YTUP

Journal of Thermal Engineering
https://jten.yildiz.edu.tr/
https://doi.org/10.47481/jten.0081

RESEARCH ARTICLE

Effect of diamond-cut twisted tapes on thermal performance of double pipe heat exchangers: a numerical approach

Siva Krishna Turlapati[1,*], Prasad L. S. V.[2], Poornodaya Venkata Krishna Varma Kola[3]

[1]Department of Mechanical Engineering, Andhra University College of Engineering (A), Visakhapatnam, Andhra Pradesh, 530003, India
[2]Department of Mechanical Engineering, Andhra University College of Engineering (A), Visakhapatnam, Andhra Pradesh, 530003, India
[3]Department of Mechanical Engineering, Raghu Engineering College, Visakhapatnam, Andhra Pradesh, 530003, India

Abstract

Double-pipe heat exchangers are typically employed in a variety of applications, including refineries and waste heat recovery systems. They are generally used for light-duty applications. They have limitations for heavy-duty applications. Heat transfer must be intensified to overcome this problem. Twisted tapes with diamond cuts of different sizes are inserted into the heat exchanger tubes to improve heat transfer. This creates added areas that disrupt the flow and help increase turbulence, thus promoting better mixing of the fluid. The diamond-cut twisted inserts have a constant twist ratio of 3 and varying scale ratios of 25, 50, and 75. The outcomes suggest that the Nusselt number has increased by 81.43% and the friction factor has increased by a factor of 1.01 compared with a plain tube. The maximum thermal performance factor is found to be 1.3 for a scale ratio of 50. The diamond-cut twisted tapes produced strong swirls, periodic boundary layer disruption, and controlled flow separation and reattachment. These changes increased turbulence, thereby intensifying heat transfer. Generalized correlations were developed for the Nusselt numbers and friction factors.

Keywords: Intensification, diamond cut twisted tapes, Reynolds number, turbulence, friction factor

Cite this article as: Turlapati, S. K., Prasad, L. S. V., & Kola, P. V. K. V. (2026). Effect of diamond-cut twisted tapes on thermal performance of double pipe heat exchangers: A numerical approach. Journal of Thermal Engineering, 12(6), 2018–2029. https://doi.org/10.47481/jten.0081

1. Introduction

In industry, heat exchangers are employed for heating or cooling. They are essential in regulating the temperatures and thus preserve the longevity of the equipment [1], [2]. They play a key role in enhancing energy efficiency in industrial operations. They are used in various applications like food processing, beverages, pharmaceuticals, oil and gas processing etc., [3], [4]. Considering the various factors like reduction in energy resources, depletion of fossil fuels and perfecting energy consumption, there is a necessity for the augmentation of heat transfer in the heat exchangers [5]. With the advent of new technologies, higher-efficiency thermal systems are needed to meet energy standards and goals. Also, the focus should be on reducing the pressure drop. An increase in velocity increases heat transfer. However, it also increases the pressure drop and so the power needed for pumping. Hence, the aim is not only to increase the heat transfer rate but also to reduce the pressure drop. Augmentation of heat transfer refers to enhancing heat transfer by applying fundamental principles of heat transfer, with minimal modifications to the equipment. By augmenting the heat transfer, time of heat transfer can be reduced, heat exchanger's size can be minimized and moreover, efficiency of heat transfer can be improved [6], [7]. Numerous approaches were available for augmenting heat transfer in heat exchangers under both turbulent and laminar flow conditions. To enhance heat transfer, many researchers have used extended surfaces [5], twisted inserts [8], rod inserts [4], and artificial roughness.

Passive methods intensify the heat transfer either by the principle of increase in the surface area or by creating turbulence or swirl and back flow, blocking the flow path or by disrupting the channel of the fluid flow [9], [10], [11], [12]. They enhance the fluid flow's residence time in the HE and thus reduce the fluid's centerline velocity. This, in turn, increases the rate of heat transmission. Increasing the base fluid's TC is another

*Corresponding Author
E-mail Adress: sivak346@gmail.com

Submitted: 3 October 2025; **Accepted:** 9 May 2026

This paper was recommended for publication in revised form by Editor-in-Chief Ahmet Selim Dalkılıç

method for enhancing heat transfer. Because passive strategies for intensifying heat transfer are compact and easy to adopt, they have become increasingly important in heat exchangers. Some researchers used the principle of increasing the base fluid's TC to enhance heat transfer in heat exchangers. Initially, researchers conducted experiments to augment the base fluid's TC by adding solid particulates. However, flow clogging in the mini heat exchanger channels and an increased pressure drop in those channels restricted the use of these micro-sized solid particles in heat transfer applications. However, with the experiments of [13] nanoparticles were suspended in the carrier fluids and the intensification in the TC of the base fluids was successful. Choi coined the term 'nanofluids'. Thereafter, many researchers carried out their research on intensification of heat transfer utilizing mono, hybrid and ternary nanofluids [14]. Even research is moving towards pico-nanofluids. Few researchers have conducted experimental and numerical investigations into the intensification of heat transfer using mono- and hybrid nanofluids. Experimental investigations were carried out on compact heat exchanger of plate type by circulating Al_2O_3 nanofluids [15]. They have reported that the Al_2O_3 nano fluids show higher heat transfer coefficient than the original fluid. Hence, they are better heat-transfer fluids than water. Similar work was carried out by [2] by circulating Cerium Oxide nanofluids in a TPHE. They have reported an enhancement in Nu, but with a penalty in the friction factor. [16] performed their experimental investigations on a TPHE circulated with TiO_2/CuO hybrid nanofluids and proved them as the better heat transfer fluids than the mono nanofluids.

Modifying the dimensions of the heat exchangers to augment the surface area for heat transfer is not a very promising method. Hence, various types of inserts are introduced into the tubes of the heat exchangers to alter flow patterns. Surface modifications in the inserts' geometry can enhance the rate of heat transfer. Many researchers conducted experimental and numerical analyses to investigate the effect of changing the inserts' geometry on Nu, friction factor and thermal performance factors. [17],[18] performed numerical simulations by inserting twisted tapes of rectangular cut in a circular pipe. They reported that inserting rectangular-cut twisted tapes improved the mixing of the fluid and thus enhanced the centrifugal forces at the wall. This has increased the heat transfer. This has also increased the friction factor. [19] performed investigations on a TPHE by inserting twisted tapes of trapezoidal cut and circulated with aluminum oxide nanofluids. According to their findings, the integrated effect of the aluminum oxide nano fluids and trapezoidal-cut twisted tapes has enhanced the heat transfer coefficient. However, there was a frictional factor in the form of a penalty.

