Study of heat transfer enhancement through the non-circular ducts using three dimensional numerical investigations and their comparisons

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Abstract. The present study numerically surveyed fluid characteristics such as friction factor and heat transfer through the different geometry of twisted ducts. A three-dimensional analysis is performed for the Reynolds number of 100-50000 for the air that acts as a working fluid and a uniform wall temperature boundary case is studied numerically throughout the simulation of ducts. Numerical results of this present study are validated against the experimental results. Heat transfer improvements are made over the hexagonally twisted channels, straight square duct and clockwise-counter-clockwise (CW-CCW) twisted square channels. Twist ratios of 7 · 5, 11 · 5 and 16 · 5 are considered for the current simulations. Laminar flow to turbulent flow regimes over the ducts are briefly presented and discussed. Results show enhancement in heat transfer through the CW-CCW , hexagonal and straight square ducts(SSD) for the laminar as well as for turbulent regimes are figured out and perfectly described. It is seen that in laminar regime, twisted CW-CCW square duct performed well than twisted square and hexagonal duct. A comparison of friction coefficients are also established for both the channels. Less friction factor is observed for twisted CW-CCW duct compared to other duct.

Keywords: Twisted CW-CCW square duct, Twist ratio, Pitch, Straight square duct, Hexagonal duct, Periodic flow

1. Introduction
Due to the industrial importance of the compact heat exchanger, many of the research publications are available in this area. So the twisted channels employed in this study specially applicable for the designing of industrial heat exchanger device. Heat transfer enhancement techniques mainly find applications in the design of more compact heat exchangers. Various industries, in particular refrigeration, automotive and chemical industries, are using this technology. The use of twisted ducts, fins, twisted strips, and ribs increases the heat transfer rates. These additives create a vortex that mixing the fluid in the direction of cross-stream which increases the heat transfer rate. Twisted channels used the passive heat transfer augmentation technique which does not require an external power supply.

Flow through an elliptic tube were analyzed up to a large twist ratio (TR=21, 53, 106) by using the finite difference method (FDM) in the laminar flow regime. They determined the axial and circumferential velocities, resistance coefficients and the streamline patterns[1]. Friction factor...
and Nusselt numbers of twisted square ducts were experimentally and numerically investigated and found that Nusselt number obtained through the twisted ducts are higher than the straight square duct for Reynolds number of 1000-100000 and Prandtl number of 0.7-2. It was also identified laminar to transition point along the stream direction of the duct by the researcher [2, 3].

The effect of V-shaped twisted strip insertion on heat transfer, friction factor and thermal performance factor properties in a circular tube for three twisted ratios (y = 2, 4.4 and 6) and three different groups of depth and width ratios. The obtained results show that the average number of Nusselt and average friction factor in the tube with V-type twisted tape (VTT) increases with lower torsional ratios (y) and width ratios (WR) and increased depth ratios (DR)[4, 5].

Flow and heat transfer characteristics in a twisted elliptic tube (TET) were defined and the effects of the aspect ratio and the number of rotations in the TET were analyzed comparatively using three-dimensional (3-D) numerical simulation[6]. Experimental investigation of the fluid properties such as friction factor and heat transfer of square duct equipped with twisted tapes with different twisting conditions were analysed properly. It is found that at a minimum twist ratio, higher heat transfer enhancement occurred [7].

The influence of spiral screw tape coupled with ribbed tubulator on thermodynamic properties inside a circular tube for Reynolds number 6000 to 20000 and rib pitch ratio 1 to 3 were reported experimentally. It is revealed that by reducing the rib pitch ratio the heat transfer and friction factor increase. They also presented the correlations for the nozzle number and friction factor [8].

The characteristics of heat transfer and friction factor of the twisted oval tube were numerically studied by a realized k-epsilon method with varying the aspect ratios and twisted pitch length. They revealed that the Nusselt number and friction factor increases with the increasing aspect ratio as well as pitch length [9]. Laminar flow of water through an air cooled vertical channel is investigated experimentally with Reynolds number 110-1500 and twist ratio of 1.62-5.49. Correlations were also developed for heat transfer coefficients by [10]. They found that the effectiveness of heat transfer increases with the growth of twisted-tape pitch at slow liquid flows after going beyond the critical speed.

