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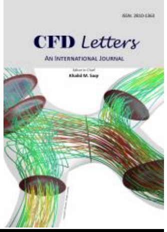

A Computational Study for Evaluating the Performance of Twisted Double Tube Heat Exchangers Fitted with Twisted Tape

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ABSTRACT

Twisted double-tube heat exchangers are promising in improving the heat transfer efficiency on the tube side, decreasing the pressure drop on the shell side, and reducing the size of the equipment. Although offering immense potential, examining heat transfer enhancement techniques inside a heat exchanger. In this study, The thermal-hydraulic characteristics of twisted double-tube heat exchangers fitted with twisted tape inserted have been numerically studied. The Naiver-stokes, energy, and turbulence equations were used to model the fluid flow and heat transfer while the turbulence was with a k- ε model. ANSYS Fluent 23.1 was used to solve the governing equations. The effect of major design elements such as mass flow rate, varied pitches of twisted double tubes and twisted tape inserts was investigated. The hot water was used in the inner tube and the cold water in the outer tube to create a counter-flow apparatus. The Length of the heat exchanger was 1 meter, and the outer and inner diameter was 0.054 and 0.018 m respectively. The thickness of the two tubes was 0.004 m. The twisted ratio of the tubes was tested for =5, 10, and 15 while the twist ratio of the tape was 4, 6, and 8. The findings demonstrated that the utilization of a double twisted tube heat exchanger with a twisted tape insert resulted in enhanced heat transfer in comparison to a plain tube heat exchanger. The numerical analysis revealed that as the twisting ratio drops, the Nusselt number, pressure drop, and overall heat transfer coefficient increase.

1. Introduction

Heat exchangers are widely used in heating and cooling systems in various industries, including petrochemical and oil companies, power plant stations, and even residential areas. The analysis of heat transfer rate, pressure drop, efficiency, and considerations for long-term durability and simple maintenance complicates the design process for heat exchangers.

Some investigators have investigated twisted tube heat exchangers. Wu *et al.,* [1] studied the impact of water's flow resistance and heat transfer properties on the twisted elliptical tube numerically. This research looked at a three-dimensional elliptical tube with a twisted shape and convection where the wall temperature was held constant. The results demonstrate that the elliptical twisted tubes produced rotating motions in the flowing fluid, which enhance the synergy between