Based on the literature, intensifying heat transfer in heat exchangers is necessary, and the exchangers must be compact and have minimal pressure loss. A lot of research has been conducted to enhance heat transfer in heat exchangers. Various methods like enhancement of thermal conductivity or enhancing turbulence have been used by various researchers to increase the cooling efficiency in the heat exchangers as said by [20], [21]. However, in practice, this area requires substantial improvement in heat transfer. Various types of twisted tapes have been used to augment heat transfer. Twisted tapes used include simple, full-length, varying-length, short-length, regularly spaced, with attached baffles, perforated, and serrated designs. The researchers investigated various factors of twisted tapes, including twist ratio, depth ratio, clearance ratio, width ratio, and perforated diameter ratio. Researchers have also investigated cut twisted tapes of various shapes. Out of which rectangular, u, perforated or slotted tapes, corrugated tubes with twisted tapes [17],[18], [22], [23],[24] are considered and an ample amount of enhancement has been achieved with these types of cut twisted tapes. These straight-edged cuts can generate two-dimensional disturbances. They have the advantage of being simple to manufacture. They provide separation along the cut edges. The separation is strong but localized and can form large wake regions downstream. Also, flow energy dissipates quickly. This results in a higher friction factor and limited useful mixing. Also, flow reattachment occurs far downstream. This allows long recirculation zones to exist and large stagnant regions to form. Heat transfer can be uneven and localized. The boundary layer grows steadily downstream and within the recirculation zones. This lowers the temperature gradient at the wall. Boundary-layer disruption occurs only at the obstacle's leading edge, resulting in a limited frequency of enhancement. Thermal performance factors will be low for regular cut geometries. In contrast, inclined cut geometries offer good separation and attachment of the flow. As a result, the separation and reattachment points act as heat transfer hotspots. The leading edge of the diamond can cause flow separation. The inclined edges can deflect the fluid laterally and radially. The trailing edge can cause reattachment. The repetition of this cycle can enhance heat transfer and result in highly localized heat transfer coefficients. Since their flow blockage ratio is lower, they show a lower friction factor. Also, because of the alternating converging and diverging passages, they can create three-dimensional flow redirection in addition to the circumferential swirl. Thus, diamond-cut twisted tapes can provide a primary swirl through tape twisting, secondary vortices at the corners of the diamond cut, and micro vortices at the inclined and sharp edges. Each diamond cut can behave like a micro diffuser-nozzle pair. On the converging side, they can enhance convection, while on the diverging side they can create turbulence regeneration. This is not possible with plain twisted tapes and regular geometric cuts. Hence, the thermal performance factor also increases with the diamond-cut twisted tapes. Considering the above factors, the problem addressed in this work is the limited thermal performance of conventional double pipe heat exchangers due to weak fluid mixing and boundary-layer development. While twisted tape inserts enhance heat transfer by generating swirl, they often cause a high friction factor. Diamond-cut twisted tapes can further improve heat transfer by inducing added turbulence and disrupting the boundary layer. However, the effect of the diamond-cut scale ratio on thermo-hydraulic performance has not been systematically studied. This study aims to evaluate the influence of different scale ratios of diamond-cut twisted tapes on heat transfer enhancement, the friction factor, and the thermal performance factor in a double-pipe heat exchanger, to find a best configuration.

Unlike earlier studies, which primarily focused on geometric cut variations at fixed proportions, the present study introduces diamond-cut twisted-tape inserts with systematic scale-ratio variation (25, 50, and 75) at a constant twist ratio, an approach that has not been comprehensively reported in prior numerical investigations. The scale ratio directly controls the interaction length and vortex regeneration frequency, offering an added and independent design parameter for performance optimization. Most existing twisted tape studies emphasize maximum heat transfer enhancement without adequately addressing the trade-off between enhancement and friction factor. In contrast, the present study evaluates thermo-hydraulic performance using the Thermal Performance Factor (TPF) as the primary optimization metric, enabling the identification of a best configuration (scale ratio = 50) instead of merely reporting peak Nusselt numbers. Furthermore, the diamond-cut geometry introduces multi-directional secondary vortices and periodic flow reattachment zones, which differ fundamentally from conventional V-cut or perforated inserts that typically generate uni-directional swirl. This results in improved boundary-layer disruption with comparatively moderate friction penalties, as proved by the superior TPF values.

2. Numerical simulation

Analyzing the complex behavior of the turbulence generated inside the heat exchanger's tubes is essential. By analyzing flow behavior inside the tubes of the HE, one can characterize the turbulence, temperature, velocity distribution and friction factor. The numerical simulations were performed using the commercial CFD software package ANSYS Fluent (version 16.0), which employs the finite-volume method to discretize the governing equations. The software was used under a valid institutional license. The length of both tubes of the HE is considered is 1000mm. The HE has two annuli and two tubes, each with a U-bend. The lengths of the annuli are the same as those of the HE tubes. The tubes of the HE is bent into a U, forming a U-bend heat exchanger at one end. The tubes are made of copper, and the annuli are fabricated from mild steel. To prevent heat loss to the environment, asbestos rope is wrapped around the annuli. Schematic representation of the twisted tapes with diamond cut, the physical model of the TPHE, and the physical model of the twisted tapes with diamond cut are shown in Figure 1(a), (b), and (c). Dimensions of the diamond-cut twisted tapes are length (L=1000mm), thickness (t=1.6mm), twist ratio (h/d=3), and diameter (D=15mm). The twisted tapes with a constant twist ratio (h/d=3) and scale ratios (25, 50, and 75) were developed using SolidWorks.

The numerical models of PT, including PT fitted with diamond-cut twisted tapes of distinct scale ratios, are meshed with poly hexa-core mesh elements, as this mesh provides high-quality hexahedral elements in the bulk region. It is a hybrid meshing approach that uses Cartesian cells in the core and tetrahedral cells next to the boundaries. These mesh elements allow a reduction in the total face count while keeping the same grid resolution. This type of mesh has 19% fewer elements than the polyhedral mesh. The solution time for the poly hexa-core mesh is 47.09% faster than that of the polyhedral mesh. It also requires 34% less RAM. Because of the above advantages, poly hexa-core elements of mesh have been used for the meshing of the numerical model [25], [26], [27]. Water is the working fluid in the domain under analysis. The numerical procedure followed for the work is illustrated in Figure 2 The geometrical parameters of the diamond-cut twisted tapes with distinct scale ratios are depicted in Figure 3.

3. Numerical and mathematical models

3.1. Governing equations

Three-dimensional numerical models were used for simulations. Certain assumptions were considered in the analysis. Flow is assumed to be turbulent, and the working fluid is regarded as incompressible. The Momentum and continuity equations are solved using Equations (2) and (1). The energy equation is solved using Equation (3). [28] respectively. The K-Ɛ turbulence model was used to simulate the turbulent flow characteristics in the heat exchanger tubes.

The flow inside the double pipe heat exchanger with twisted tape inserts is characterized by strong swirl, secondary motion, and streamline curvature. Therefore, the SST k–ω turbulence model with curvature correction was employed in the present numerical study. This model combines the near-wall accuracy of the k–ω formulation with the robustness of the k–ε model in the free stream, making it suitable for complex swirling flows induced by twisted tape inserts. Earlier studies have also proved its superior predictive capability in enhanced heat-transfer devices showing strong rotational effects.

The twisted tape insert was modelled as an adiabatic surface, and heat conduction within the insert material was neglected. This assumption is commonly adopted in numerical studies on twisted tape inserts, as the primary heat transfer enhancement mechanism arises from flow mixing and secondary vortex generation rather than conduction through the insert. Moreover, the thermal conductivity of typical insert materials is significantly lower than that of the working fluid–wall heat-transfer pathway.

Equation for continuity

$$\frac{\partial}{\partial x_s}(\rho'' U''_s) = 0 \tag{1}$$

Equation for Momentum:

$$\frac{\partial}{\partial x''_s}('U''_s U''_t \rho') = \frac{\partial}{\partial x''_s}\left(\frac{\partial U''_t}{\partial U''_s}\mu''\right) - \frac{\partial P''}{\partial x''_t} \tag{2}$$

Equation for Energy:

$$\frac{\partial}{\partial x''_s}('U''_s T'' \rho') = \frac{\partial}{\partial x''_s}\left(\frac{K''}{C_p}\frac{\partial T''}{\partial x''_s}\right) \tag{3}$$

ρ” denotes the density of the working fluid. U” denotes the velocity part. P” stands for pressure. Dynamic viscosity is denoted by μ”. Cp stands for the specific heat. K" is the fluid's TC, and T" shows the fluid's temperature. Subscripts k and s denote the k and s directions.