Heat transfer enhancement using nanofluids in equilateral triangular duct ($Al_2O_3$/water) and also the effect of nanoparticle shape and size and its volume fraction (1%,2%,3%) for the 100-500 Reynolds number range was thoroughly studied by [11]. They find that the platelet’s nanoparticle form shows the biggest improvement in heat transfer. The average Nusselt number rises up to 25% at the maximum Reynolds number when using platelet nanoparticle shape compared to pure water. Heat transfer is also increases by increasing the fraction of the nanoparticle volume.

He et al.[12] simulated the characteristics of fluid flow and heat transfer of CuO-water nanofluids with volume concentration in the range of 1-4% using single and double phase models at Reynolds number range of 300-36,000 inside the tube. They found that two-phase model gives the closest results in reality compared to single-phase model and at higher concentration of nanofluid i.e at 4%, performance efficiency is more.

The heat transfer and friction factor characteristics of a twisted hexagonal model were compared with the twisted square duct. It was reported that the twisted hexagonal model enhanced more heat transfer than the twisted square duct for the certain Reynolds number due to an increase in the number of sides of the hexagonal duct [13]. An experiment on convection inside a novel alternating elliptical axis tube and heat transfer and flow resistance correlations are made for Reynolds number of 500-50000. It is depicted that heat transfer enhanced greatly with less flow resistance [14].

The characteristics of fluid flow and heat transfer of helically finned tube with the assumptions of steady, three dimensional and incompressible fluid flows of continuity, momentum, and energy.
They also classified the effect of geometry on the calculation of local pressure coefficient, vorticity of fluid and the Nusselt number of the helically finned tube stated by [15]. Characteristics of friction factor and heat transfer of fluid flow through the twisted square duct (TSD) where nano-fluid used as a working fluid. It was found that nano-fluids used in TSD are greater friction factor and Nusselt number compared to the straight square duct [16].

Kurnia et al. [17] were numerically investigated the flow behavior of air and the related heat transfer in a helical tube with a twisted tape insert subjected to constant wall temperature. Based on conservation equations of mass, momentum, and energy, a three-dimensional computational model is created and validated against the proven empirical correlations for the Reynolds number range of 100-2000 with different twist ratios. Lower twist ratios have been found to increase heat more than higher twist ratios.

A numerical study of the characteristics of heat transfer and fluid flow through swirling channels of different cross sections by varying the Reynolds numbers from 50 to 2000 and comparing these results with the results of a flat pipe and obtained that a twisted pipe enhances heat transfer more than a flat pipe [18].

Augmentation of heat transfer in heat exchanger by using twisted tape in inclined and horizontal tube is investigated numerically by [19] using twist ratios 4 and 6 for Reynolds number of 1056 to 2002 in FLUENT software. It showed that twisted tape enhanced the coefficient of heat transfer more than the smooth tube. They also presented the correlations of Nusselt number and friction factor.

It is clarified from literature survey that the most of the researchers have used twist ratios in between 2 to 20 for the design purpose of the industrial compact heat exchanger [2, 3]. Based on this survey, this present study uses the twist ratios value of 7.5, 11.5 and 16.5. There are several investigations have done to improve the coefficient of heat transfer and characteristics of friction factor by the non-circular ducts. But the results of the heat transfer coefficients and properties of friction factors by the construction of twisted CW-CCW square ducts are not yet made by any researchers and the comparison with twisted hexagonal ducts and straight square ducts is not available either. So the objectives of this research work is to (a) determine the properties of the friction factor and the heat transfer coefficient through the twisted hexagonal and CW-CCW square duct with twist ratios 16·5 and 11·5 and (b) compare the results of friction factor and Nusselt number of twisted CW-CCW square duct with twisted hexagonal and straight square duct.