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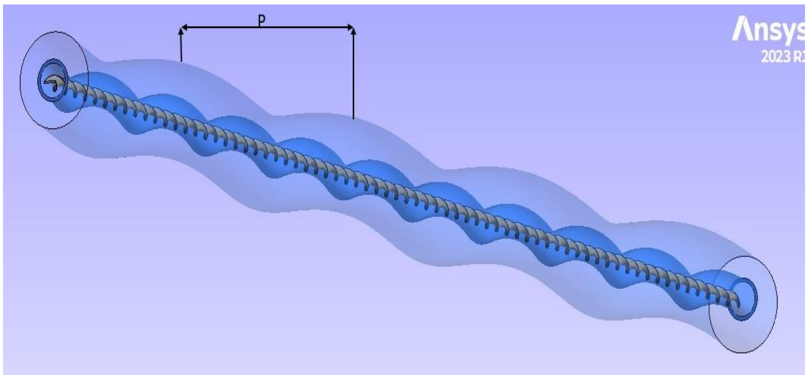
velocity vectors and the temperature gradient compared to an oval tube. The d = 128 mm twisted elliptical tube performs best in thermal-hydraulic. Mushatet and Hmood [2] investigated numerically how adding a twisted triangle tube would affect the heat exchanger's performance. Reynolds numbers from 5,000 to 25,000 and Tr=5 to 20 were considered in the analysis. They discovered an improved thermal performance when comparing the twisted pipe to the plane tube. Razzaq and Mushatet [3- 6] simulated the double-twisted tube heat exchanger operation with nanofluids. The flow was turbulent, several nanofluid concentrations were used, and a twist ratio of five was utilized throughout the experiment. The results showed a significant connection between the twist ratio, concentration, and thermal-hydraulic performance. Luo *et al.,* [7] studied numerically the fluid flow and heat transfer characteristics for two co-twisting oval pipes that twist in opposite directions and have different pitch ratios. The flow was laminar, outer and inner pipes with a twist ratio of 1.0 to 2.0. The results showed that heat transfer increased initially as the twist pitch ratio increases, then decreases. The highest differences in friction factor and Nusselt number across various twist pitch ratios were 19.0% and 71.4%, respectively. Chaurasiya *et al.,* [8] simulated numerically the corrugation effect on the inner tube's outer and inner surfaces in the twin-pipe heat exchanger. The working Reynolds number (Re) ranged from 4000 to 20000, and the tube wall was subjected to a constant temperature. Internally corrugated inner tubes (ICIT) and externally corrugated inner tubes (ECIT) were the two types of tubes that are modelled and numerically examined at three different helix angles (15°, 20°, and 25°). The results showed that externally corrugated inner tubes (ECIT) perform more effectively than internal corrugated tubes. At = 15°, the PEC was higher for ECIT. Heeraman *et al.,* [9] experimentally studied the heat transfer and friction qualities of a horizontal double-pipe heat exchanger (DPHE) using twisted tape with dimple inserts of varying dimple diameters and dimple-to-dimple half-length (H) ratios (D/H). At a diameter of 4 mm, the best performance of Nusselt number (Nu) and performance evaluation criteria (PEC) was identified. As a result showed, dimpled twisted tape additions was a great, low-cost way to optimize heat transformation in heat exchangers. Mushatet *et al.,* [10] studied experimentally and mathematically novel configurations of tapered twisted tape within a hot tube to achieve optimal thermal and hydrodynamic performance levels. The research was conducted in turbulent Reynolds number regimes ranging from 10,000 to 400,000. The best thermal performance had been improved by 137%. Fagr *et al.,* [11] conducted an experimental and numerical study to identify the effect of utilizing reduced tapered twisted tape models. The study discovered an insignificant variation in the coefficient of thermal performance when compared to a plain twisted tube. Naik *et al.,* [12] experimentally investigated the convective heat transfer and friction factor properties of water/propylene glycol-based CuO nanofluids flowing in a tube with and without twisted tape inserts. Experiments were carried out with CuO nanofluids having Reynold values ranging from 1000 to Re 100000 and volume concentrations of 0.025%, 0.1%, and 0.5%. The results indicated that comparing the twisted tape's friction factor to the base fluid's friction factor reveals an improvement of 10.08%. Salam *et al.,* [13] experimentally evaluated the heat transfer coefficient in a circular tube equipped with a rectangular-cut twisted tape insert. Water was the working fluid, and the Reynolds numbers was from 1000 to 19000. These investigations showed that Nusselt numbers in tubes with rectangular-cut twisted tape inserts increased by 2.3 to 2.9 times, while friction factors increased by 1.4 to 1.8 times compared to a smooth tube. Murali *et al.,* [14] numerically investigated the heat transfer and friction factor characteristics of a circular tube fitted with full-length twisted trapezoidal cut. The design parameters such as the Reynolds number range of 2000–12,000 and trapezoidal-cut of twist ratio y = 4.4 and (d1 = 0.0285 m, d2 = 0.0545 m L = 2 m) and cold water flows in counter flow through the annulus. The results revealed that the heat transfer and friction factor properties of a circular tube fitted with trapezoidal-cut twisted tape with a twist ratio of 4.0 compared to a plain tube with a twist ratio of 4.0, the heat transfer coefficient and friction factor rise. Karimi *et al.,* [15] investigated mathematically the nanofluid and twisted tape effect of the thermal performance. It was expected that the nanofluid composed of alumina and water and plain water would make suitable working fluids. According to the findings, utilizing twisted tape elevated the Nusselt number by as much as 22%. Furthermore, adding alumina particles to water elevated the rate of heat transfer by as much as 30% and the rate of pressure decreased by as much as 40%.Nakhchi and Esfahani [16] investigated the effect of turbulence characteristics and thermal enhancement parameters of CuO-water nanofluids through heat exchangers enhanced with double V-cut twisted tapes. The Reynolds number is 5,000 to 15,000, and the twist ratio of the twisted tapes is 5.25. The results showed that employing double V-cut twisted tapes improves the Nusselt number of the nanofluid flow inside heat exchangers by 138%. Li *et al.,* [17] analyzed numerically the effects of the tube insert's centrally hollow narrow-twisted tape's influence on heat transfer improvement. The flow is laminar of water at $400 \leq Re \leq 1200$. According to the findings, the cross-hollow twisted tape's best overall heat transfer performance improves by 3.7–12.3 and 4.3–28.1%, respectively. Hussein and Mushatet [18] investigated experimentally the heat transfer and turbulent flow structure of a heated pipe equipped with converge diverge conical turbulators and coupled twisted and screw tapes. The working fluid was air for a Reynolds number between 15000 and 65000. The results revealed that transfer is improved by approximately 47.7%, 43.18%, and 39.7% for twist ratios of 3.0, 5.0, and 7.0 when tapes are used in conjunction with conical turbulators. Mushatet *et al.,* [19] investigated numerically the impact of the length of central cut twisted tape on the features of the thermodynamic and hydrodynamic fields. A tube in which air flows in a turbulence-like pattern contains this tape; the tube was tested under identical heat flux. 10000–40000 Re was the airflow Reynolds number range. The results showed that with an increase in the Reynolds number, the Nusselt number rises, and the friction and thermal performance factors decrease. The maximum Nusselt number was 171.172 at Re = 40000, which grew as the length ratio increased. Mushatet and Youssif [20, 21] examined numerically nanofluid Al_2O_3 moving through a horizontal tube with a twin twisted tape insert. The Reynolds number ranges from 5000 to 35000, and Tr= 2, 4, and 6 with concentration volume (ϕ) of 0.5% and 4 %. The result showed that at the highest volume concentration (=4%) and twisted ratio (Tr=2), Al_2O_3/water twin-twisted tape could improve heat transfer by approximately 214% compared to plain tube. Heeraman *et al.,* [22] investigated experimentally the characteristics of heat transfer rate and pressure drop in a double pipe heat exchanger made of twisted tape with dimples. The twisted ratio for all cases was 5.5, range of Reynolds number (Re) 6000–14,000, and (D/H) of 1.5, 3 and 4.5. As the Reynolds number improves, the impact of twisted tape with dimples on the friction factor becomes more effective when compared to the Nusselt number as Reynolds number improves. Singh and Sarkar [23, 24] experimentally examined the influence of V-cuts twisted tape inserts on the double-tube heat exchanger's heat transfer and pressure drop properties. It takes a volume concentration of 0.1% to produce the Al_2O_3+TiO_2 hybrid nanofluid with varying twist ratios, and the Reynolds number will range from 9000 to 14000. Results indicated that the Nusselt number and friction factor rise when the twisting ratio, width ratio, and temperature decrease.

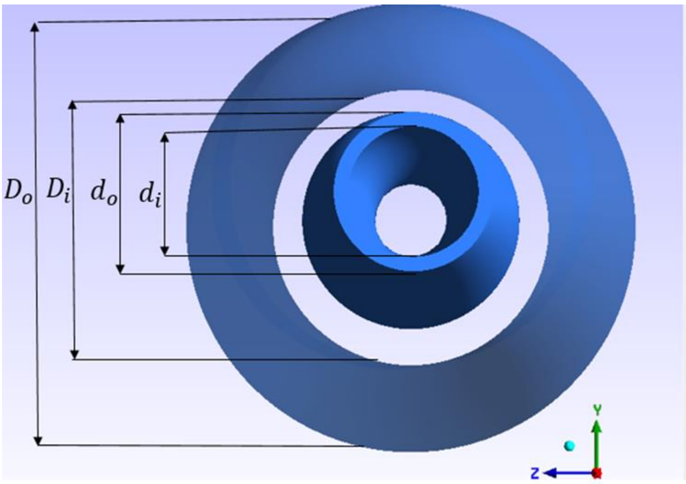
In the literature explored above, different investigation works were offered on diverse types of swirl flow devices, but only a few research works were reported on twisted tubes with inserted twisted tape. However, in the current work, two cases of heat exchanger configuration are examined; the first is the twisted tubes without twisted tape, and the second, the twisted tubes are integrated with twisted tape. Different values of twist ratio for twisted tubes and twisted tape are to be tested. The effect of studied parameters on effectiveness, the outlet temperatures, Nusselt number, and overall heat transfer coefficient are to be displayed.