3.2. Boundary conditions

In the outer tube of the double-pipe heat exchanger, hot fluid (water) is allowed to circulate. The cold fluid (water/nanofluid) is circulated through the inner tube of the HE. At the tube's inlet, the hot fluid is kept at a consistent temperature of 333K. The temperature of the cold fluid, which is kept constant at the tube inlet, is 303K. While the cold fluid's flow rate varies between 3 and 15 LPM, the hot fluid's flow rate is kept at 6 LPM. The Velocity is assumed to be uniform at the inlet. At the tube's outlet, the pressure is selected. A uniform velocity profile corresponding to the specified Reynolds number was imposed at the inlet. To ensure hydrodynamically developed flow, an entrance length was provided upstream of the heated section. A pressure-outlet condition was used to model the outlet. No-slip boundary conditions were applied at all solid surfaces. The outer wall of the inner tube was subjected to a constant heat flux, while the twisted tape insert was assumed to be adiabatic. The periphery of the twisted tapes is chosen to have a no-slip, adiabatic wall condition. The Material of the outer pipe is chrome steel. Copper is used for the inner pipe. Two tubes are found inside a single tube and are connected by a U-bend, forming a twin-pipe HE. The outer diameter of the outer tube is 63mm. The outer tube's inner diameter is 60mm. The thickness of the tube being 3mm. The outer diameter of the inner tubes is 17 mm. The inner tube's diameter is 15mm. The thickness of the tube is 2mm. The tubes' length being 1400mm. The outer tube is treated as an adiabatic wall to prevent losses to the environment. Automatic mesh generation was used. Conduction is considered in the walls of the inner and outer tubes. Hence, they are not modelled and meshed separately. Convection is considered in the fluid-structure interaction. Mesh refinement is performed using an advanced size function. This is done to capture the heat transfer at the boundary. Also, this is being used because the model is cylindrical. A no-slip boundary condition was applied at the solid–fluid interfaces (inner surfaces of the tube walls and the twisted tape). The thermophysical properties of the working fluid were assumed to be constant and evaluated at the bulk mean temperature. The inner surface of the tube is the only heated surface and is subjected to a uniform heat flux. All surfaces of the diamond-cut twisted tape are treated as adiabatic walls. Heat conduction within the tape and heat exchange between the tape and the fluid are neglected. The tape acts purely as a hydrodynamic and thermal boundary-layer disruptor. The outer wall of the tube is assumed to be thermally insulated(adiabatic), ensuring that heat transfer occurs only from the inner tube wall to the fluid.

3.3. Numerical method

Eddy-viscosity models two-equation such as k-ε and k-ω are generally used for the analysis of heat exchangers fitted with twisted tapes. Because the Reynolds number is above 2000, the K-Ɛ model was chosen as the turbulence model. However, there are limitations to using this model. They are insensitive to flows characterized by streamline curvature, flow separation, recirculation, and rotational effects. The diamond-cut twisted tape inserts generate strong swirl, secondary flows, and streamline curvatures, which are known to cause overprediction of turbulence production in conventional eddy-viscosity models. Curvature correction improves prediction accuracy by suppressing excessive turbulence production in regions of high curvature and rotation. To suit them for the analysis of problems related to tubes fitted with the twisted tapes, Spalart–Shur curvature correction applied to the SST k–ω turbulence model with enhanced wall treatment was considered for low Reynolds numbers as mentioned by [29], [22]. The researchers also suggest models that use Reynolds stress with curvature-damping corrections. However, because of their complexity and associated computational costs, they are not widely used. Hence, the K-Ɛ model was found to be suitable for the present problem. The k–ε model has known limitations for swirling, separated, and curvature-influenced flows. Even though the standard K-Ɛ turbulence model shows certain limitations in accurately predicting strong swirl, separation, and curvature flows, it can be used for twisted tapes that induce moderate and steady swirl rather than in cases of strong vortex breakdown or large unsteady separation. No large recirculation bubbles were seen, and the flow remained attached to the tube wall along its length. Hence, in this case, the limitations of K-Ɛ turbulence model under strong swirl conditions were not predominant. Also, the swirl number remained within the range typical of eddy-viscosity models. Moreover, the major aim of this study is to evaluate the relative thermo-hydraulic performance trends for the considered configurations rather than resolve anisotropic turbulence structures. K-Ɛ can reliably capture global quantities, such as average heat transfer and the friction factor because they are less sensitive to higher-order Reynolds-stress anisotropy. Various researchers used K-Ɛ model for twisted tapes and changed twisted tapes with an acceptable accuracy of within ±10–15%. By contrast, the advanced turbulence models such as SST K-ω and Reynolds stress models require finer near-wall meshes and exhibit convergence sensitivity for complex insert geometries. The K-Ɛ model offers a robust balance between accuracy and computational economy. Moreover, the limitations of K-Ɛ model can be mitigated using enhanced wall treatment keeping appropriate y^+ values and careful grid independence testing. Hence, the K-Ɛ model was used to simulate the characteristics of the turbulent flow in the heat exchanger tubes. The models chosen for the solution were simple. Second-order accuracy is set for momentum, pressure, and energy. 10^{-3} was set for the convergence for mass as well as the momentum. However, 10^{-5} was set for the energy's convergence. The y^+ values considered are 0.5-1 for scale ratio of 25, 1-3 for scale ratio of 50 and 2-5 for scale ratio of 75. The skewness value was 0.31. The present model is considered a conjugate heat transfer model. Hence, solid domains are not meshed. Heat transfer is modelled using a constant wall heat flux or a constant wall temperature. Solid conduction was either implicitly accounted for or neglected. The steady-state energy equation was solved only in the fluid domain. A uniform heat flux was imposed at the tube wall. The surfaces of the diamond-cut twisted tape insert were treated as adiabatic (zero heat flux), and no heat conduction within the twisted tape material was considered. Thus, heat transfer occurred solely between the tube wall and the fluid, while the twisted tape influenced the thermal field indirectly through flow mixing, swirl generation, and boundary-layer disruption.

3.4. Study of grid independence

The grid-independence test confirms the model's accuracy by using smaller cell sizes. The standard procedure in CFD is to start with a coarse mesh and progressively increase its size. Carry out the simulations with coarse, medium, and fine meshes. When the mesh is refined and simulations are performed, grid independence is achieved if the output values change by less than predefined minimum values. A grid-independence study for PT was performed at hot and cold fluid flow rates of 9 LPM and 6 LPM, respectively, by changing the element size in the cell. The mesh was changed from course to medium. Results are verified with respect to the changes. A slight variation in the results was seen. Then the coarse mesh is replaced with a fine one. Again, the change in the results was seen. Changes in the outlet temperatures were seen when the grid was switched from coarse to medium. However, little variation in the outlet temperatures was seen when the medium mesh was switched to fine mesh. Changing from a medium to a fine mesh results in a temperature variation of less than one degree. It shows that the outlet temperatures are independent of the selected grid. This shows that this mesh is suitable for simulations. Hence, the fine mesh is selected for conducting the simulations. Initially, the model is meshed using a coarse mesh having 11,20,000 elements. At the outlet, the impact of this mesh on the cold fluid's temperature was examined during the simulations. At the cold fluid's temperature was changing. Eventually, the coarse mesh is replaced by a medium mesh forming 40,20,000 mesh elements. A shift in the cold fluid's outlet temperature is still seen. Next, the medium mesh is switched to a fine mesh. With a fine mesh, the model has almost 6,000,000 elements. When the medium mesh is switched to the fine mesh, the cold fluid's outlet temperature does not vary significantly. The temperature difference is less than one degree. Hence, all the simulations are carried out using fine mesh. The meshed model of the diamond-cut twisted tapes inserted in the TPHE and the grid-independence study are illustrated in Figure 4(a) and (b). The Grid Convergence Index method adopted for this work is illustrated in Table 1. The Variation in outlet temperature with changes in mesh size is shown in Table 3. During the iterations, the monitored global quantities are presented in Table 2. The percentage deviation across grid levels for the Nusselt number is 0.77%. The friction factor is 2.56%. For a scale ratio of 25, the deviations are 2% for Nu and 3% for the friction factor. For a scale ratio of 50, the uncertainty is 1.5% for Nu and 3% for the friction factor. For a scale ratio of 75, the deviation is 1% for Nu and 2% for the friction factor. The grid-induced numerical uncertainty was estimated from the difference between the two finest grids. The grid uncertainty for Nu is ±0.8% and for the friction factor is ±2.6%. Residual convergence below 10^{-6} for energy and below 10^{-4} for momentum was achieved. Monitoring of the surface-averaged Nusselt number showed fluctuations below 0.3%, which were taken as the iteration uncertainty.

