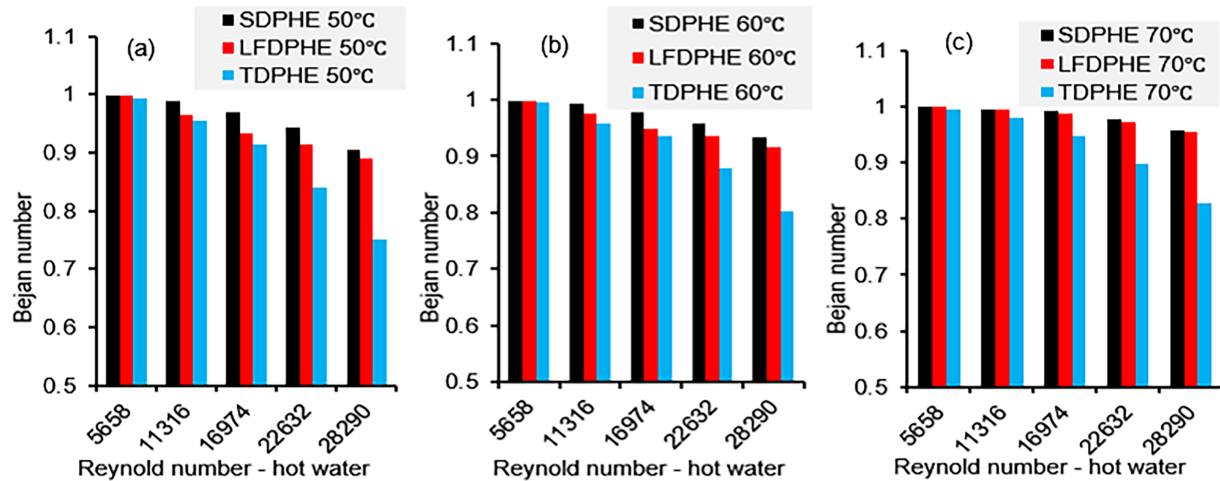


Figure 14: Bejan number for SDPHE, LFDPE, and TDPHE at different inlet hot water temperatures at (a) 50°C, (b) 60°C, and (c) 70°C of $Re_c = 2000$.

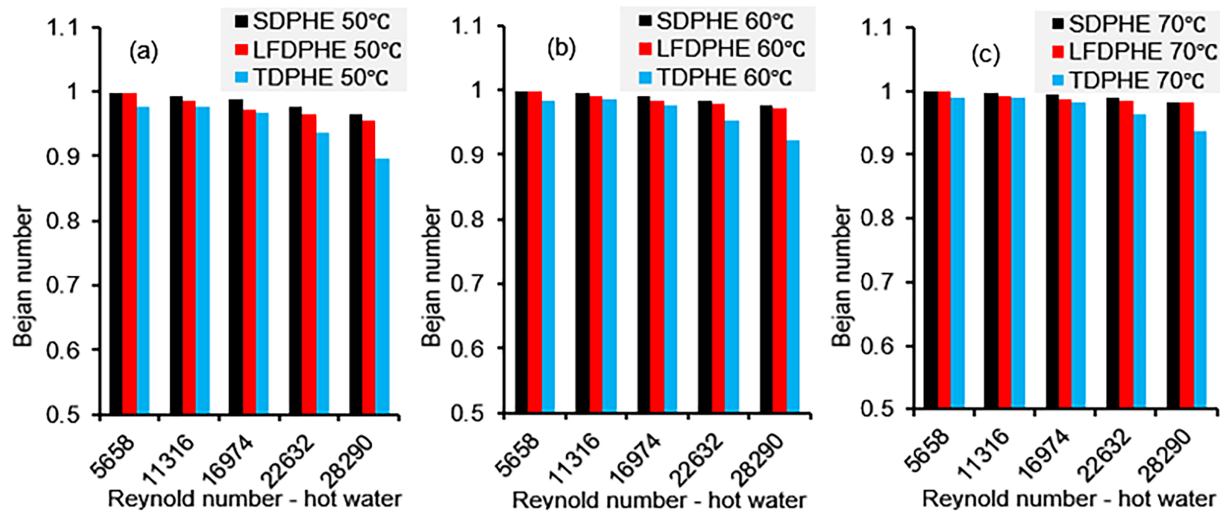


Figure 15: Bejan number for SDPHE, LFDPE, and TDPHE at different inlet hot water temperatures at (a) 50°C, (b) 60°C, and (c) 70°C of $Re_c = 10,000$.