2. Phraseology

General terms related to twisted pipe are expressed in Figures (2) and (1):

Pitch (P): For tube length, the distance between two points in a row, since the direction of the cross section is directly aligned. The cross section rotates by 360° along the distance of one pitch.

Twist ratio (Tr): Relationship between the pitch (p) and the hydraulic diameter \(d_h = \frac{4A}{p}\) of the duct is called the twist ratio.

3. Geometrical configurations

The inside diameter used for the twisted square section is 18·5 mm and the sides used for the hexagonal twisted duct are 18·5 mm. The thickness (δ) used for both the models are 2 mm. Tube rotates spirally along the channel axis and the axis extends through the center of the section. Twist ratios used are 16·5, 11·5, and 7·5. The outer cylinder inner shell is a straight circle \((d_i)\). The outer diameter \((d_o)\) for both models are 31 mm and 37 mm depicted in Figure (3). The tube rotates spirally along the channel axis and the axis extends through the center of the section. The twist ratios used are 7·5, 11·5 and 16·5.
4. Analytical calculations
In this present study, the concentric tubes were used in heat exchangers and cooled with reflux to provide bigger temperature gradient in the tube and in the housing surface. The flow is incompressible, steady, completely hydrostatic and thermally developed from time to time. Flow is deemed to be periodic when flow characteristics are repeated along the length (L) of the channel. The following are controlled equations for such flow are as

\[ \nabla \cdot \nu = 0 \]  \hspace{1cm} (1)
Energy: \( \rho \nu \cdot \nabla \nu = -\nabla p + \nabla^2 \nu \) \hspace{1cm} (2)

Momentum: \( \rho \nu \cdot \nabla T = \nabla^2 T \) \hspace{1cm} (3)

\( \rho \nu \cdot \nabla T \) is the convective acceleration term that cannot be readily solved analytically due to its non-linear nature for the helically twisted tube. Thus, an approximate solution can be found by using the finite discretization method.

4.1. Boundary conditions

The twisted wall was maintained at a constant temperature at all times and the outer wall of the cylinder was considered thermal insulation. A no-slip boundary conditions were applied at the walls of the duct. A constant success rate \( \dot{m}=0.35 \text{ kg/s} \) is used inside the outer cylinder. But with a constant flow rate at the entrance to each repeated model, the periodic boundary condition for the upper channel was considered. In general, the degree of dependence is \( 10^{-6}, 10^{-3}, \text{ and } 10^{-5}, \) is considered for energy, continuity and velocity and the parameters that used for the simulation of ducts are represented in Table(3).

5. Grid independency test

The unstructured mesh was generally employed for both twisted hexagonal duct as well as for twisted CW-CCW square duct for the simulation due to their high complexity of the structures. Grid independence test was carried out for both the cases. Nusselt numbers are calculated at different unstructured grids at a constant Reynolds number and twist ratio of 600 and 16·5 for both structures. It is found that a grid that consists of 265745 elements of face size 1·25*10\(^{-3}\) mm shows better results for the CW-CCW square duct which is seen from Table (1). Similarly, the same procedures were utilized for twisted hexagonal duct as shown in Table (2) and found that at grid elements of 86453 with a face size of 5·6 * 10\(^{-3}\) mm are considered for the ducts shown in Figures (4) and (5).

| Element | Nusselt number | Face size*10\(^{-3}\)(mm) |
|---------|----------------|--------------------------|
| 140569  | 10.47          | 1.55                     |
| 162432  | 10.510         | 1.48                     |
| 175428  | 10.545         | 1.41                     |
| 205797  | 10.539         | 1.35                     |
| 236327  | 10.532         | 1.30                     |
| 265745  | 10.521         | 1.25                     |
| 290273  | 10.519         | 1.22                     |

6. Methodology

Ansys’ proficient software combined three-dimensional simulations for the inconsistent, stable stage flow across the model. As the pipe geometry is repeated over the length of the channel, it is stated that periodic flow is used. The numerical simulation may be restricted to a single repetition unit of total duct for periodic flows. Therefore, length of the model (L) of P/2 is sufficient for twisted hexagonal duct and CW-CCW duct. According to numerical results, the
Table 2. Constants used for hexagon duct.

