Enhancement of Heat Transfer in an Inclined Tube Using Vortex Generator

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ABSTRACT: The performance of heat exchanger can be ameliorated via several methods. This research presents a numerical and experimental investigation to study the heat transfer enhancement in an inclined tube having a twisted tape with v-cut. This study includes constructing a test section, which consists of a tube made of copper having (1 m) long, with internal and external diameter of (23, 25 mm) respectively. The coolant flows through the inner pipe under laminar flow (1056 ≥ Re ≥ 2002) regime and uniform heat flux (2420-12261W/m2). The pressure drop, wall temperature, flow rate, temperature of distilled water through a tube with a circular cross-sectional area and twisted tape inserts were measured. Fluent, the CFD (Computational Fluid Dynamics) extension of the software Package of Ansys 15.0, was used to solve the governing three-dimensional equations with the finite volume technique and to evaluate the effect of Reynolds number, twisted tape types and twist ratios on the enhancement of heat transfer, loss of friction, average Nusselt number, velocity profile, the performance of thermal characteristic factor within a tube having a circular cross section and also with a laminar flow. The simulations were performed with different values of Reynolds numbers using water as working fluid. Experimental data were compared with the numerical results and this showed a high similarity, with a maximum error of 10% that was obtained while using the twisted tape having a v-cut with a twist ratio (TR) equal to 4.

KEYWORDS: Augmentation of heat transfer; Laminar flow; Twisted tape.

INTRODUCTION

The heat exchangers are widely utilized in the engineering and industrial applications. The approach to design them is more difficult and requires more analysis of efficiency, rate of heat transfer and friction factor estimations apart from issues, such as the economic aspect of the equipment and long-term performance. Therefore, any enhancement tool or methods used in heat exchanger must be optimized between higher pumping cost and advantages of the coefficient of heat transfer.

In fact, energy transport is so far enhanced when the flow is properly blended. That is the main concept in the evolution of augmentation technical approaches that produce swirl flows. The nature of twisted tape helps to promote the secondary flow generation which leads to enhance the rate of heat transfer significantly. Enhancement in the convective transfer of heat resulted from using twisted-tape inserts is accompanied by increasing friction factor.

Bharadwaj et al., 2009 [1] calculated the pressure drop and heat transfer rate of the water flow within a ‘75-start spirally grooved tube’ having a twisted tape with three ratios of twist; y = 3.4, 7.95 and 10.15 under laminar and turbulent flow. The thermo - hydraulic characteristics tend to increase remarkably in the laminar and turbulent flows independently of the existence of tapes on the SGT (Spirally Grooved Tube). The BST (Bare Spiral Tube) and the ST (Spiral Tube) performance, as well as the drop in the rate of heat transfer with anti-clockwise twisted tape (y = 10.15) at a transient region were shown.

Eiamsa-ard, et al., 2009 [2] performed experimentally an evaluation to find the average Nusselt no. values, thermal performance factor and friction factor within a tube having a short-length twisted tape using boundary conditions of constant heat flux. The measured data in the experiments were calculated at Reynolds numbers in
the turbulent region utilizing the air as working fluid. The single twist ratio (4.0) was used for the twisted tape with a full-length, while tapes with a short-length located at the test section inlet were utilized with many LRs (Length Ratios of Tape) of (0.29, 0.43, 0.57 and 1.0), which represent a tape with a full-length. The results that showed an inferior value of rate of heat transfer and pressure drop were SLI (Short Length Inserts) with an LR of 0.29, 0.43 and 0.57, whilst the FLI (Full Length Inserts) demonstrated the values of rate of heat transfer and drop of pressure about (14%, 9.5%, and 6.7%) and (21%, 15.3% and 10.5%), respectively for each value of LR evaluated. Murugesan, et al., 2009 [3] presented experimental works to calculate the heat transfer augmentation value (Nusselt no.) and friction factor properties (pressure drop) of a tube with a round cross-section using a twisted tape through its whole length with a trapezoidal-cut for several ranges of Reynolds Number (starting from 2000 to 12000).