2. Physical Model

Figure 1 shows a diagrammatic representation of a double circular twisted tube heat exchanger with twisted tape (DCTTHE-TT). This figure illustrates the inner tube has a diameter (di=0.018m) while the diameter of the outer tube (Di=0.05m); both tubes are 1m long and the thickness is (t=0.004m). Three values of tube twist ratios (Tr= 5, 10, 15) and twisted tape ratios (TR=4, 6, 8) are examined. Different values of hot mass flow rate (m_h) is selected as (0.024, 0.049, .073, 0.098, and 0.122) kg/s and constant value of mass flow rate cold fluid (m_c=0.19)kg/s.



(a) Double circular twisted tube with a twisted tape



(b) Cross section for double circular twisted tube

Fig. 1. Schematic representation of the physical problem

3. Mathematical Model and Numerical Analysis

Full Navier-Stokes describe the equations governing fluid flow and energy as follows [25]:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \tag{1}$$

$$\left(\frac{\partial u^2}{\partial x}+\frac{\partial uv}{\partial y}+\frac{\partial uw}{\partial z}\right) = -\frac{\partial p}{\partial x}+\frac{\partial}{\partial x}\left(2\mu_{eff}\frac{\partial u}{\partial x}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial u}{\partial y}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial u}{\partial z}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial v}{\partial x}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial w}{\partial x}\right) \tag{2}$$

$$\left(\frac{\partial uv}{\partial x}+\frac{\partial v^2}{\partial y}+\frac{\partial vw}{\partial z}\right) = -\frac{\partial p}{\partial y}+\frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial v}{\partial x}\right)+\frac{\partial}{\partial y}\left(2\mu_{eff}\frac{\partial v}{\partial y}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial v}{\partial z}\right)+\frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial u}{\partial y}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial w}{\partial y}\right) \tag{3}$$

$$\frac{\partial \mathrm{uT}}{\partial \mathrm{x}}+\frac{\partial \mathrm{vT}}{\partial \mathrm{y}}+\frac{\partial \mathrm{wT}}{\partial \mathrm{z}}=\frac{\partial}{\partial \mathrm{x}}\left(\Gamma_{\mathrm{eff}}\frac{\partial \mathrm{T}}{\partial \mathrm{x}}\right)+\frac{\partial}{\partial \mathrm{y}}\left(\Gamma_{\mathrm{eff}}\frac{\partial \mathrm{T}}{\partial \mathrm{y}}\right)+\frac{\partial}{\partial \mathrm{z}}\left(\Gamma_{\mathrm{eff}}\frac{\partial}{\partial \mathrm{z}}\right) \tag{4}$$

$$\Gamma_{\mathrm{eff}} = \Gamma + \Gamma_{\mathrm{t}} \tag{5}$$

$$\mu_{\mathrm{eff}} = \mu + \mu_{\mathrm{t}} \tag{6}$$

3.1 Turbulence Model

The impact of turbulence on the flow is accounted for using the standard k-ε. Standard k-ε is widely used in heat transfer because it provides reasonable accuracy, robustness, and economy for a broad class of turbulent flows. The transport equations for classical k-ε as follows. [26]:

Turbulent kinetic energy (k) equation:

$$\frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}\left(\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right)+G_k+G_b-\rho\varepsilon \tag{7}$$

Turbulent energy dissipation (ε) equation:

$$\frac{\partial}{\partial x_i}(\rho\varepsilon u_i) = \frac{\partial}{\partial x_j}\left(\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right)+C_{1\varepsilon}\frac{\varepsilon}{k}(G_k+C_{3\varepsilon}G_b)-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \tag{8}$$

G_k and G_b are measures of the kinetic energy produced by turbulent motion due to gradients in mean velocity. The model constants C_1, C_2, and C_3 for the diffusion of k and ε.

The turbulence viscosity:

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{9}$$

Where $C_\mu = 0.09, C_{1\varepsilon} = 1.44, C_{2\varepsilon} = 1.92, \sigma_k = 1, \sigma_\varepsilon = 1.3$

3.2 Boundary Condition

Boundary conditions are applied to the analyzed model as follows:

i. At the inlet:
m_c is set as 0.19 kg/s
m_h is selected as (0.024, 0.049, .073, 0.098, and 0.122) kg/s
The value of hot fluid inlet temperature (T_{hi}) is set at 353 k, and the value of cold fluid inlet temperature (T_{ci}) is set at 298 k.
ii. At walls: no-slip condition, the outer wall of the outer pipe is insulated.
iii. Zero relative gauge pressure is assumed at the outlets.

3.3 Grid Independency

Figure 2 presents a general view of the computing grid in this study, a structured mesh is produced using ANSYS meshing software in consideration of higher quality outputs, faster convergence, and fewer cells are only a few of the benefits of structured mesh over unstructured mesh.

The average overall heat transfer coefficient for different mesh sizes is calculated to ensure grid independence. Table 1 shows the total heat coefficient for various mesh sizes in a double circular tube heat exchanger (DCTTHE) and a Reynolds number of 5000. The difference between the two successive examples is shown in the last column. This is calculated by taking the total heat coefficient for the smaller mesh size, subtracting the larger mesh size, and then dividing by the smaller number. The number of mesh elements was increased to 1,208,000, resulting in a 1.08 % inaccuracy, as seen in the table. This suggests that further refining the mesh has little effect on the total coefficient. As a result, the mesh size is proportional to the number of mesh elements used in the setup above.