| Element | Nusselt number | Face size*10^{-3}(mm) |
|---------|---------------|----------------------|
| 85475   | 6.71          | 6.05                 |
| 83229   | 6.87          | 5.7                  |
| 84843   | 6.93          | 5.5                  |
| 86453   | 6.86          | 5.6                  |
| 93842   | 6.85          | 5.4                  |

Figure 4. Mesh for Hexagonal duct (Tr-16.5)

Figure 5. Mesh for CW-CCW duct (Tr-16.5)

range of laminar regime is considered within the Reynolds range of 100-3000. From the software processing module, we get the amount needed to calculate. The quantities used in the calculation:

\[ q' = \text{heat flux at wall} \]
\[ T_w = \text{mean wall temperature} \]
\[ T_w = \text{shear stress at wall} \]
\[ T_b = \text{mean bulk temperature} \]

The thermal strength of hot liquid (water) and cold liquid (air) is calculated taking into account the equations below (4) and (5)

\[ Q_h = \dot{m}_h C_{ph}(T_{hi} - T_{ho}) \]  \hspace{1cm} (4)

\[ Q_c = \dot{m}_c C_{pc}(T_{ci} - T_{ho}) \]  \hspace{1cm} (5)
Numerical models are isolated to prevent heat loss in the surrounding area. The average heat output ($\bar{Q}$) can be calculated by Eq. (6)

$$\bar{Q} = \frac{(Q_h + Q_c)}{2} = UA_p(\Delta T_{LMTD})$$ (6)

With known values, it is possible to calculate the heat transfer area ($A_p$) of the heat exchanger, $\Delta T_{LMTD}$ (average temperature difference) and the total heat transfer coefficient ($U$). To calculate the total heat transfer coefficient ($u$) in the equation, the coefficient of heat transfer for sides of the flow, the hot and the cold sides expressed in Eq.(7)

$$\frac{1}{(U_p)} = \frac{1}{(hA_p)c} + \frac{1}{(hA_p)h} + R_w$$ (7)

Because the thickness of the THD (twisted hexagonal channel) is 2 mm. Therefore, the wall resistance of the channel = Thickness of the wall / thermal conductivity of steel = 0·002/50·2 = (3·98)*($10^{-5}$) $m^2$k/w. Due to the small wall size and the high temperature flow of the square channel, the strength of the wall can be ignored. The coefficient of heat for the thermal transfer of the length of the twisted channel is calculated according to the formula:

$$h = q'/(T_w - T_b)$$ (8)

Nusselt values can be calculated by Eq.(9)

$$Nu = (h * d_h)/k$$ (9)

In various places along the entire length of the twisted channel, the stress on the shear wall ($t_w$), heat flux ($q'$), and volumetric temperature ($T_b$) are calculated using respective software. The average friction coefficient obtained due to the shear stress of the wall as

$$\bar{f} = 2 * (\tau_w / \rho V_m^2)$$ (10)

| Models          | Boundary conditions          | Discretization                    |
|-----------------|------------------------------|-----------------------------------|
| K/ε model - RNG | Condition - Mass flow specified | Turbulent Kinetic energy - Second order upwind |
| Viscous model - k - ε | Periodic type - Translational | Pressure/Velocity coupling - Second order upwind |
| Space - 3D      | Outlet - Periodic            | Energy - Second order upwind      |
| Formulation - Implicit | Inlet - Periodic         | Momentum - Second order upwind   |
| Solver - Segregated | Wall - Constant temperature | Pressure - Second order upwind    |