4.5 Pressure Difference across Heat Exchangers

Fig. 16 shows a basic trade-off in heat exchanger design: as heat transfer effectiveness increases, pressure loss also increases. LFDPE and TDPHE transport heat better than SDPHE, but they also generate higher turbulence and friction, resulting in a much higher pressure drop, as also noted by Rashid et al. [35]. This phenomenon means that pumping the fluid through the system requires more energy when the temperature differential between the cold and hot water is greater. The LFDPE generates around 1.24 pressure loss, while the TDPHE generates around 4 times more than the SDPHE. Twisted tape shows a significantly higher pressure loss at any given Re . This increase is the hydraulic penalty for the HT enhancement it provides.

The pressure loss is due to the fluid moving in a swirling direction. A straight pipe doesn't move this way naturally, and it takes work to make and keep it going. The swirling makes the fluid more turbulent and increases friction, both along the pipe walls and on the twisted tape surface. This swirling motion causes a considerably higher drop in pressure. The ΔP is caused by the energy needed to make the swirling motion and resist the increasing drag.

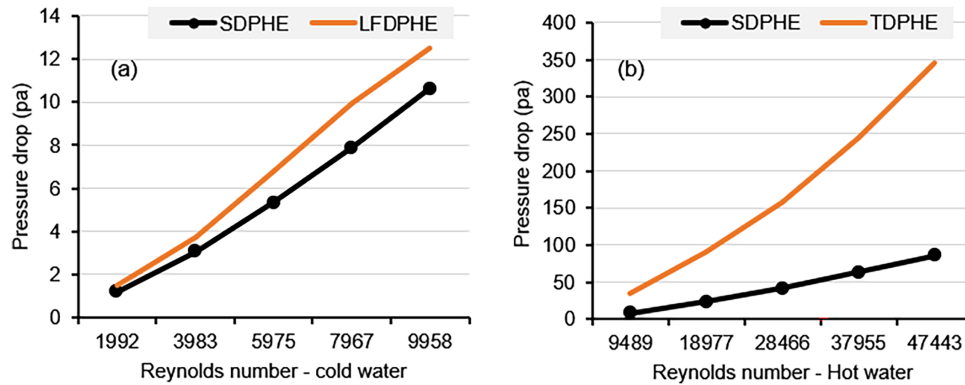


Figure 16: Comparison of pressure drop between (a) SDPHE and LFD PHE, and (b) SDPHE and TDPHE.

4.6 Thermal—Hydraulic Performance Evaluation Criteria

Fig. 17 presents the PEC for the LFD PHE configuration as a function of Re_c at two hot-side Reynolds numbers, $Re_h = 5500$ (minimum) and $Re_h = 29,000$ (maximum). The PEC values generally lie in the range of 1.02–1.17, depending on the cold-side Re and the inlet temperature of the hot water. In Fig. 18a, the PEC achieves a maximum of 1.17 at $T_{h,i} = 70^\circ\text{C}$ and $Re_c = 6000$, corresponding to a 17% improvement over the smooth configuration, which is considered the optimum operating point. In contrast, at $T_{h,i} = 50^\circ\text{C}$, the maximum PEC is only ~ 1.08 . As Re_h increases to 29,000 in Fig. 18b, PEC values approach unity, with a peak of ~ 1.10 at 70°C ($\sim 10\%$ enhancement). At low inlet temperatures (50°C) combined with high Re_c , PEC can reduce to nearly 1.0, indicating that the additional surface area provided by low integral fins offers negligible benefit once turbulence has fully developed. These results emphasize that fins are most effective at moderate Reynolds numbers and higher inlet temperatures, while their contribution diminishes under fully turbulent operating conditions. The PEC behavior in the flow transitional region agreed with the previous study [36].