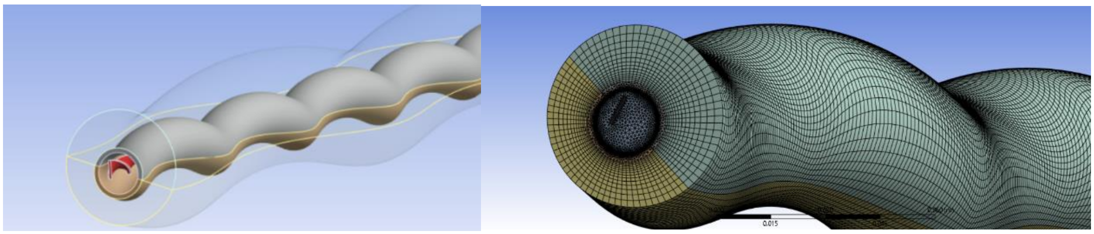


Fig. 2. Structure of the computational domain produced with meshing

Table 1
Overall heat transfer coefficient for different mesh size

Cases	Mesh size	Overall heat coefficient	Percentage error (%)
1	164,500	534.48	-
2	198,492	546.81	2.3
3	300,000	558.43	2.12
4	468,384	567.19	1.6
5	761,362	574.12	1.2
6	1,208,000	580.36	1.08
7	1,782,044	584.44	0.7

3.4 Code Validation

Figure 3 Illustrates the hot and cold water exit temperatures for the present computational model and experimental data by Kotian *et al.,*[26] in terms of hot flow rate. To validate the numerical model, the identical experimental conditions outlined by Kotian *et al.,* [26] are incorporated, including the inner and outer tube lengths of 1200 mm, inner tube diameters of 26 mm and 68 mm, and inner and outer tube thicknesses of 4 mm. It can be seen that the results from the current model and the experimental data given by Kotian *et al.,* [26] accord well with one another. The largest discrepancy between the outcomes is 2.3%. As a result, this technique has been established as robust and accurate.

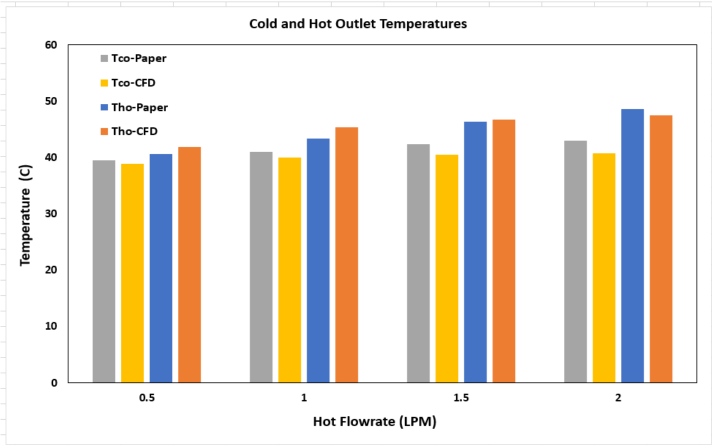


Fig. 3. Validation between the present results and published results of Kotian *et al.,* [26]

3.5 Data Reduction

$$LMTD = \frac{\Delta T1 - \Delta T2}{In(\frac{\Delta T1}{\Delta T2})} \tag{10}$$

$$\Delta T1 = (Thi - Tco) \tag{11}$$

$$\Delta T2 = (Tho - Tci\ ) \tag{12}$$

"The heat exchanger's effectiveness is"

$$\epsilon = \frac{qact}{qmax} \tag{13}$$

$$qact = C_c(T_{ci} - T_{co})$$

$$qmax = C_{min}(T_{hi} - T_{ho})$$

$$C_h = m_h cp_h$$

$$C_c = m_c cp_c$$

Where $Cmin$ is the minimum value of Cc and Ch;

$$R_{ei} = \frac{\rho D_{hi} V_i}{\mu} \tag{14}$$

$$V_i = \frac{m_h}{\rho A_{c,i}} \tag{15}$$

$$R_{eo} = \frac{\rho D_{ho} V_o}{\mu} \tag{17}$$

$$V_o = \frac{m_c}{\rho A_{c,o}} \tag{18}$$

Where $D_{hi} = \frac{4A_{c,i}}{p_{w,i}}$ and $D_{ho} = \frac{4A_{c,o}}{p_{w,o}}$ (19)

$$D_{hi} = d_i \, , D_{ho} = D_i - d_o \tag{20}$$

The overall heat transfer coefficient is determined as follows:

$$U = \frac{Q_{avg}}{A_{s,o} LMTD} \tag{21}$$

Where $A_{s,o} = \pi d_o L, Q_{avg} = \frac{Q_h + Q_c}{2}$

4. Results and Discussion

The results are obtained for the double twisted tube integrated with a twisted tape. The effect of changing the twist ratio and mass flow rate on the heat transfer and flow field is deliberated by presenting the outlet temperature, Nusslte number, overall performance of the heat exchanger and effectiveness of the heat exchanger.

4.1 The Effect of the Double Twisted Tube on Heat Exchanger Performance

Figure 4 depicts the change of the hot water outlet temperature with mass flow rater for double twisted circular tubes heat exchanger (DCTTHE) with varied twist ratios. The figure shows that the (T_{ho}) rises as the hot water mass flow rate increases and lowers as the Tr drops. As shown in the graph, as the twisted ratio drops, it indicates an increased heat exchange rate with the cold fluid. The flow manner is redirected by the (DCTTHE) design, and a smooth secondary flow is produced close to the tube wall. The flow encounters rotating motions close to the wall as the twisted ratio decreases; this enhances the secondary flow, the mixing effect, and the thermal boundary layer disturbance; all this can be seen more in Tr = 5

Figure 5 shows the variation in the cold outlet temperature versus the mass flow rate of hot water for the double circular twisted tubes with different twisted ratios. The cold water outlet temperature increases significantly when mass flow rates increase for all cases. Also, the double circular twisted tube (DCTTHE) has an average improvement in cold water outlet temperature for (Tr=5,10 and 15) respectively by 0.015%,0.012%, and 0.011%over (DCTHE) and this figure demonstrates that due to rising secondary flow with a greater number of twists, (T_{co}) is larger at the height of Tr = 5.

