Heat transfer and fluid flow characteristics within non-circular duct

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Abstract. Heat transfer and the characteristics of a fluid flow of water flows in the twisted square duct are studied numerically using commercially available software. Numerical investigations of fully-developed laminar fluid flow within duct was performed for water with constant wall temperature boundary conditions, twist ratios of 16.5, 11.5, and 7.5, Reynolds number 100-6000, Prandtl numbers 8 and 10. The numerical results of friction factor and Nusselt number found in this present study were compared with the results obtained from the correlations for the same working fluid such as water. Nusselt number in the twisted square duct is higher than the result found by correlation together with a lower friction factor. Percentage increase in average Nusselt number found for twist ratios 16.5, 11.5, and 7.5 compared to the theoretical values was 25.56, 33.21, and 37.73 The heat transfer coefficients of the square twisted duct are increased with a lower twist ratio and higher Prandtl number.

1. Introduction

Energy saving is the main need of this universe in these recent years. Due to the industrial importance of the compact heat exchanger, most of the research works are available in this area. The twisted duct used in this study is therefore especially relevant to the design of industrial heat exchanger devices. The reason of using twisted duct is that it creates swirl inside the duct which intermixed the flow rapidly and causes to increase the heat transfer more. Based on this key point, two techniques are employed to improve the efficiency of heat exchangers such as passive and active heat transfer enhancement techniques. Passive heat transfer enhancement techniques that employed no external power supply and in opposite active techniques that work on the principle of external power supply. Twisted ducts work on the principle of passive heat transfer technique [1] and found that ducts with larger traced circle diameters offer higher values of Nusselt number and friction factor.

Wang et al. [2] investigated the flow of air within three ducts, namely, the square duct of twisted uniform cross-section, twisted divergent, and convergent, and verified that the duct with a divergent twist improves the heat transfer while convergent section decline.

Yang et al. [3] experimentally examined the water flow performance in the twisted elliptical tubes and tested the impact of parameters obtained by different geometry such as aspect ratio and twist pitch. The tubes examined were found to increase pressure drop and heat transfer compared to straight tubes. Khoshvaght et al. [4] investigated heat transfer in laminar and monitored workflows in twisted mini-
channel with variant cross sections and revealed that the best thermohydraulic performance was obtained in a twisted square mini channel. An experiment on convection inside a novel alternating elliptical axis tube and heat transfer and flow resistance correlations were performed for Reynolds number of 500-50000. It was depicted that at low flow resistance, the heat transfer rate is higher [5].

Mahato et al. [6] studied that use of nanoparticles with different concentrations into water inside the T\textsubscript{w}SD (twisted square duct) enhanced more heat than normal water. Maximum limits of concentration of nanoparticles mixing into the water were also specified to 1%.

The characteristics of heat transfer and friction factor of the twisted oval tube were numerically studied by a realized k-epsilon method with varying 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 [7].

Yadav et al. [8] studied flow of fluid inside the non-circular duct and investigated the properties of heat transfer and friction factor through it for a laminar flow regime. They used the twisted band inside the channel with several twisting ratios such as Tr = 3.5, 4.5, 5.5, and 6.5. They revealed that the twisted tape insertion into the duct increases the Nusselt number and the friction factor.

Mahato et al. [9] found heat transfer gain using a twisted hexagonal channel and with a twisted square channel. Comparisons between them were also presented. They found that increasing the number of channel sides further improved heat transfer.

Masliyah and Nandakumar [10] simulated a fully developed stable laminar flow with a rotational coordinate system through the twisted square channels. The axial and fluid lines fall to preserve the natural dimension of the registered problem. The temperature was assumed to be constant along the periphery of each wall. They observed that the amplification of heat transfer is greater when the fluid is in rotary motion for the rotation ratio of 2.5 and the Reynolds number range from 100 to 1000.

Bhadouriya et al. [11, 12] tested the airflow through the twisted square channel to define the properties of heat transfer and pressure drop covered by different rotational conditions, the Reynolds and Prandtl numbers are (16.5 and 11.5), (1, 00 to 10,000) and (0.7, and 20). They noticed that Nusselt and friction values of twisted square duct (T\textsubscript{w}SD) are higher in comparison to a square straight channel.

The research work declared above defines that variant research has been performed to increase the transfer of heat by incorporating twisted tapes inside the plain duct which proces swirl in it. As twisted tape restrict the flow area inside the plane duct causes to increase the higher pressure drop. This problem was improved by incorporating twisted square duct (T\textsubscript{w}SD) instead of twisted tape considering twist ratios like 11.5, 16.5, and 7.5. Very few research works were available in the area of heat transfer enhancement within the T\textsubscript{w}SD where water is considered as a working fluid with different Prandtl numbers (Pr). This research seeks to find the characteristics of heat transfer and friction factor in T\textsubscript{w}SD by considering water as a normal fluid and compare these results with the Gnielinski correlation.

Some important terms of twisted duct are depicted in Figure 1.

Pitch (P): The distance among two successive points on the pipe length where the position of the cross-section completely matches one another. Along with the pitch distance, the cross-section is rotated by 360°.

Twist ratio (T\textsubscript{wr}): The ratio between pitch (P) to hydraulic diameter (d\textsubscript{h} = 4*Area / Perimeter) of the twisted square duct defines the T\textsubscript{wr}. 

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2. Methodology

2.1. Geometry of twisted duct
The inlet diameter \(d_i\), outlet diameter \(d_o\), and thickness \(\delta\) used for the twisted duct are 18.5 mm, 20.5 mm, and 2 mm. Twist ratios used are 7.5, 11.5, and 16.5, respectively. The inside diameter \(D_i\) and outside diameter \(D_o\) of the straight circular cylinder tube is 31 mm and 37 mm depicted in Figure 2.