The uncertainty in the numerical predictions primarily arises from discretization errors, turbulence modelling assumptions, and thermophysical property correlations. Following standard numerical uncertainty estimation procedures, the overall uncertainty in the predicted Nusselt number and friction factor was evaluated by combining individual uncertainty sources using the root-sum-square (RSS)method. Based on grid refinement studies and turbulence model sensitivity, the estimated uncertainties were found to be within ±6.5% for the Nusselt number and ±7.2% for the friction factor over the investigated Reynolds number range.

A grid convergence study was conducted using three successively refined structured meshes consisting of approximately 1.12 million, 4.02 million, and 6 million cells, with a uniform refinement ratio of 1.3. The Grid Convergence Index (GCI)was evaluated following the ASME procedure. The observed order of accuracy was 1.92. The GCI for the finest grid was 1.8% for the Nusselt number and 2.3% for the friction factor, showing that discretization errors are minimal and the numerical solution is grid independent. The agreement between numerical and reference data was quantified using mean absolute percentage error (MAPE), which was found to be 6.3%, corresponding to a confidence level exceeding 93%.

3.5. Validation

Simulations are carried out on the 3D numerical model of the TPHE, initially using water as the heat-transfer medium, i.e., the plain tube (PT). Later simulations were carried out by inserting diamond-cut twisted tapes into the tubes of the HE and using water as the heat-transfer medium. Simulations were performed at distinct water flow rates (3 to 15 LPM). The Reynolds numbers varied from 4586 to 22932. Throughout the analysis, the twist ratio is kept constant (twist ratio h/d=3) and scale ratios are varied to 25, 50, and 75, respectively. This was decided for the above-mentioned cases, and then parameters such as Nusselt numbers (Nu) and friction factors were assessed from equations available in the literature. Nu is calculated using Equation (11). The friction factor is found in Equation (12). Initially, the Nu data for the PT was confirmed against correlations available in the literature. The Nu data of the PT were compared with the Gnielinski's correlation [30] (Equation (13)) and the Notter and Sleicher correlation [31] (Equation (15)). The friction factor (f) data of the PT were differentiated using the correlations of Equation (16) of Blasius [32] and Petukhov [33] equation of Equation (17). The outcomes from the simulations for PT were compared with the results obtained from the experiments. Figure 5 shows how the Nu's data from the CFD for plain tube is Compared with the standard Gnielinski's correlation [30] as well as the correlation suggested by Notter and Sleicher [31]. Similarly, Figure 4 illustrates the comparison of the data of the f for PT obtained from CFD with that of the standard correlations of Blasius [32] and Petukhov [33]. The experimental and CFD results differ by 7.52%, which is well within the allowable range. Thus, these data can be used to analyze HE with high confidence. Figure 6(a) compares the Nu data obtained from the simulations with that obtained from the Gnienlinski and Notter Rouse correlations. Similarly, Figure 6(b) compares FF data from the simulations with FF data from the correlations of Blasius and Pethukov.

3.5.1. Experimental setup and validation of experimental results

Nusselt numbers and friction factors were decided experimentally. The experimental setup used for this purpose is illustrated in Figure 7. The setup forms two circuits. One is the hot-water circuit and the other is the cold-water circuit. It forms two outer tubes and two bent inner tubes. The inner tubes are centrally placed within the outer tubes. Diamond-cut twisted tapes are placed inside the inner tubes. Hot water is allowed to flow inside the outer tubes and through the inner tubes along with the inserted diamond-cut twisted tapes. Two tanks, one for circulating hot water and the other for circulating cold water, are provided in the setup. Two valves are used to control the flow through the setup, and two are used to bypass the water in the setup. Two flow meters are used to regulate flow within the setup on both the hot- and cold-water sides. Thermocouples are used to decide the inlet and outlet temperatures of the system. A U-tube manometer was used to measure the differential pressure on the cold side of the system. Figure 7(a) compares Nu data from simulations with Nu data from experiments. Similarly, Figure 7(b) compares FF data obtained from CFD with FF data obtained from the experiments. The simulation data deviated from the experimental data by 9.3%. A 5.6% deviation was seen between simulation and experimental data. Equation (4) can be used to assess heat transfer from the heated fluid. Equation (5) can be used to assess how much heat the cold fluid has absorbed.

$$\dot{Q}_{hf} = \ddot{m}_{hf} C_{p,hf} (T_{hfi'} - T_{hfo'}) \tag{4}$$

$$\dot{Q}_{cf} = \ddot{m}_{cf} C_{p,cf} (T_{cfo'} - T_{cfi'}) \tag{5}$$

Where m_{hf} represent the hot fluid's rate of flow and $\ddot{m}_{cf}$ represent the cold fluid's rate of flow. Both the hot as well as the cold fluid's specific heats are denoted by $C_{p,hf}$ and $C_{p,cf}$. Hot fluid's inlet temperature at the heat exchanger's tubes is denoted by T_{hfi}' and at the exit it is denoted by T_{hfo}'. Similarly, Cold fluid's inlet temperature at the heat exchanger's tubes' is denoted by Tci and at the outlet it is denoted by T_{co}'. Average heat transfer between the cold as well as the hot fluids can be estimated with Equation (6). The inner heat transfer coefficient, Ui can be found using the Equation (7).

$$Q_{avg'} = \frac{\dot{Q}_c + \dot{Q}_h}{2} \tag{6}$$

$$Q_{avg'} = U_{i'} A_{i'} \Delta T_{LMTD'} \tag{7}$$

Where U_i' is the inner heat transfer coefficient. A_i, is the inner tube's area of the and $\Delta T_{LMTD'}$ is the LMTD. Where Equation (8) can be used to estimate Ai. Overall heat transfer coefficient can be assessed using the Equation (9).

$$A_{i'} = \pi dl \tag{8}$$

$$\Delta T_{LMTD'} = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}}$$

$$\Delta T_1 = (T_{hi'} - T_{co'})$$

$$\Delta T_2 = (T_{ho'} - T_{ci'})$$

$$U_{i'} = \frac{Q_{avg'}}{A_{i'} \Delta T_{LMTD'}} \tag{9}$$

The inner heat transfer coefficient can be found from Equation (10).

$$\frac{1}{U_{i'}} = \frac{1}{h_{i'}} + B \tag{10}$$

Where the constant B can be found using Wilson chart, which is drawn between $\left(\frac{1}{U_{i'}} \text{vs} \frac{1}{Re^{0.8}}\right)$ from the intercept on y-axis. Equation (11) and Equation (12) can be used for assessing the friction factor and Nu for the PT for flow rates 3 LPM to 15 LPM can be obtained using.

$$Nu = \frac{h_i d_i}{k} \tag{11}$$

$$f = \frac{\Delta T}{\frac{1}{d_i}\left(\frac{\rho v^2}{2}\right)} \tag{12}$$

$$Nu' = \frac{\frac{f}{2}(Re - 1000) Pr}{1.07 + 12.7\left(\frac{f}{2}\right)^{0.5}\left(Pr^{\frac{2}{3}} - 1\right)} \tag{13}$$

Using Equation (14), the friction factor f may be decided.

$$f = (1.58 \ln(Re) - 3.82)^{-2} \tag{14}$$

In the above expression, the range of Re is from 2300 to 5×10^6 and Pr is within the range of 0.5 to 2000.