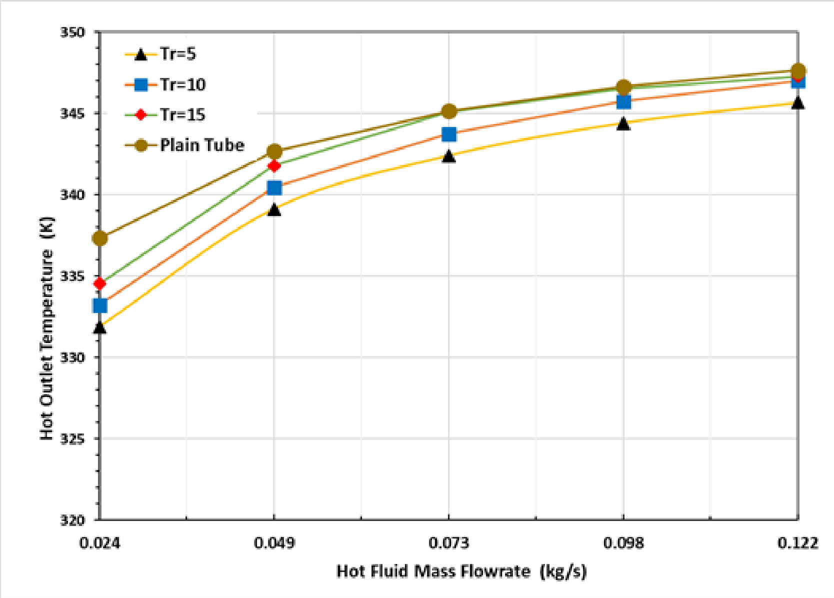


Fig. 4. Variation of the hot fluid outlet temperature with hot fluid mass flow rate for DCTTHE with varying twist ratio

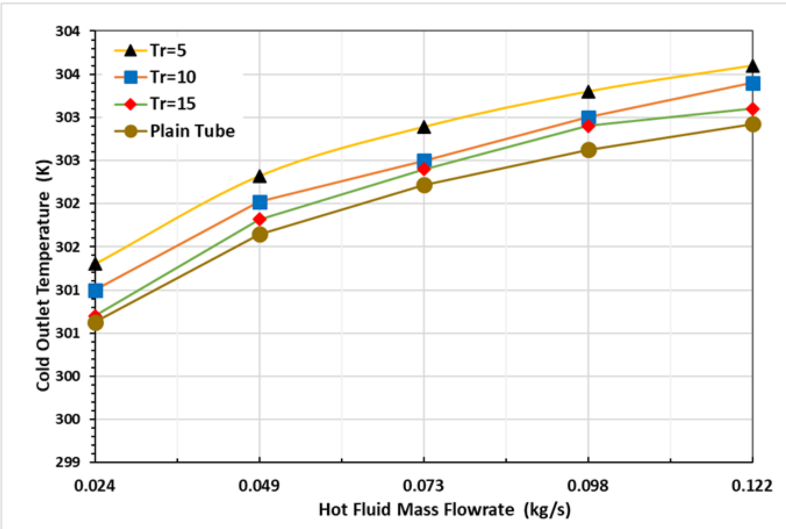


Fig. 5. Variation of the cold outlet temperature with the mass flow rate of hot water for DCTTHE with varying twist ratios

Figure 6 describes the relationship between the coefficient of overall heat transfer and the mass flow rate of hot water with different twist ratios, this figure shows that as the hot water mass flow rate increases, so does the overall heat transfer coefficient because a large amount of heat is transported. As the double tubes are twisted, heat transfer is improved because a secondary flow is generated drawing the fluid closer to the tube wall. The overall heat transfer coefficient improves when the mass flow rate is 0.122 and the twisted ratio is 5 by 25% compared with the plain tube.

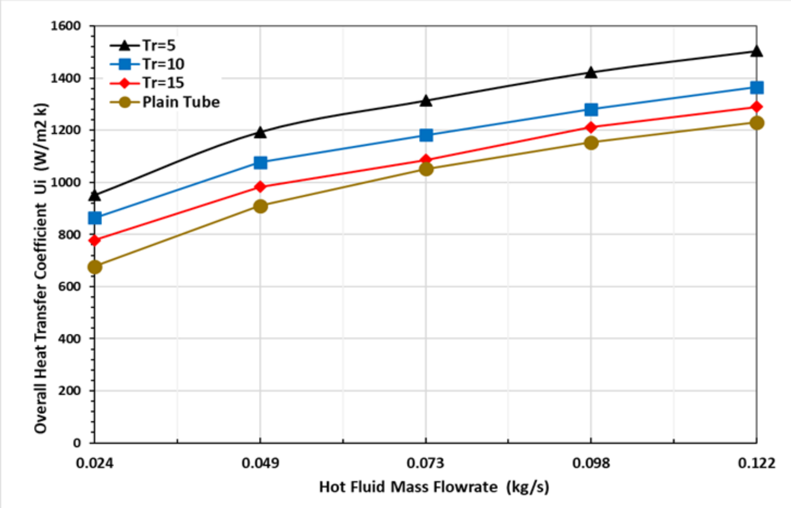


Fig. 6. Variation of the coefficient of overall heat transfer versus mass flow rate of hot water for different twist ratios

Figure7 illustrates the variation in the effectiveness of the heat exchanger with a mass flow rate of hot water for DCTTHE with different twist ratios. This graph illustrates that heat exchanger effectiveness improved with the increase in the mass flow rate of hot water. The effectiveness of the heat exchanger improves by 43%, 35%, and 32% when Tr= 5, 10, and 15 compared with the plain tube. Because the momentum of the fluid and the turbulence of the flow both increase as the double tubes are twisted, more heat is transferred from the hot fluid to the cold fluid. When determining the effectiveness of the heat exchanger, it is sufficient to divide the actual heat by the extreme heat, as the latter mostly determines the former. This diagram also shows that increased secondary flow makes the heat exchanger more efficient above the Tr = 5.

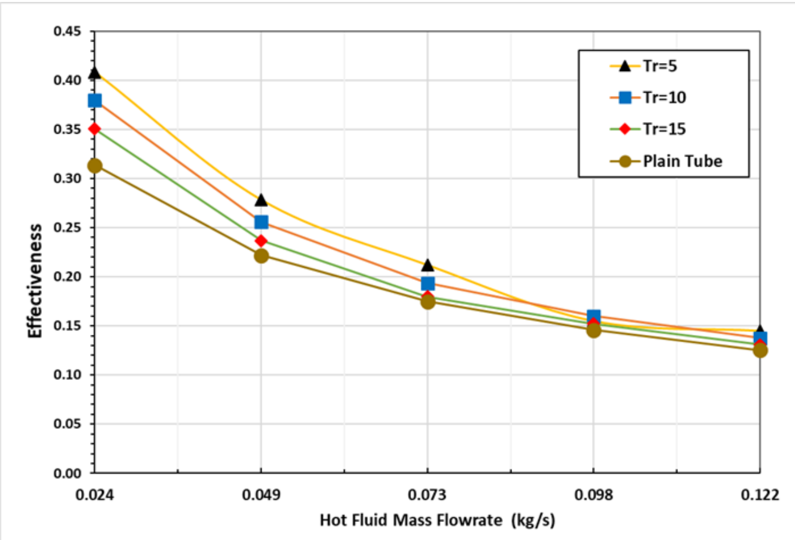


Fig. 7. Variation of the effectiveness versus mass flow rate of hot water for different twist ratios

4.2 The Influence of Twisted Tape on Heat Exchanger Performance

This section examines the effect of combined twisted tubes heat exchanger and insert twisted tape on the performance and the heat. Combining double twisted circular tubes heat exchangers with twisted tape (DCTTHE-TT) creates multiple swirling flows along the wall and the core region. This research unit performs diverse numerical simulations using various twist ratios for twisted tape (TR=4, 6, and 8).