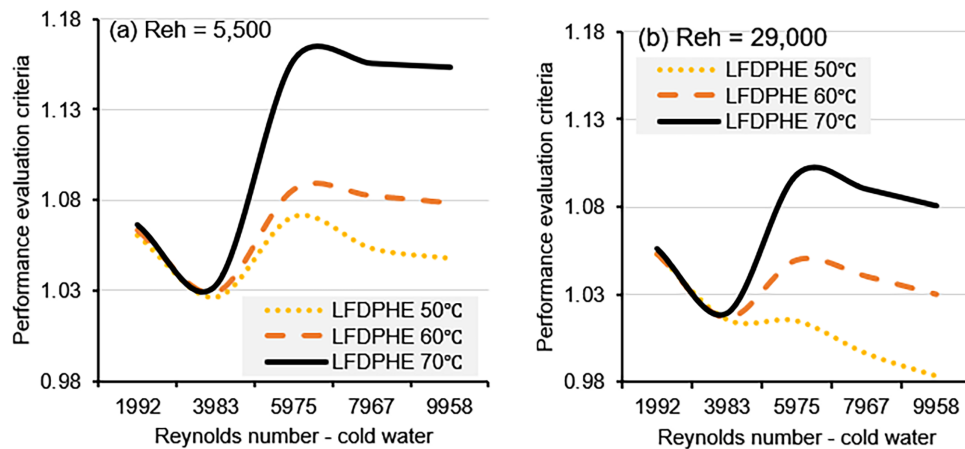


Figure 17: Performance evaluation criterion for LFD PHE (a) at $Re_h = 5500$ and (b) at $Re_h = 29,000$.

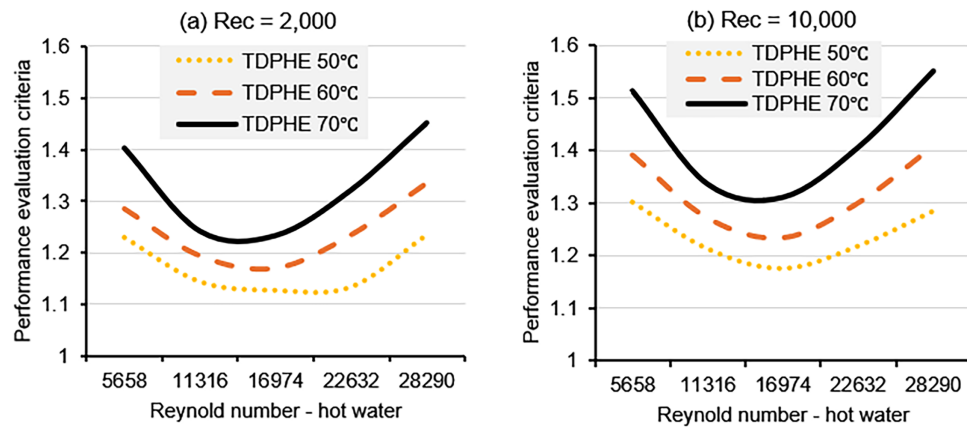


Figure 18: Performance evaluation criterion for TDPHE (a) at $Re_c = 2000$ and (b) at $Re_c = 10,000$.

Fig. 18 illustrates the PEC for TDPHE. The PEC values are consistently higher than those of the smooth and finned models, ranging from 1.12 to 1.55 depending on Re_c and inlet temperature. At $Re_c = 2000$, PEC values begin at ~ 1.12 , corresponding to a 12% improvement, and increase to ~ 1.45 at $T_{h,i} = 70^\circ\text{C}$, indicating a 45% enhancement over the baseline. Under higher turbulence conditions ($Re_c = 10,000$), twisted tapes deliver their strongest performance, with PEC peaking at ~ 1.55 for $T_{h,i} = 70^\circ\text{C}$ and $Re_h = 29,000$, equivalent to a 55% enhancement, which is considered the optimum operating condition. These results confirm that swirl-induced turbulence and secondary mixing generated by twisted tapes substantially increase convective heat transfer, even with the associated frictional penalties, especially in fully turbulent flow regimes. The PEC behavior agreed with the previous study [37].