The equation suggested by Notter Rouse is given in Equation (15).

$$Nu = 0.015 (Re^{0.856})(Pr^{0.347}) + 5 \tag{15}$$

This expression is valid over the range of $2300 < Re < 5 \times 10^6$ and $0.5 < Pr < 2000$

Equation (16) presents the correlation proposed by Blasius, while Equation (17) presents the expression proposed by Petukhov.

$$f = (Re)^{-0.25} \times 0.3164 \tag{16}$$

This equation applies to the range of $3000 < Re < 10^5$.

$$f = (0.790 \ln(Re) - 1.64)^{-2} \tag{17}$$

Equation (17) works over the range of $2300 < Re < 5 \times 10^6$

$$TPF = \frac{(Nui/Nup)}{(fi/fp)^{1/3}} \tag{18}$$

Where TPF refers to the thermal performance factor, N_{ui} is the Nusselt number of the inserts and N_{up} is the Nusselt number of the plain tube. Whereas f_i is the friction factor of the inserts and f_p denotes the friction factor of the plain tube.

4. Results and discussions

The heat transfer area, flow rate, conduction and convection phenomena, and other variables affect the effectiveness of the HE. Also, the effectiveness depends on the flow phenomena and turbulence generation inside the tubes of HE. The greater the turbulence generated inside the heat exchanger tubes, the faster the rate of heat transfer. In this work, diamond-cut twisted tapes are used to augment heat transfer in the HE by creating added turbulence in the tubes of the HE. Since the flow phenomenon is complex, it is necessary to analyze flow behavior. It is also critical to examine how the twisted tapes affect the ability of the heat-exchanger tubes to increase heat transfer. Therefore, this work aims to examine the effects of the diamond-cut twisted tapes on the parameters such as temperature, velocity, pressure, and turbulence inside the HE tubes. Simulations were carried out on TPHE fitted with diamond-cut twisted tapes. The results obtained from the simulations are presented in the preceding sections.

4.1. Temperature contours

Simulations were performed on a TPHE to analyze temperature variation when twisted tapes with diamond cuts were inserted into the tubes of the HE. The hot fluid's flow rate stays steady at 6 LPM, while the cold fluid's flow rate fluctuates between 3 LPM and 15 LPM. The temperature of the hot fluid at the outer-tube inlet is kept at 333 K. At the inlet of the inner tubes, the temperature of the cold fluid is kept at 303 K. The temperature changes in the outer tube of the HE is analyzed and shown in Figure 8.

It illustrates the PT's temperature profiles for a 9 LPM flow rate in a TPHE when equipped with twisted tapes featuring diamond cuts of different scale ratios (25, 50, and 75), in comparison with the temperature contour of PT, i.e., when the working medium is water. The contours clearly show that, for the plain tube, the temperature varies from 333 K to 326 K. When diamond-cut twisted tapes are inserted, the temperature variation increases. The maximum temperature variation is seen as diamond-cut twisted tapes with a scale ratio of 75. The temperature varied from 333 K to 314 K. The temperature difference increased when diamond-cut twisted tapes were inserted. The turbulence within the heat exchanger tubes has increased, which accounts for this. The swirl motion produced by inserting twisted tapes intensified the turbulence. Along with this, the added turbulence is generated by the diamond cut in the twisted tapes. The increase in temperature difference may be due to the working fluid's longer retention time. The residence time is increased because the working fluid must pass around the periphery of the twisted tapes and across the diamond-cut regions on their surfaces. Another reason could be that the twisted tapes, together with the diamond-cut regions, allow the working fluid to interact more with the region near the wall. This will enhance conduction in the near-wall region and convection in the fluid. Similar trends were seen by [1], [34].

4.2. Velocity contours

Working fluid velocity significantly influences heat transfer improvement in the HE. If the axial velocity is high, the rate of heat transfer will be lower. This is because the fluid will not be kept in the heat exchanger tubes for a longer period. The working fluid's retention period will therefore increase if the working fluid's axial velocity decreases. As a result, it increases the heat transfer rate of the HE tubes. Hence, it is important to analyze the velocity contours to assess variations in axial velocity inside the HE tubes. Velocity contours for the plain tube and for scale ratios of 25, 50, and 75 are depicted in Figure 9. For the PT, little reduction in velocity is seen along the length of the HE tubes, except in the entry and exit regions of the heat exchanger's U-tube. This variation results from the U-bend of the heat exchanger, which lowers the axial velocity of the working fluid. A decrease in the axial velocity is seen at the U-bend's entry and exit in other instances where diamond-cut twisted tapes are inserted. In addition, the tubes' axial velocity decreases at the outlet where diamond-cut twisted tapes are present. Diamond-cut twisted tapes with a scale ratio of 75 showed the greatest decrease in axial velocity. Similar phenomenon was seen in [35].

4.2.1. Secondary flow visualization

The secondary flow vectors reveal the formation of counter-rotating vortices induced by the twisted tape. These vortices periodically disrupt the thermal boundary layer, thereby sustaining high local heat transfer coefficients along the tube length. Figure 11 illustrates the visualization of the secondary flow in the diamond-cut twisted tapes.

4.3. Turbulence contours

Turbulence in the flow can enhance the heat transfer rate in the heat exchanger. Hence, the more turbulence generated, the higher the rate of heat transfer. Hence, the aim is to maximize turbulence within the heat-exchanger tubes. Simulations are carried out to analyze the heat exchanger fitted with diamond-cut twisted tapes. Figure 10 illustrates the variation in turbulence in a plain tube and in a tube fitted with diamond-cut twisted tapes. It shows that turbulence is not generated in the PT because the working fluid follows a straight path. The boundary layer stays uninterrupted because no mechanism exists to disrupt the flow path. The heat transfer rate is reduced because no turbulence is created in the heat exchanger tubes. However, along the length of the twisted tapes, turbulence is created close to the tube wall when the PT is applied by inserting twisted tapes having a diamond-cut. The turbulence between the twisted tapes and the tube walls is increased by the swirl generated by the twisted tapes and by the added turbulence produced by the diamond-cut sections. Consequently, the heat transfer coefficient of the working fluid will increase, thereby accelerating heat transfer through the heat exchanger tubes. Additionally, turbulence increases fluid mixing, which improves the heat transfer coefficient. This, in turn, enhances convective and conductive heat transfer in the heat exchanger tubes. The primary enhancement in heat transfer arises from the swirl generated by twists in the tape. This is because the fluid's axial momentum is being converted into the tangential momentum. The centrifugal force generated drives high-momentum fluid from the center toward the wall and pushes the low-momentum fluid away from the wall. This suppresses the growth of the thermal boundary layer and thus keeps higher temperature gradients at the fluid–wall interface. The twists also create counter-rotating vortices, which can improve radial mixing and transverse heat transport. Approximately 50-60% of the enhancement of heat transfer can be attributed to the swirl. Diamond cuts act as a secondary heat-transfer enhancement. They provide flow separation and reattachment. This renews the thermal boundary layer repeatedly. Also, these cuts generate small-scale vortices and wake regions that interact with the primary swirl. This results in an intensification of mixing in the near-wall region. This results in an intensification of heat transfer by 15-30%. This was also seen in the works carried out by [28].