Figure 8 illustrates the hot fluid outlet temperature varies with the hot water mass flow rate for (DCTTHE-TT) varied twist ratios of twisted tape. Based on this graph, it can be seen that the (T_{ho}) of water increases as the mass flow rate increases, but decreases as the number of twisted ratios decreases. Furthermore, at the height of TR = 4, the (T_{ho}) of water is reduced because the co-implementation for (DCTTHE –TT) results in a more uniform temperature distribution along the tube wall. The influence of twisting generates a rise because of the secondary flow formed in the tube and along the wall regions.

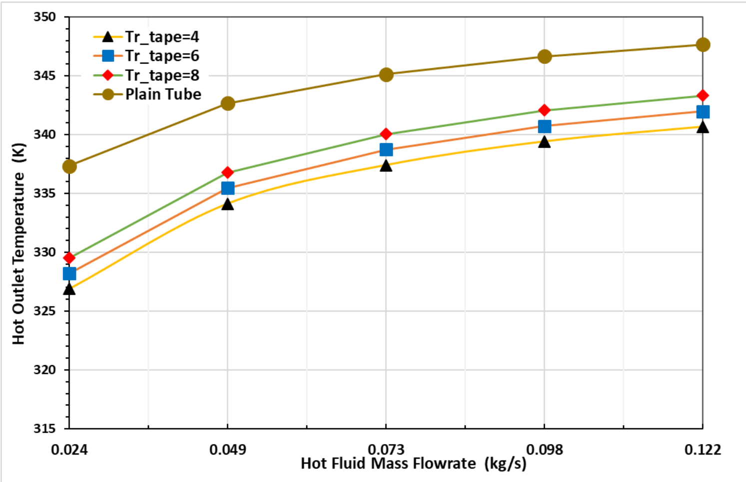


Fig. 8. Variation of the hot fluid outlet temperature with mass flow rate of hot water for DCTTHE-TT with different twist ratios

Figure 9 shows the relationship between the cold water outlet temperature and the hot mass flow for (DCTTHE-TT) with different twisted ratios. It can be seen that the (T_{co}) of water increases as the mass flow rate increases. The cold water outlet temperature of the (DCTTHE-TT) with a twisted ratio of 4 was higher than the other twisted ratios of 6, 8, and plain double tube. This can be explained by the fact that it results from higher swirl intensity and flow mixing effects.

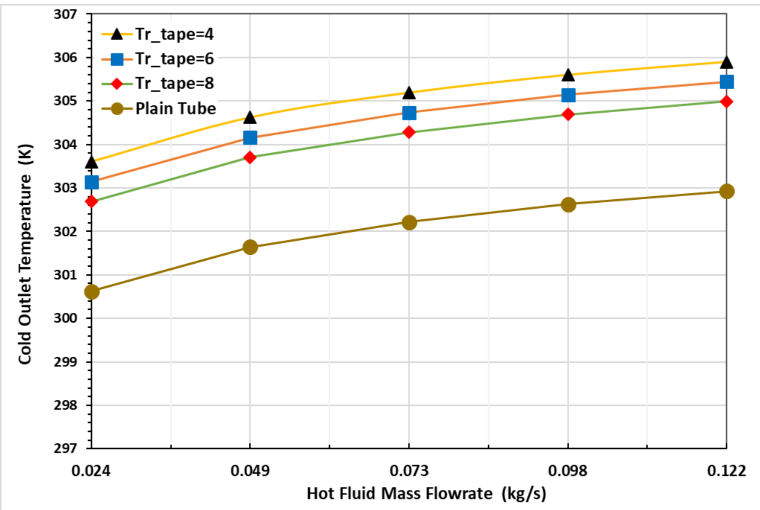


Fig. 9. Variation of the cold water outlet temperature varies with the hot water mass flow rate with varying twist ratios

Figure 10 illustrates the change in the overall heat transfer coefficient for (DCTTHE -TT) with twisted ratios (TR=4, 6, and 8). This figure shows that the overall heat transfer coefficient rises with the hot mass flow for all twisting ratios and declines with the twisting ratio due to increased turbulence paired with excess heat rise transmission. It was discovered that the overall heat transfer coefficient at TR=4 is higher than at (TR=6 and 8). The lowest (TR=4) value generates a powerful whirl intensity, leading to more effective thermal boundary layer disturbance along the flow direction.

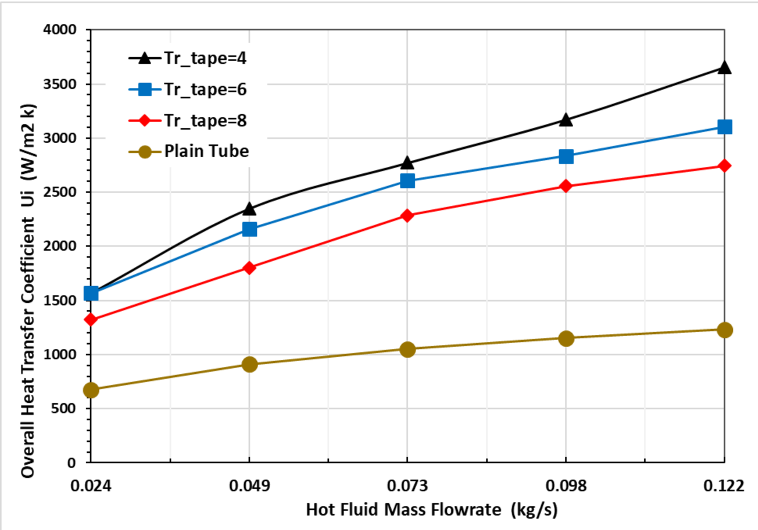


Fig. 10. Variation of the coefficient of overall heat transfer versus with mass flow rate of hot water with different twist ratios

Figure 11 displays the variation of the Nusselt number with the mass flow rate of hot water for (DCTTHE-TT) with different twisted ratios. The figure shows that the Nu number is increased when the mass flow rate increases, which leads to more turbulent flow and more momentum in the fluid. The hot water flow velocity increases the heat transfer coefficient due to the increase in the hot water flow velocity. Also, this figure indicates a higher Nu number at (TR=4). The smaller twist ratio can explain this producing stronger swirl intensity, leading to more functional obstruction of the boundary layer along the flow path.