7. Code Validation
The same geometric parameters such as inlet and outlet diameters, twist ratios, operating conditions, and the same boundary conditions of [2] are used to determine the standard for comparison for a twisted hexagonal duct, twisted clockwise-counter-clockwise square channel. Experimental verification of the heat transfer and friction coefficient characteristics of a twisted hexagonal duct and a clockwise-counterclockwise twisted square duct using air as a working fluid and numerical values obtained by [2] for the twist ratios of 16.5 shown in figures (6) and (7). Figure (6) shows that Nusselt number obtained in present study is validated against the Nusselt number of [2]. It can be observed that the results found from experimentally and
numerically in [2] are closely agreed with the results of present study; average deviation which is observed for the Nusselt number is 1.33% for the twisted hexagonal duct and 1.52% for the twisted CW-CCW square duct compared to [2]. Figure (7) depicts that average deviation for friction factor is found to be 1.13% for twisted hexagonal duct and 1.42% for twisted CW-CCW square duct validated against the experimental data of [2].

8. Results and discussions

After checking the numerical methodology, modeling is performed for twisting factors 7·5, 11·5 and 16·5 for Re 100-50000. Nusselt values plotted against the Reynolds number shown in Figures (8),(9),(10),(11),(12),(13),(14),(15) for a twisting hex duct (THD) and a twisting square CW-CCW duct and the results are compared with the corresponding values of SSD(straight square duct) use air as the functioning fluid. The result shows that Nu values were higher in the twisted square duct CW-CCW using air as the working fluid than in the THD and SSD for a specific value of the twist ratios in the laminar flow regime. Comparison of results of twisted CW-CCW square duct, hexagonal duct with the experimental results of Bhadauriya et al.[2] at twist ratio 11.5 is shown in Table (4). It is seen that twisted CW-CCW duct has 55% higher Nusselt number than Bhadauriya et al.[2] in laminar flow regime i.e up to Re of 3000 at the same twist ratio. But in turbulent regime i.e Re value more than 3000, Nusselt number of Bhadauriya et al.[2] is high.

Numerical results for both laminar and turbulent systems are depicted for ratios between...
Table 4. Comparison of results at Tr-11.5

| Result type | Re   | Tr   | Twisted(CW-CCW)duct | Hex duct | Bhadauriya et al. |
|-------------|------|------|---------------------|---------|------------------|
| Nusselt number | 1000 | 11.5 | 13.02               | 8.1     | 5.4              |
|              | 2000 | 11.5 | 15.13               | 9.3     | 6.56             |
|              | 3000 | 11.5 | 16.14               | 11.62   | 7.46             |
|              | 10000| 11.5 | 32.78               | 27.95   | 43.45            |
|              | 30000| 11.5 | 71.83               | 64.19   | 79.51            |
|              | 50000| 11.5 | 106.95              | 97.44   | 114.1            |

16 · 5, 11 · .5 and 7 · 5. Figures(8) to (14) is a comparison between the Nusselt number and a friction factor with a twisted hexagonal duct and straight square duct. Figure (8) shows that the Nusselt number maximum for twisted squared duct CW-CCW compares to the twisted hexagonal duct and straight square duct for a twist ratio of 16 · 5 in the laminar flow system. But in the turbulent flow regime, it is seen from Figure (9) that the Nusselt values are more in straight square duct than the other CW-CCW and hexagonal twisted duct. Similarly, Figures (10) to (13) represents the Nusselt values for twist ratios of 11 · .5 and 7 · 5 in the laminar as well as in a turbulent flow regime. It is depicted that in the laminar regime, CW-CCW twisted duct dominant more than the other ducts but in turbulent regime, straight duct bears the high Nusselt number. Figures(14) and (15) indicated the comparison between the twisted hexagonal duct and the twisted CW-CCW square duct at different twist ratios such as 7 · 5, 11 · .5 and 16 · 5 in laminar and turbulent regime. It is found that in the laminar regime, the CW-CCW duct with a twist ratio of 7 · 5 has a higher Nusselt number. Likewise, in the turbulent regime, the straight square duct has higher Nusselt values than the twisted CW-CCW square duct and twisted hexagonal duct.