It was shown that the twisted tape with the trapezoidal-cut generated an increment in the coefficient of heat transfer as well as in pressure drop. From the otherside, the Nusselt Number was logical for this arrangement, whilst the performance factor was greater than 1.

Bodius et al., 2010 [4] achieved an experimental study to find the coefficient of convective heat rate of the water in a tube with a circular shape under turbulent flow regime. The tube is circular and uses a twisted tape with TR = 5.3 made of stainless steel. Nichrome wire covered with fiber glass was wrapped around the test section in order to get a fixed heat flux state. Temperature of the tube external surface in test section was measured at five different spots. T-type thermocouples were used for measuring the temperature with a thermometer put in a blending chapter at the outlet section. The study was conducted over a range of 9500-20000 of Reynolds numbers. Heat flux was changed from (9) to (18) kW/m2 for a plain tube and (15) to (31) kW/m2 for a tube having a twist tape insert. It was noted that over the same Reynolds number, the twist tape insert caused an increasing in Nusselt number by (2.9) to (4) times in comparison with the tube having a flat shape. The Dittus and Boelter correlation was compared with the results, giving an error from -13% to 18%.

Ray and Date, 2001 [5] conducted a numerical study to find the fluid flow and improve the transfer of heat in a duct with a square shape and having a twisted tape in a laminar flow and uniform heat flux circumstances. The thermal performance of square and circular and ducts suited with same TR of twisted tapes was studied too. Results showed a clear increasing in the rate of convective heat transfer with the duct of a square shape, especially at smaller ratios of twist and larger Prandtl number.

Masoud et al., 2009 [6] executed an experimental and numerical evaluation to find the coefficient of heat transfer, friction factor, coefficient of heat transfer as well as the thermal performance index of a tube having four types of twisted tape inserts. A 3-D simulation was conducted in the numerical part by using FLUENT6.2 code. The three kinds of twisted tape inserts (Notched, Jagged and Peforated) were used in the experimental section to compare with a distinctive twisted tape to evaluate their efficiency. Width of the all used twisted tape inserts was 15 mm, with 5 cm pitch length and TR = 2.94. The experimental data was validated and performed using the achieved modeling. The heat transfer coefficient and thermal performance ratio showed a higher magnitude for the jagged type of insert compared to other types of twisted tapes. The comparison with the distinctive twisted tape showed that 31% and 21% were the maximum increases in the Nusselt Number and the Thermal Index, respectively.

Yangjun, et al., 2011 [7] presented a numerical study (CFD modeling) to optimize the efficiency of a tube (transporting air as working fluid in a turbulent states) having a circular cross section and a twisted tape with a short length that was uniformly spaced. Three parameters were studied, including twist ratio (TR), space ratio (SR) and rotated angle (RA). They observed that the increasing in (RA) and decreasing in (TR) lead to a higher coefficient of heat transfer with better thermal performance index utilizing various twist ratios that range from (2.5) to (8.0) with exception for a big Reynolds no. and a big angle of rotation.

Sami et al., 2013 [8] presented a numerical study to simulate a tube having a circular shape suited with an elliptic-cut and typical twisted tapes under a laminar flow. The authors also investigated the effect of different twisted ratios (2.93, 3.91, 4.89) and various depth of cuts (0.4, 0.8, 1.4 cm) on the Nusselt no. in addition to friction factor. The study was carried out using ANSYS Fluent V. 6.3.26 and varying the operational conditions, so a Reynolds Number from 200 to 2100 was obtained. The friction factor and coefficient of heat transfer with an elliptic-cut twist tape in a plain-tube were greater than the results obtained for the classical
twisted tape.

Gulia and Parinam, 2014 [9] implemented a numerical study for calculating the rates of heat transfer in a tube having a circular shape and suited with a twisted tape for y/w = 4, under a turbulent flow. The study performed revealed that a Length Ratio of 0.29, 0.43, 0.57 and 1 for the twisted tapes resulted in an increase of 15%, 18.8%, 22.6% and 31% of the rate of heat transfer compared to a smooth tube, respectively. From the other side, a Length Ratio of 0.29, 0.43, 0.57 and 1 for the twisted tapes resulted in a thermal performance index of 1.2, 1.3, 1.32 and 1.37, respectively.