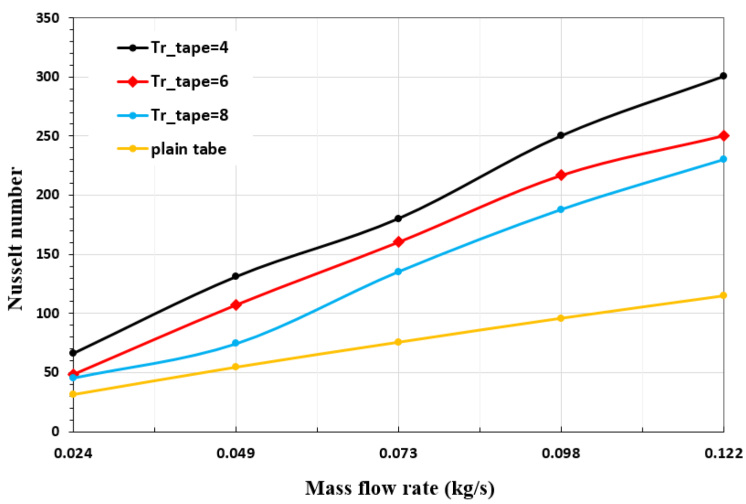


Fig. 11. Variation of the Nusselt number with mass flow rate of hot fluid for different twisted ratios

Figure 12 explains the variation of the total pressure drops (ΔPT) of the (DCTTHE-TT) with different twisted ratios. This figure displays that the (ΔPT) is increased when the mass flow rate of hot fluid increases due to the increase in the flow momentum. In the double twisted tube with twisted tape higher (Δp) because of the increase in the flow velocity in the hot water side due to the reduction in the flow area. When the twisted ratio of twisted tape (TR=4), vortices occur because of the high twist, increasing the strength of the secondary flow and leading to the pressure decrease.

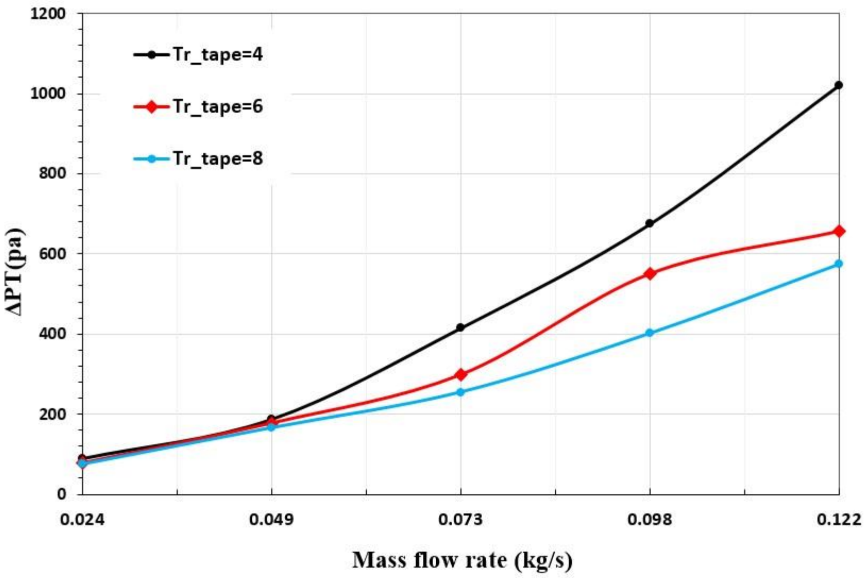


Fig. 12. Variation of the total pressure drop with a mass flow rate of hot fluid for DCTTHE-TT

Figure 13 represents the temperature distribution in a cross-section for (DCTTHE) and (DCTTHE – TT) at $m_h$of 0.024 kg/s and m_c is the constant. As seen in this figure, in section (a) Fluid mixing is enhanced, the boundary layer is disturbed, and the thickness is significantly reduced in double twisted circular tubes with a twisted ratio of 5. In section (b) to improve fluid mixing in the annulus, the inner twisted circular tube with twisted tape insert causes the fluid to flow rotatable and procedures the secondary flow. Compared to the straight inner tube model, the twisted inner tube model has a higher temperature in the annulus. Heat transmission is improved as the hot flow enters the tube at a lower temperature, thanks to the annular flow caused by the twist. When the twisted ratio is 4, the thermal boundary layer in the inner tube wall is ruptured due to the increased velocity and secondary flow. This can be explained by the smaller twist ratio producing stronger swirl intensity, leading to more functional obstruction of the boundary layer along the flow path.

Figure 14 represents the velocity distribution in a cross-section for DCTTHE and DCTTHE-TT at m_h=0.024 kg/s and m_c =0.19 kg/s. Because of wall shear stress, the velocity is the lowest closest to the tube wall and greatest in the center of each cross-section. There is an increase in velocity at the tube's center in the DCTTHE-TT velocity curves. The creation of the boundary layer is disrupted, and its thickness is reduced due to flow-mixing effects when the twisted tape is added to the inner tube.