9. Average Coefficient of Friction
For different Reynolds numbers and twist ratios, the friction coefficient of the medium (f) of the fluid (air) flowing through the THD and the square twisted channel CW-CCW is calculated by the formula Eq.(10). The results of CW-CCW and THD are compared with the square twisted channel (SSD). The friction factor values in the CW-CCW twisted channel are very small as THD and SSD using air as the working fluid shown in Figure (15). It also shows that the friction factor first increases when the Reynolds number increases. When some Reynolds number is reached, the rate of decrease of the friction factor slows down. It is referred to the value of Reynolds number, which causes a abrupt changes in the slope of the coefficient of friction, as the critical Reynolds number. As shown in Figure (16), the critical Reynolds number was found between 2700-3000 for both structures, heralding the end of the laminar system and the beginning of the transition to the turbulent flow system.

10. Conclusion
The numerical results shown above concluded that the Nusselt number increases as the twist ratios decrease. In laminar mode, the twisted CW-CCW square channel has a higher Nusselt number than all the other ducts used for the design of heat exchangers such as twisted hexagonal channel, straight square channel and twisted square duct for the same twist ratios under the same operating and boundary conditions and constant pumping power. Percentage increase in Nusselt number of twisted CW-CCW square duct is nearly about 55% more than the results of Bhadouariya et al. [2] and 37.7% more than twisted hexagonal duct respectively. But in turbulent mode, it is observed that twisted CW-CCW square duct is not performed well compared to other duct used in this research. The friction factor can also be inferred that the
friction factor of the CW-CCW twisted square channel is smaller than the twisted hexagonal channel by the same values of the twist ratios. In this present work, the result shows that in the laminar regime, it is referred to use the twisted CW-CCW square duct than the other ducts due to the clockwise and counterclockwise design of the duct which generates higher secondary flow inside the duct resulting in high heat transfer. In future, the use of nanoparticle in the place of normal working fluid such as air may increase the heat transfer.
Figure 11. Turbulent flow regime (Tr-11.5)

Figure 12. Laminar flow regime (Tr-7.5)

Figure 13. Turbulent flow regime (Tr-7.5)

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12. Nomenclature
A : cross-sectional area of twisted duct, $m^2$
Tr : Duct twist ratio, dimensionless ($P/d_h$)
h : Coefficient of heat transfer, $W/m^2 K$
Figure 14. All ducts comparison at different twist ratios (Laminar)

Figure 15. All ducts comparison at different twist ratios (Turbulent)

Figure 16. Friction factor comparison at several twist ratios

\( Q_c \) : heat duty of cold fluid, W  
\( q' \) : heat flux of twisted wall, W/m\(^2\)  
CW : clockwise twist  
CCW : counter-clockwise twist  
T : twist wall temperature, K  
\( d_h \) : Hydraulic diameter, m  
\( C_p \) : fluid specific heat, J/Kg K  
\( Q_h \) : heat duty of hot fluid, W
P : pitch of the twisted channel, m
$P_r$ : Prandtl number, dimensionless
$\dot{m}_c$ : Mass flow rate of cold fluid, Kg/sec
$T_{hi}$ : inlet temperature of hot fluid, K
$T_{ho}$ : outlet temperature of hot fluid, K
$T_{ci}$ : inlet temperature of cold fluid, K
Re : Reynolds number,$(\rho v_m d/\mu)$
$T_{co}$ : outlet temperature of cold fluid, K
V : velocity of fluid flow, m/sec
di : internal diameter of cylinder, m
k : conductivity of fluid, W/m K
do : external diameter of cylinder, m
Nu : Nusselt number, dimensionless
SSD : straight square duct, m
L : periodic length, m
TSD : twisted square duct, m
\bar{f} : average friction factor
CW-CCW : clockwise-Counterclockwise
THD : twisted hexagonal duct
$\dot{m}_h$ : mass flow rate of hot fluid, Kg/sec

Subscripts

m : mean
o : outlet
i : inlet
w : wall
b : fluid’s bulk temperature

Greek symbols

$\rho$ : fluid’s density, kg/m$^3$
$\tau$ : shear stress of local wall, N/m$^2$
$\nabla$ : vector calculus operator
$\mu$ : dynamic viscosity, N-sec/m$^2$

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