Osama, 2014 [10] studied experimentally and numerically the most auspicious parameters for the design of a tube in form of a helical coil. A heat exchanger with helical tubes having 15 mm diameter, the effects related to the insertion of a coil wire of twisted tape into the helical tube heat exchanger were examined. The Dean number is in the range of 700 to 2000. Firstly, a coiled wire with different insertions of 15, 20 and 30 mm was used. Secondly, experiments were conducted at a constant pitch of 15 mm with different sections (circular or square), and thirdly, at a constant insert pitch of 15 mm with different square wire thicknesses (a=1 and 2 mm). Nanoparticles of Al2O3 and TiO2 (d=80 nm, 30 nm), respectively were dispersed in distilled water. The volume concentration was in the range of (0.08, 0.1, 0.2 and 0.3%) so as to simulate the heat transfer flow in a helical coil tube with nanofluids, Perfect outputs were obtained (φ = 0.3, P = 15 mm, Al2O3, De = 1889 and t = 2 mm square cross-section). The enhancement of heat transfer utilizing a pair of mechanisms together in comparison with the use of a pure fluid in a plain helical tube surpasses above (120%) (33.2% Al2O3 nanofluid + 87.26% coiled wire enhancement). A commercial program ANSYS Fluent 14.5 was used. An empirical correlation with a data error of (±20 %) for Nusselt number was obtained.

In the current study, the influence of employing different twisted tapes will be investigated numerically. V-cut twisted tape and alternated axis twisted tape with variant twist ratios (y = 4 and 6) are to be employed for the enhancement of heat transfer within heat exchangers having inclined tubes. A numerical analysis will be carried out using ANSYS FLUENT Package 15.0 for two cases (inclined plain tube and inclined tube with a twisted tape).

Test Rig Description

The experimental test rig includes the following parts. Figure (1) presents the photographic view of the experimental setup, which contains:

1. The test section consisting of copper pipe, heater, electrical and heat insulators.
2. The cold working fluid cycle consisting of fluid tank, pump, flow meters, control valves, test section pipe and piping system.
3. Variac transformers.

Most of those parts are designed and manufactured. Care was taken into account in order to prevent the leakage of water between the connected sections and re-fixing.
Figure 1. Photograph view of experimental setup.

Experimental Methodology

1. Filling the water storage tank with (3 liters) of distill water.
2. Running the test rig (heated section, centrifugal pump).
3. Adjusting the water amount required to the flow meter scale using the main pump and by pass valves.
4. Adjusting the electric current across the heater coils using Variac voltage transformer to get the desired constant heat flux.
5. Pumping the distilled water at 20°C to the test section where it is exposed to a constant heat flux from the heater, then draining out the heated water and allowing the cold water supply for the next test.
6. Repeating the procedure steps mentioned above for the two twist ratios with changing in the discharge and the electric power supplied.

Numerical Simulation

The simulation by CFD is analyzing the regimes including the fluid flow, phenomena of heat transfer and so on. At first, a mathematical model is constructed from a set of differential equations which governs the behavior of the fluid flow. The mentioned differential equations are then converted into algebraic equations and then solved using a proper solver to find the flow variables of the flow domain. The development and application of Computational Fluid Dynamics have taken a considerable care and propagate rapidly, that is the reason why the CFD has turned into a suitable tool for designing and analyzing the majority of the engineering processes [11].

For analyzing the flow field in a plain tube heat exchanger with a twisted tape under constant heat flux, inner forced convection, and conduction through the tube material, a solution of continuity, Navier-Stokes equations as well as the energy equation is required. Because of the complexity of twisted tapes configurations, it is impossible to find an analytical solution for the set of differential equations that describe the behavior of the fluid flow. That is the reason why the numerical simulations are required for analyzing the complex domains of fluid flow. Therefore, Ansys-Workbench Fluent code 15.0 [12], which uses the Finite Volume Numerical Technique, has been used to solve those equations.

System geometry

The system geometry in the present study consists of a plain tube, where the working fluid flows, together with two types of inserts (v-cut and Alternate axis twisted tape inserts).