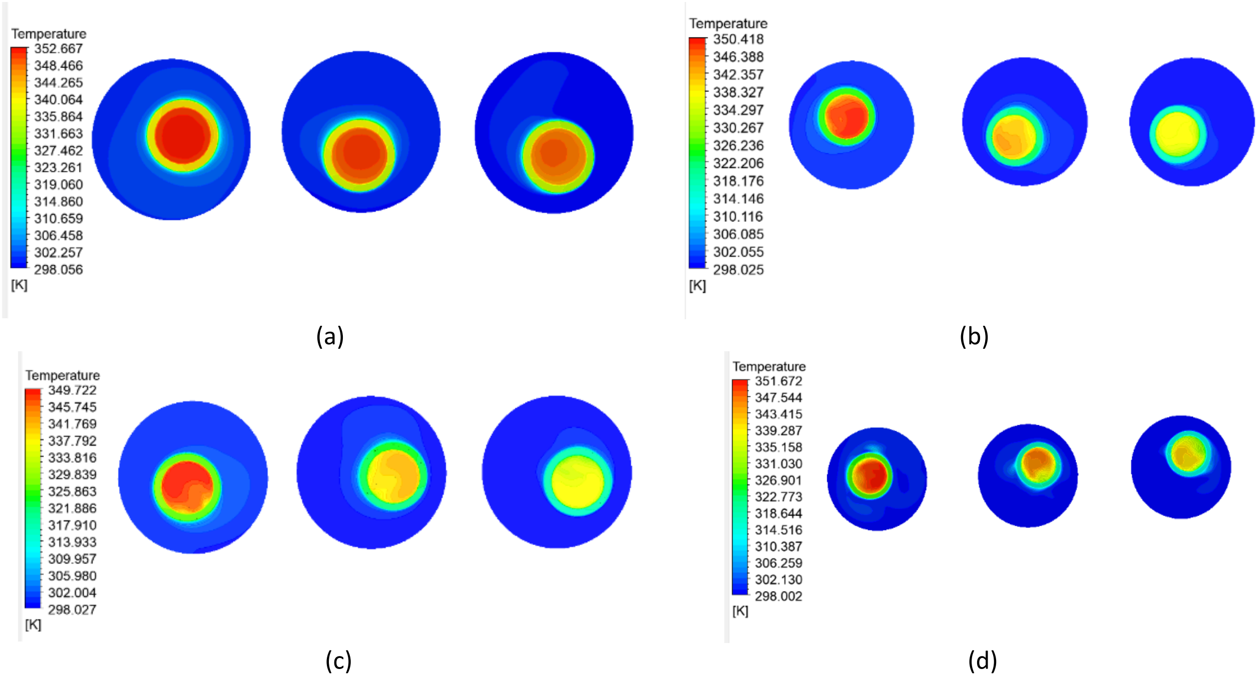


Fig. 13. Cross-section temperature contour at x=0.2, x=0.5, x=0.8m (a) double twisted tube (DCTTHE) with Tr =5(b) compound double twisted tube and twisted tape (DCTTHE-TT) with TR=4 (c) DCTTHE-TT with TR =6 (d) DCTTHE-TT with TR=8

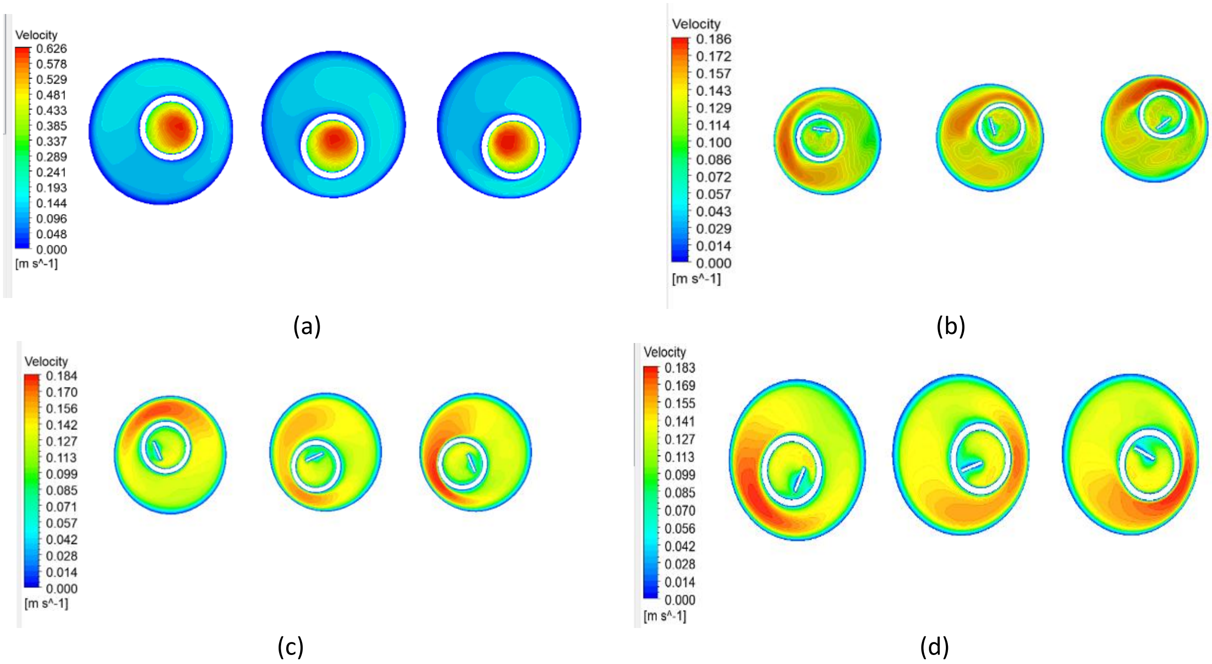


Fig. 14. Cross-section velocity contour at x=0.2, x=0.5, x=0.8m (a) double twisted tube DCTTHE with Tr =5(b) compound twisted tube and twisted tape DCTTHE-TT with TR=4 (c) DCTTHE-TT with TR=6 (d) DCTTHE-TT with TR=8 for

5. Conclusions

This work presents the results of numerical research that demonstrates the impact of a double twisted circular tube with twisted tape in a double-tube heat exchanger for different twisted ratios (4, 6, 8). The most important findings are summed up as follows:

i. A heat exchanger with a double twisted circular tube can achieve better thermal performance than one with a plain tube.
ii. Redirecting the flow from the core region to the heat was facilitated by the insertion of a twisted tape inside the tube (DCTTHE-TT), which twisted the inner tube and increased fluid mixing in the tubing annulus.
iii. According to the simulations, the optimal twist ratio for thermal performance is 4.
iv. When the hot water flow rate is 0.024 and Tr =5, the effectiveness of the double twisted tube heat exchanger (DCTTHE) is about 40%.
v. The tubes' twists are the key factor impacting heat transmission via twisted tubes in low-mass flow.

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