While the implementation of passive enhancements increases entropy generation as shown in Figs. 11 and 12, this thermodynamic penalty is justified by the gains in the overall thermal-hydraulic performance, as quantified by the PEC values in Figs. 17 and 18. As demonstrated by the current study, the net thermal-hydraulic performance ($PEC > 1$) demonstrates that the enhancement in heat transfer is effectively more than the increase in the irreversibility thereby validating these techniques for practical industrial optimization.

5 Conclusion

This study provided a methodical comparison of two passive thermal enhancement techniques, namely circular fins and twisted tape inserts in double-pipe heat exchangers. The study clarifies the advantages and disadvantages of increasing heat transfer, introducing friction, and making thermodynamics processes irreversible. Based on the results presented, we can conclude the following:

Low-finned DPHE shows some improvements, with ΔT_c and Q increasing by 19% and 11%, respectively, with PEC values reaching 1.17.

A twisted tape insert in the SPHE outperforms fins for moderate to high Reynolds numbers. ΔT_c and Q increase up to 26% and 31%, respectively, with PEC values reaching 1.55. In comparison to smooth pipes.

The highest PEC values for low-finned double-pipe heat exchangers range from 1.02 to 1.17, with a hot-water Reynolds number of 5500, cold-water Reynolds numbers ranging from 2000 to 10,000, and an inlet hot-water temperature of 70°C .

The highest PEC values for a twisted tape insert in the DPHE range from 1.5 to 1.55, with a cold-water Reynolds number of 10,000 and a hot-water Reynolds number ranging from 5500 to 29,000, with an inlet hot-water temperature of 70°C .

Twisted tape generates the most entropy (20%–82%), while low-integral fins generate less entropy (15%–70%) by increasing entropy.

The pressure losses for low-finned pipe increased by a factor of 1.24. While, for twisted tape increased it by a factor of 4.

When it comes to actual use, whether to use fins or twisted tapes depends on the conditions in which they will be used. Low fins perform better in low-flow or transitional regimes because they give you predictable gains with few penalties. Twisted tapes, on the other hand, are the best choice when there is a lot of turbulence and flow. Even if they cost more to flow through, they are more efficient at transferring heat.

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Ethics Approval: Not applicable.

Conflicts of Interest: The authors declare no conflicts of interest.

Nomenclature

Symbols

C_p	Specific heat (J/kg·K)
D_h	Hydraulic diameter (m)
f	Friction factor (-)
h	Heat transfer coefficient (W/m ² ·K)
k	Thermal conductivity (W/m·K)
$\dot{m}$	Mass flow rate (kg/s)
Nu	Nusselt number (-)
P	Pressure (Pa)
Q	Heat transfer (W)
Re	Reynolds number (-)
T	Temperature (K)
u	Velocity (m/s)

Greek Symbols

μ	Dynamic viscosity (Pa·s)
ρ	Density (kg/m ³)
ΔP	Pressure drop (Pa)
ΔT	Temperature difference (K or °C)

Subscripts

avg	Average
c	Cold
f	Entropy due to friction
gen	Entropy generation
h	Hot
ht	Entropy due to heat transfer
i	Inner
o	Outer

Abbreviations

CFD	Computational Fluid Dynamics
DPHE	Double Pipe heat exchanger
DP	Pressure drops
HEX	Heat exchanger
HT	Heat transfer
LFDPE	Low-finned double pipe heat exchanger
PEC	Performance Evaluation Criterion
SDPHE	Smooth double pipe heat exchanger
TDPHE	Double-Pipe Heat Exchangers with Twisted Tape Inserts
TT	Plain twisted tape
TPF	Thermal performance factor

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