4.3.1. Turbulence intensity

The turbulence intensity was evaluated to quantify the size of velocity fluctuations. A scale ratio of 50 shows an average turbulence intensity of 12.6%, compared with 9.8% for a scale ratio of 25. While higher turbulence enhances heat transfer, it also contributes to increased friction losses, explaining the observed friction factor trend.

4.4. Pressure contours

The working fluid circulates in the heat-exchanger tubes. This exerts a certain pressure. When the working fluid is circulated, there is a pressure drop in the heat exchanger tubes. This drop in pressure will increase pumping power. This increases the friction factor. The TPF will decrease because of this increase in the friction factor. Hence, analysis of pressure contours is needed to investigate the impact of the friction factor on the TPF. Figure 11 illustrates the variation in pressure in the PT and in the PT equipped with diamond-cut twisted tapes of distinct scale ratios. It shows that for the PT, the drop in pressure ranges from 3576 Pa to 289 Pa. This trend was also clear in the work carried out by [19]. This decrease in pressure reduces TPF. Inserting diamond-cut twisted tapes, the pressure drops further increased, and the increase grew with the scale ratio. Diamond-cut twisted tapes with a scale ratio of 75 have produced the greatest drop in pressure.

4.5. Swirl strength

The swirl intensity induced by the diamond-cut twisted tape was quantified using the ratio of circumferential velocity to axial velocity (V_θ/V_z). The results show that a scale ratio of 50 produces the highest average swirl intensity, approximately 1.52 times greater than that in the scale-ratio-25 case. This enhanced rotational motion strengthens radial mixing, leading to a higher convective heat transfer rate.

4.6. Nusselt number

Figure 12 shows the distribution of numerical Nu as Re varies. It shows that as Nu increases, Re increases for PT and all scale ratios of the twisted tapes. Additionally, it was noted that as the scale ratio increased, Nu also increased. The highest Nu for the scale ratio of 75 increased by 81.43% compared with the PT. This occurs because twists in twisted tapes generate swirls. In addition, the disturbance caused by the diamond cuts provided in the surface region of the diamond-cut twisted tapes and the backflow generated by the U-bend in the tubes of the HE was seen. For the above reasons, because of swirl generation, fluid mixing has been enhanced and thus turbulence in the vicinity of the wall has increased. This, in turn, enhanced the rate of heat transfer in the HE tubes. Additionally, the diamond cuts generated disturbances, which further improved the mixing and turbulence along the entire length of the HE tubes in the near-wall regions. This resulted in improved convective and conductive heat transfer in the HE tubes. The results of this work are like that of the work executed by [36], [37]. The Nusselt numbers obtained with the insertion of twisted tapes of diamond cut are found to be higher than the Nusselt number obtained by [38].

4.7. Friction factor

Figure 13 shows how the friction factor varies with Re. It proves that, under all circumstances, the friction factor increases with Re. It was seen that as the scale ratio increases, the friction factor increases. In the case of twisted tape with a scale ratio of 75, the highest friction factor was seen. This is attributed to fluid–surface interactions in near-wall regions, which increase the pressure drop. This in turn increases the friction factor. This, in turn, increases the pumping losses. Additionally, the friction factor increases when fluid flows past the peripheries of twisted tapes and the diamond cuts on the tapes' surfaces. This results in a penalty. When the scale ratio is 75, the friction factor reaches its maximum. It is 2.01 times as great as the PT. When twisted tapes with diamond cuts are inserted, the friction factors bought are found to be lower than those obtained when twisted tapes with bowl cuts are inserted (Mande et al. 2025). General empirical correlations for the Nusselt number and the friction factor for various scale ratios (25%, 50%, and 75%) of the diamond-cut twisted tapes are formulated and presented below:

$$Nu = 0.020\, Re^{0.85}\, Pr^{0.4} (SR)^{0.28}$$

$$f = 0.4\ Re^{-0.27}(SR)^{0.42}$$

Where Nu is the Nusselt number, f is the friction factor Re is the Reynolds number, Pr is the Prandtl number. SR is the scale ratio. The proposed Nusselt number and friction factor correlations were statistically evaluated using standard regression performance indicators. The coefficient of determination (R^2), the root means square error (RMSE), and the mean absolute percentage error (MAPE) were used to assess predictive accuracy. The Nusselt number correlation yielded an R^2 value of 0.94, with RMSE and MAPE values of 3.6 and 7.1%, respectively. Similarly, the friction factor correlation achieved $R^2 = 0.95$, with RMSE = 0.0031 and MAPE = 8.4%.

4.8. Thermal performance factor

The Thermal performance factor of the plain pipe fitted with diamond-cut twisted tapes at different scale ratios was decided using [40] Eq. (18). Variations in thermal performance factors with mass flow rate for distinct scale ratios of the diamond-cut twisted tapes were plotted. The variation of thermal performance with mass flow rate is illustrated in Figure 14. It shows that, although the maximum scale ratio of 75 yields the maximum Nusselt number, the maximum TPF occurs at a scale ratio of 50. The maximum TPF is 1.3. A scale ratio of 75 produces a strong swirl and greater turbulence, while a scale ratio of 25 produces a weak swirl and less turbulence. On the other hand, the scale ratio of 25 provides a very high friction factor. Hence, the TPF is very low. At a scale ratio of 50, the strong, stable swirl produces a high Nusselt number. However, the friction factor is moderate. For this reason, the TPF is highest among the scale ratios. This was the same as in the work carried out by [39].

5. Conclusions and future scope

In this work, numerical simulations were conducted to decide the heat transfer rate, expressed as the Nusselt number, in a double-pipe heat exchanger with minimal friction losses by inserting diamond-cut twisted tapes with a constant twist ratio and various scale ratios. Initially, contours of velocity, temperature, pressure, and turbulence were analyzed. In addition, the analysis was carried out with the aim of deciding the friction factor and Nusselt number for various cases, such as a plain tube and a plain tube fitted with diamond-cut twisted tapes. The following conclusions, drawn from the analysis, are presented below:

- Velocity contours show that, in the plain tube, the streamlines are almost parallel to the tube axis and thus do not provide improved heat transfer, because the cold fluid's residence time is very short.
- However, when diamond-cut twisted tapes are inserted, the streamlines are not parallel but curved. This is due to the intersection of the flow with the diamond-cut regions of the twisted tapes.
- Pressure contours show that, for the plain tube, there is little pressure drop in the heat exchanger's tubes. However, in the case of diamond-cut twisted tapes, there is a slight decrease in pressure as measured by the friction factor.
- Temperature contours showed greater temperature differences with the insertion of diamond-cut twisted tapes. This increase in the temperature gradient has resulted in an increase in Nu.
- Turbulence profiles illustrate an increase in turbulence at the tube walls when diamond-cut twisted tapes are installed in a plain tube. This is due to the decreasing thickness of the fluid's boundary layer and the increasing surface velocity of the tube. This has also increased convection and conduction at the wall surface, resulting in added mixing of the fluid. Finally, this enhanced heat transfer.
- The Maximum Nusselt number is reached at a scale ratio of 75 and is 81.43% greater than that for the plain tube.
- The scale ratio of 75 yields the greatest friction factor, which is 1.01 times that of the plain tube.
- Generalized correlations were developed based on the CFD data.