Tube geometry

The test section tube has a diameter and length (Din =0.023 m, Z=1 m), respectively. This tube is made from copper with thickness (1 mm). Its wall is heated under a fixed heat flux ranging from (2420) to (12261) W/m², and with constant inlet fluid temperature (20°C). The working fluid (DI-water) enters at a variable volume flow rate with a constant inlet temperature of 20°C. The convection heat transfer between the heated inner tube surface and the inlet working fluid is in a horizontal x-y-z plane.

Twisted tape geometry

It is strip of a metal with a finite length twisted with various twist ratios and pitches (the pitch is a distance that the strip needs for rotating 180°).

The twist ratio (y) is:

\[ y = \frac{P}{W} \]  

(3-1)

The tape width and the tube’s inner diameter are taken approximately the same (W=0.023 m), and the tape length and thickness are (L=1 m, δ=1 mm), respectively. Two TR (4 and 6, respectively) were employed in both twisted tape types.
The dimensions of the v-cut are taken from [13] who found the optimum cut depth ratio and width ratio as 
(DR=0.43, WR=0.34), respectively for all twist ratios, as shown in figure (2) where:

\[
\begin{align*}
\text{DR} &= \frac{e_d}{W} \\
\text{WR} &= \frac{e_w}{W}
\end{align*}
\]  
(3-2)

(3-3)

The v-cut is located in each pitch in the opposite direction to the previous one.

**Figure 2.** Dimensions of the v-cut twisted tape.

**Governing Equations for Fluids in Motion**

To describe the fluid flow, differential equations are used, such as the Continuity and the Navier-Stokes equations. For flows including heat transfer phenomena, the energy conservation equation should be enabled.

**Continuity equation**

\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0
\]  
(3-4)

**Momentum equation**

**X-Momentum equation:**

\[
\rho \left( u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} \right) = -\frac{\partial p}{\partial x} + \mu \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right)
\]  
(3-5)

**Y-Momentum equation:**

\[
\rho \left( u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} \right) = -\frac{\partial p}{\partial y} + \mu \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right)
\]  
(3-6)

**Z-Momentum equation:**

\[
\rho \left( u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} \right) = -\frac{\partial p}{\partial z} + \mu \left( \frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right)
\]  
(3-7)
Energy equation

\[\left(\rho C_p\right)\left(u \frac{\partial T_{af}}{\partial x} + v \frac{\partial T_{af}}{\partial y} + w \frac{\partial T_{af}}{\partial z}\right) = K \left(\frac{\partial^2 T_{af}}{\partial x^2} + \frac{\partial^2 T_{af}}{\partial y^2} + \frac{\partial^2 T_{af}}{\partial z^2}\right)\]  \hspace{1cm} (3-8)

Assumptions of Heat Exchanger with a Circular Plain Tube

The assumptions used in the present model are:

1. A three-dimensional laminar steady flow.
2. Incompressible and Newtonian fluid.
3. Constant inlet temperature and neglecting viscous dissipation.
4. The kinetic and potential energies change are negligible.
5. Usually unimportant axial conduction along tube.
6. Disabled transfer of heat transfer by radiation.
7. Buoyancy force influence is not considered.
8. No heat generation within the heat exchanger.
9. The current and voltage delivered from heater tape across the tube wall are assumed to be constant to achieve constant heat flux condition at the outer surface tube.

Strategy for the numerical solution

To create the system’s geometry simulations, the (CFD) software Fluent 15.0 was depending upon the finite volume numerical approach to solve a set of differential equations that governs the fluid flow. For the discretization of the terms of energy, convection, and laminar (dissipation and kinetic energy), the second-order upwind scheme was applied, which ensures a satisfactory accuracy, convergence and stability for tri, tetrahedral and polyhedral meshes for the flow domain [14]. The Semi-Implicit Method for Pressure-Linked Equations (SIMPLE) was the employed algorithm to obtain the solution of the velocity and pressure fields coupling [15]. It is essentially a gauss and correct procedure for calculation of pressure. For all cases study, the converged solutions were regarded if the residuals that determined by the iterative procedure for the whole governing equations did not change with the iteration progress and the computational error may be ignored, and then the iteration manually stopped. For most of the cases, the iterations are around (1600-2000). The equations of velocity, energy and continuity, as well as their respective scaled residual were monitored for each iterative calculation. For the convergence, the following values were considered:

Residual for continuity = $10^{-3}$

Residual for velocities = $10^{-3}$

Residual for energy = $10^{-6}$

RESULTS

Behavior of the thermal and hydrodynamic characteristic of the flow caused by the twisted tape in inclined Tubes.

Twisted tape thermal effect in inclined tube ($\Theta=20^\circ$)

The experimental results of the inclined tube suited with the v-cut twisted tape were compared with a plain tube for a laminar flow regime.

Otherness of Reynolds and Nusselt number at the tube entrance (at test section) using a v-cut twisted tape having TR 4 and 6, as well as the plain tube are shown in figure (3). With reference to this figure, it can note that the average Nusselt no. raises with the Reynolds no. By contrast, results showed that the reduction in TR will cause an increment in average Nusselt no. This happens because an inferior value of the twist ratio develops a more chaotic fluid flow, which breaks the boundary layer in the adjacencies of the tube’s inner surface. This chaotic flow generates an energy variation in the fluid which causes a higher rate of heat transfer.
across the mentioned layers.

In this evaluation, the average Nusselt no. using a v-cut twisted tape with TR of 4 and 6 was, respectively, 1.6 and 1.4 times higher in compared to a smooth tube. The Nusselt no. has been found as follows:

\[
\overline{Nu}_i = \frac{\bar{h}d_i}{k_h}
\]

(4-1)

**Figure 3.** Nusselt no. variation with Reynolds no. and twist ratio of the v-cut twisted tape for (TR 4 and 6) for an inclined tube at \( q^* = 2421 \text{ W/m}^2 \)

Twisted tape hydrodynamic effect in inclined tube (\( \Theta = 20^\circ \))

The decrease of the friction factor with respect to the increase in the Reynolds no. is shown in figure (4) for TR = 4, and it reveals that this decrease is higher in comparison with a TR = 6 due to the chaotic behavior of the fluid generated by the twisted tape with a v-cut at the inferiors values of Twist Ratio. Friction factors for this twisted tape with a v-cut and TR = 4 and 6, respectively, revealed the results of 1.3 and 1.1 times greater than that for plain tube. Also, it was noticed from this figure that friction factor in inclined tube is more than in horizontal one. Generally, raising the inclinations of tube raises the flow acceleration close the wall and, as a result, greater skin friction takes place. As a rule, it is recommended that one stays away from using the inclined tube in practical applications.

The calculation of friction based on this equation:

\[
f = \frac{2Dg}{LV^2} \left[ \frac{p_1 - p_2}{\rho g} + (z_1 - z_2) \right]
\]

(4-2)
Figure 4. Friction factor variation with Reynolds no. and twist ratio of the v-cut twisted tape for (TR 4 and 6) for an inclined tube at $q^* = 2421$ W/m$^2$.

Thermal performance factor evaluation in inclined tube

The thermal enhancement index as a function of different twist ratios for a v-cut twisted taped was compared due to the fact that it links the Nusselt no. and friction factor. Hence, comparisons of the performance results of heat exchanger are useful when using thermal index than comparing them using friction factor and Nusselt no. for several twist ratios presented in the present project.

Figure (5) shows the comparison of the v-cut twisted tape used in the current investigation for the twist ratios (4 and 6), respectively.
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Figure 5. Thermal performance variation with Reynolds no. and twist ratio of V-cut twisted tape.

Numerical results analysis in inclined tube
Velocity vectors in inclined tube (θ=20°)

Figures (6) and (7) show the velocity vector of the inner tube for without and using twisted tape insert, respectively. The secondary flow field presents a wider range of velocity as compared to the smooth plain tube. The principal source of the augmentation of heat transfer is the incorporated motion in the inner tubular section with twisted tapes. A remarkable modification has been applied in the secondary flow field for smooth plain tube because of the incorporated chaotic motion. It can be noticed that the eddies’ number was increased in comparison with the case of plain tube. Mixing of the temperature field raises due to this process and this improves the rate of the overall heat exchange.