Based on the above findings, future studies could incorporate solid conduction through the diamond-cut twisted tapes and the tube wall using conjugate heat transfer. Parametric studies can be used to optimize diamond size, cut depth, inclination angle, pitch, and blockage ratio. The performance of diamond-cut twisted tapes can also be studied under laminar flow conditions in compact heat exchangers. are useful for enhancement of heat transfer in the double pipe heat exchangers. Future work can be extended to include applications such as solar thermal collectors, waste heat recovery systems, and compact HVAC units. The novel contribution of this study is the detailed numerical evaluation of diamond-cut twisted tape inserts with varying scale ratios, an area that has not been widely explored in the literature.

Author contributions statement

Turlapati Siva Krishna: Literature review, Problem identification, Numerical Investigation, Experimental investigation, Detailed analysis of the work, Systematic arrangement of the work in the article, Proof reading. LSV Prasad: Aid in Problem identification, Analysis in Numerical Investigation, aid for experimental investigation, guidance for systematic arrangement of the work in the article, proof reading. Kola Poornodaya Venkata Krishna Varma: Literature review, aid in problem identification, aid in Numerical Investigation, experimental investigation, aid in detailed analysis, proof reading

Disclosure of interest

No potential competing interest was reported by the authors.

Funding

No funding was received.

Data Availability

Data supporting the findings of this study are available from the corresponding author upon reasonable request.

Declaration of generative AI and AI-assisted technologies in the writing process

No generative AI or AI-assisted technologies were used in the writing process.

Nomenclature

C_p	Specific heat, J/kg K
d_i	Inner diameter of the tube, m
f	friction factor
h_i	inside heat transfer coefficient, W/m^{2}K
h_0	outside heat transfer coefficient, W/m^{2}K
k	Thermal conductivity, W/m K
Nu	Nusselt number
L_t	Length of the tube, m
Re	Reynolds number
HE	Heat exchanger
HT	Heat transfer
TPHE	Twin pipe heat exchanger
HTC	Heat transfer coefficient
LMTD	Logarithmic mean temperature difference
TPF	Thermal performance factor

Greek symbols

Ø	Particle volume concentration of the nanoparticles, %
ρ_{np}	Density of the nanoparticles, kg/m^3
ρ_w	Density of water, kg/m^3
μ	Viscosity, Pa-s

References

[1] Reddy, G. S., Kalaivanan, R., Kumar, R. U., and Krishna Varma, K. P. V., 2022, "Convective Heat Transfer in Heat Exchangers Using Nanofluids: A Review," Ecol. Eng. Environ. Technol., 23(3), pp. 193–201. https://doi.org/10.12912/27197050/147679.

[2] Reddy, G. S., Kalaivanan, R., Uday Kumar, R., and Jadhav, P. H., 2024, "Influence of W-Cut Twisted Tape Inserts on Heat Transfer Characteristics of a Counter Flow Heat Exchanger Using CeO2 Nanofluid—An Experimental and Numerical Investigation," Numer. Heat Transfer A: Appl., 86(23), pp. 8425–8445. https://doi.org/10.1080/10407782.2024.2359060.

[3] Krishna Varma, K. P., Kishore, P. S., and Tulasi, T., 2017, "CFD Analysis for the Enhancement of Heat Transfer in a Heat Exchanger with Cut Twisted Tape Inserts," SSRG International Journal of Mechanical Engineering (SSRG-IJME), 4(3), pp. 46–52.

[4] Krishna Varma, K. P., Venkateswarlu, K., Nutakki, U. K., and Satya Mohan, K. S., 2023, "The Effect of Threaded Rod Inserts and Aluminum Oxide–Water-Based Nanofluids on Performance of a U-Bend Double Pipe Heat Exchanger," Journal of Nanofluids, 12(1), pp. 173–182.

[5] Stehlík, P., Jegla, Z., and Kilkovský, B., 2014, "Possibilities of Intensifying Heat Transfer through Finned Surfaces in Heat Exchangers for High Temperature Applications," Applied Thermal Engineering, Vol. 70, No. 2, pp. 1283–1287, doi: 10.1016/j.applthermaleng.2014.05.052.

[6] Ravi Babu, S., Krishna Varma, K. P., and Mohan, K. S., 2022, "Artificial Neural Network Technique for Estimating the Thermo-Physical Properties of Water-Alumina Nanofluid," Ecological Engineering & Environmental Technology, 23, pp. 1–7.

[7] Estelle, P., Halelfadl, S., and Maré, T., 2015, "Thermal Conductivity of CNT Water-Based Nanofluids: Experimental Trends and Model's Overview," Journal of Thermal Engineering, 1(2), pp. 381–390.

[8] Dandoutiya, B. K., and Kumar, A., 2022, "W-Cut Twisted Tape's Effect on the Thermal Performance of a Double Pipe Heat Exchanger: A Numerical Study," Case Studies in Thermal Engineering, 34, p. 102031.

[9] Arun, M., Barik, D., Chandran, S. S., Govil, N., Sharma, P., Khan, T. Y., Baig, R. U., Bora, B. J., Medhi, B. J., Kumar, R., and Rizal, A., 2024, "Twisted Helical Tape's Impact on Heat Transfer and Friction in Zinc Oxide (ZnO) Nanofluids for Solar Water Heaters: Biomedical Insight," Case Studies in Thermal Engineering, Vol. 56, Article No. 104204, doi: 10.1016/j.csite.2024.104204.

[10] Eiamsa-Ard, S., Kiatkittipong, K., and Jedsadaratanachai, W., 2015, "Heat Transfer Enhancement of TiO2 Water Nanofluid in a Heat Exchanger Tube Equipped with Overlapped Dual Twisted-Tapes," Engineering Science and Technology, an International Journal, 18(3), pp. 336–350.

[11] Kumar, P. M., Kumar, J., Tamilarasan, R., Sendhilnathan, S., and Suresh, S., 2015, "Review on Nanofluids Theoretical Thermal Conductivity Models," Engineering Journal, 19(1), pp. 67–83.

[12] Hussein, A. K., Walunj, A., and Kolsi, L., 2016, "Applications of Nanotechnology to Enhance the Performance of the Direct Absorption Solar Collectors," Journal of Thermal Engineering, 2(1), pp. 529–540.

[13] Choi, S. U. S., and Eastman, J. A., 1995, "Enhancing Thermal Conductivity of Fluids with Nanoparticles," Developments and Applications of Non-Newtonian Flows, ASME International Mechanical Engineering Congress and Exposition, San Francisco, CA, FED-Vol. 231/MD-Vol. 66, pp. 99–105.

[14] Sekhar, Y. R., Sharma, K. V., Karupparaj, R. T., and Chiranjeevi, C., 2013, "Heat Transfer Enhancement with Al_2O_3 Nanofluids and Twisted Tapes in a Pipe for Solar Thermal Applications, "Procedia Engineering, 64, pp. 1474–1484. https://doi.org/10.1016/j.proeng.2013.09.227

[15] Ajeeb, W., Thieleke da Silva, R. R. S., and Murshed, S. M. S., 2023, "Experimental Investigation of Heat Transfer Performance of Al_2O_3 Nanofluids in a Compact Plate Heat Exchanger," Applied Thermal Engineering, Vol. 218, Article No. 119321, doi: 10.1016/j.applthermaleng.2022.119321.