Figure 6. Velocity vector in (m/s) at different locations of inclined plain tube along the test section with (Re =2002, q*=2421 W/m2).

Figure 7. Velocity vector in (m/s) at different locations of v-cut twisted tape along the test section with (Re =2002, TR= 4 and q*=2421 W/m2).

Temperature and Velocity Contours in Inclined Tube (θ=20°)
The experimental results of inclined plain tube and v-cut twisted tape compared with the Computational Fluid Dynamics (CFD) simulation results were found 1.5°C as maximum. The temperature contours across the length were estimated as shown in figures (8) and (9) for inclined plain tube and v-cut insert, respectively. These figures show the comparison in temperature contours of plain tube and twisted tape insert in three dimensions at
a constant inlet velocity.

For more explanation, figures (10) and (11) depict the temperature contours at different locations along the inclined tube for plain tube and v-cut insert, respectively. From these figures, the temperature distribution gradient along the tube sections can be seen.

The temperature field of twisted tape was shown to be highly homogeneous compared to plain tube, also this causes that the fluid chaotic motion increases the rate of heat transfer significantly.

Figures (12) and (13) highlight the velocity contours at locations (Z = 0, 100 cm) along the test section for the plain tube and the v-cut twisted tape, respectively. From these figures, one can notice that the twisted tape presence produced higher velocity, then higher turbulence and better heat transfer due to the high mixing and good heat distribution between flow core and boundary.
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Figure 10. Temperature contours in (°C) at the various locations of test section for the inclined plain tube at Re = 2002 and $q^*$ = 2421 W/m².

Figure 11. Temperature contours in (°C) at the various locations of test section for the inclined tube having v-cut twisted tape at (Re = 2002, $q^*$ = 2421 W/m²).
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Figure 12. Velocity contours for the inclined plain tube at various axial distances at (Re =2002) and heat flux (q'' = 2421 W/m²).

Figure 13. Velocity contours for the inclined tube with a v-cut twisted tape at various axial distances at (Re =2002) and heat flux (q'' = 2421 W/m²)

Figure 14. Pressure contours for the inclined plain tube along the test section at (Re =2002) and heat flux (q'' = 2421 W/m²) at (θ = 20°)

Pressure contours in inclined tube (θ=20°)

In the domain of an inclined plain tube and a tube having a v-cut insert, it is noted from the contours in figures (14) and (15) that the pressure drop starts from the inlet tube section to the downstream of flow along the tube to the outlet section due to the friction losses between the fluid viscous boundary layer and the tube inner wall. From these figures, it can be noted that the pressure drop for distilled water flowing through the plain tube having a v-cut twisted tape is larger than the drop of pressure for plain tube.
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Figure 15. Pressure contours for inclined plain with v-cut twisted tape along the test section at (Re =2002) and heat flux (q"= 2421 W/m²) at (ϴ = 20°)

CONCLUSIONS

The investigation of convective heat transfer and the characteristics of pressure drop for a v-cut twisted tape induced in the single inclined tube heat exchanger with two twist ratios (4 and 6) have been evaluated experimentally and numerically. In concordance to previous analysis related to this evaluation, it is shown that a variation on the typical twisted tape (as small cuts on the tapes) will improve both thermal enhancement and heat transfer. The conclusions drawn from the current work are presented in the following:

1. The TR=4 in the tapes resulted in a higher thermal improvement in comparison with the having TR = 6 due to the fact that the increases of the flow’s residential time with a chaotic motion generates a flow with more time to exchange the thermal energy between the core wall and the wall.
2. Generally, the v-cut twisted tape results in an increase in the thermal performance in comparison with the plain tube since the cut on the tape peripheral region increases the turbulence of fluid.
3. The inclined tube introduced more friction factor than the horizontal tube for all cases with a lower Nusselt number than horizontal tube.
4. The temperature of the flow increases and becomes more homogenous with increasing the flow rate and decreases with the twist ratio.

REFERENCES

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