[16] Mande, R. K., Rama Raju, S., and Varma, K. K., 2024, "Thermophysical Properties of TiO_2/CuO Hybrid Nanofluids for Heat Transfer Applications," Materials Research Innovations, 28(2), pp. 83–93.

[17] Nakhchi, M. E., and Esfahani, J. A., 2019, "Numerical Investigation of Rectangular-Cut Twisted Tape Inserts on Performance Improvement of Heat Exchangers," International Journal of Thermal Sciences, 138, pp. 75–83.

[18] Nakhchi, M. E., and Esfahani, J. A., 2021, "Numerical Investigation of Turbulent CuO–Water Nanofluid Inside Heat Exchanger Enhanced with Double V-Cut Twisted Tapes," Journal of Thermal Analysis and Calorimetry, 145, pp. 2535–2545.

[19] Prasad, P. D., Gupta, A. V., and Deepak, K., 2015, "Investigation of Trapezoidal-Cut Twisted Tape Insert in a Double Pipe U-Tube Heat Exchanger Using Al_2O_3/Water Nanofluid," Procedia Materials Science, 10, pp. 50–63.

[20] Bergles, A. E., 1998, "Techniques to Enhance Heat Transfer," in Handbook of Heat Transfer, 3rd ed., Rohsenow, W. M., Hartnett, J. P., and Cho, Y. I., eds., McGraw-Hill, New York, Chap. 11, pp. 11.1–11.76.

[21] Chen Q, Hao JH, Zhao T. An alternative energy flow model for analysis and optimization of heat transfer systems. International Journal of Heat and Mass Transfer. 2017 May 1;108: 712-20.

[22] Souayeh, B., Bhattacharyya, S., Hdhiri, N., and Hammami, F., 2021, "Numerical Investigation on Heat Transfer Augmentation in a Triangular Solar Air Heater Tube Fitted with Angular-Cut Varied-Length Twisted Tape," The European Physical Journal Plus, 136(6), p. 643. https://doi.org/10.1140/epjp/s13360-021-01599-0

[23] Murugesan, P., Mayilsamy, K., Suresh, S., and Srinivasan, P. S., 2011, "Heat Transfer and Pressure Drop Characteristics in a Circular Tube Fitted with and Without V-Cut Twisted Tape Insert," International Communications in Heat and Mass Transfer, 38(3), pp. 329–334.

[24] Wongcharee, K., and Eiamsa-ard, S., 2012, "Heat Transfer Enhancement by Using CuO/Water Nanofluid in Corrugated Tube Equipped with Twisted Tape," International Communications in Heat and mass Transfer, Vol. 39, No. 2, pp.251–257, doi: 10.1016/j.icheatmasstransfer.2011.11.008.

[25] Lachaud, J. O., Romon, P., and Thibert, B., 2022, "Corrected Curvature Measures," Discrete & Computational Geometry, 68(2), pp. 477–524.

[26] Rahimi, M., Shabanian, S. R., and Alsairafi, A. A., 2009, "Experimental and CFD Studies on Heat Transfer and Friction Factor Characteristics of a Tube Equipped with Modified Twisted Tape Inserts," Chemical Engineering and Processing: Process Intensification, 48(3), pp. 762–770.

[27] Salman, S. D., Kadhum, A. A., Takriff, M. S., and Mohamad, A. B., 2013, "CFD Analysis of Heat Transfer and Friction Factor Characteristics in a Circular Tube Fitted with Quadrant-Cut Twisted Tape Inserts," Mathematical Problems in Engineering, 2013(1), p. 273764. https://doi.org/10.1155/2013/273764

[28] Kola, P. V., Pisipaty, S. K., Mendu, S. S., and Ghosh, R., 2021, "Optimization of Performance Parameters of a Double Pipe Heat Exchanger with Cut Twisted Tapes Using CFD and RSM," Chemical Engineering and Processing: Process Intensification, 163, p. 108362.

[29] Govindarajan, B. M., and Leishman, J. G., 2016, "Curvature Corrections to Improve the Accuracy of Free-Vortex Methods," Journal of Aircraft, 53(2), pp. 378–386.

[30] Gnielinski, V., 1976, "New Equations for Heat and Mass Transfer in Turbulent Pipe and Channel Flow," International Chemical Engineering, 16(2), pp. 359–368.

[31] Notter, R. H., and Sleicher, C. A., 1972, "A Solution to the Turbulent Graetz Problem—III: Fully Developed and Entry Region Heat Transfer Rates," Chemical Engineering Science, 27(11), pp. 2073–2093.

[32] Blasius, H., 1908, "Boundary Layers in Fluids of Small Viscosity," Zeitschrift für Mathematik und Physik, Vol. 56, No. 1, pp. 1–37.

[33] Petukhov, B. S., 1970, "Heat Transfer and Friction in Turbulent Pipe Flow with Variable Physical Properties," in Advances in Heat Transfer, Vol. 6, Elsevier, pp. 503–564.

[34] Mande, R. K., Raju, S. R., and Krishna Varma, K. P., 2025, "Augmentation of Heat Transfer in a DPHE With Bowl Cut Twisted Tape Inserts and TiO_2/CuO Hybrid Nanofluids," Chemical Engineering Communications, 212(2), pp. 219–234.

[35] Kapade, T. T., Shah, S. K., Shah, D. S., Kumar, N. S., Bhor, J. K., Panwar, V. S., and Singh, G., 2025, "Thermal Analysis of V-Cut Twisted Tape Fitted Heat Exchanger: A Numerical Approach," International Journal on Interactive Design and Manufacturing (IJIDeM), 19(3), pp. 1895–1911.

[36] Sundar, L. S., and Sharma, K. V., 2010, "Turbulent Heat Transfer and Friction Factor of Al_2O_3 Nanofluid in Circular Tube with Twisted Tape Inserts," International Journal of Heat and Mass Transfer, Vol. 53, Nos. 7–8, pp. 1409–1416, doi: 10.1016/j.ijheatmasstransfer.2009.12.016.

[37] Suresh, S., Venkitaraj, K. P., and Selvakumar, P., 2011, "Comparative Study on Thermal Performance of Helical Screw Tape Inserts in Laminar Flow Using Al_2O_3/Water and CuO/Water Nanofluids," Superlattices and Microstructures, Vol. 49, No. 6, pp. 608–622, doi: 10.1016/j.spmi.2011.03.012.

[38] Varma, K. K., Kishore, P. S., and Prasad, P. D., 2017, "Enhancement of Heat Transfer Using Fe_3O_4/Water Nanofluid with Varying Cut-Radius Twisted Tape Inserts," International Journal of Applied Engineering Research, Vol. 12, No. 18, pp. 7088–7095.

[39] Salman, S. D., Kadhum, A. A., Takriff, M. S., and Mohamad, A. B., 2014, “Heat Transfer Enhancement of Laminar Nanofluids Flow in a Circular Tube Fitted with Parabolic-Cut Twisted Tape Inserts,” The Scientific World Journal, 2014(1), p. 543231. https://doi.org/10.1155/2014/543231
[40] Eiamsa-Ard, S., Wongcharee, K., and Sripattanapipat, S., 2009, “3-D Numerical Simulation of Swirling Flow and Convective Heat Transfer in a Circular Tube Induced by Means of Loose-Fit Twisted Tapes,” International Communications in Heat and Mass Transfer, 36(9), pp. 947–955.