3. Numerical methods

3.1. Governing equations and boundary conditions. Generally, shell and tube heat exchanger used concentric tubes which are utilized in this study. This tube was cooled by a counter-flow heat technique to provide a greater temperature gradient over the shell and tube surfaces. The flow considered in this study is fully developed hydrodynamically and thermally, periodic, incompressible, and in steady-state. The characteristics of the flow are repeated along the length \(L\) of the channel, the flow is said to be periodic. Governing equations for such type of flows are the following

\[
\text{Continuity: } \nabla \cdot \vec{v} = 0 \\
\text{Momentum: } \rho \nabla \vec{v} = -\nabla p + \mu \nabla^2 \vec{v} \\
\text{Energy: } \nabla T = \alpha \nabla^2 T
\]

Where \(\alpha = \frac{K}{\rho C_p}\) is thermal diffusivity.
The acceleration term which is treated for the twisted tube, that is, $\rho \frac{\partial \vec{V}}{\partial t} \vec{V}$ is not linear, and therefore analytically, it is not easy to solve. This can be solved by the final volume Discretization method which provides an approximate solution.

The twisted wall was kept at a constant temperature at all times and the outer walls of the cylinder were considered adiabatic which is illustrated in Figure 2. On the interior and exterior walls, it is considered a slip limit. A constant success rate ($\dot{m} = 0.35 \text{ kg} / \text{s}$) is used inside the outer cylinder. At the input of each repeated module, a constant flow rate and the periodic boundary condition are chosen for the upper channel. In general, the degree of dependency is $10^{-3}$, $10^{-5}$, and $10^{-6}$ considered for continuity, velocity, and energy respectively.

4. Grid independence test

An unstructured mesh is generally used for numerical analysis of the twisted square duct consequently of high complexity and frictionally twisted configuration. A grid-independent study is carried out to make sure the independence of the grid is considered for calculations.

Nusselt numbers were established for several disorderly grids with a constant Reynolds number of 500 in the grid independence study. Grid independence was found for the values of grid which consist of 284647 elements and size of face $1.1419 \times 10^{-3}$ mm. It is illustrated in Figure 3 that after the grid elements of 284647, there was not much increase in the Nusselt number observed. Therefore, this grid is used for subsequent calculations. Grid independence values are outlined in table 1.

| Elements of grid | Nusselt values | Size of the face (mm) $\times 10^{-3}$ |
|------------------|----------------|-------------------------------------|
| 194159           | 6.152          | 1.3411                              |
| 217349           | 6.273          | 1.2787                              |
| 243679           | 6.294          | 1.2212                              |
| 284647           | 6.266          | 1.1419                              |
| 305526           | 6.265          | 1.0831                              |
| 346225           | 6.267          | 1.0092                              |

Figure 3. Schematic of grid validation
5. Mathematical evaluation

The three-dimensional simulations for the periodic, stable, and incompressible flow through the model were compiled by commercially available software. Since the geometry of the channel repeats the path several times along the pipe, the use of periodic flow is indicated. As the flow considered is periodic, the numerical simulation may be limited to the repetitive units of all pipelines. Therefore, the modeling length (L), P / 4 is sufficient for modeling a square channel. Based on the numerical results, the laminar flow regime is treated within the Reynolds number range of 1,00-2,000.

Equation (4) and equation (5) depict the heat transfer rate for hot fluid (Q_h) i.e. water and cold fluid (Q_c) i.e. air where T_hi, T_ho, C_ph and m represent the inlet and outlet bulk temperature, specific heat of hot fluid, and mass flow rate. Equation (6) is used for the calculation of the average heat transfer rate (Q̅) where A_p and ΔT_LMTD stand for heat transfer area and logarithmic mean temperature difference.

\[ Q_h = \dot{m}_h C_{ph} (T_{hi} - T_{ho}) \quad (4) \]
\[ Q_c = \dot{m}_c C_{pc} (T_{co} - T_{ci}) \quad (5) \]
\[ Q̅ = (Q_h + Q_c)/2 = U A_p \Delta T_{LMTD} \quad (6) \]

Overall heat transfer coefficient (U) is calculated by equation (7) where R_w acts as the resistance of twist wall.

\[ 1/(U A_p) = 1/(h A_p) + 1/((h A_p) + R_w) \quad (7) \]

Resistance (R_w) obtained by the wall of the twisted duct is neglected because of the smaller wall thickness (δ = 2 mm) and superior thermal conductivity of the twisted duct. Heat transfer coefficient (h) is studied towards length of twisted duct illustrated in equation (8) where T_b, T_w, and q′ denote the mean bulk and wall temperature, and wall heat flux.

\[ h = \frac{q′}{T_w - T_b} \quad (8) \]

The Nusselt values are computed by the formula displayed in equation (9)

\[ Nu = (h d_h)/k \quad (9) \]

To evaluate the hydraulic performance of duct, friction factor determined by the equation (10) where τ_w, and q′ represents the wall shear stress and heat flux.

\[ f̅ = 2 \times \frac{\tau_w}{\rho V_{in}^2} \quad (10) \]

6. Results and discussions

6.1. Validation of results

The numerical analysis of pure water flowing through the channel was verified and the results were obtained with theoretical values given by standard correlations that are widely used in the literature. The numerical values of the Nusselt number were correlated with the theoretical values estimated by Gnielinski equation shown in equation (11) and it was realized that the deviations were within 4.5% Figure 4.

\[ Nu = \frac{\xi/2(Re-100)Pr^{0.5}}{1+12.7(\xi/2)^{0.8} (Pr^{2/3}-1)} \quad (11) \]
Where \( \xi = \frac{1}{(1.58 \ln Re - 3.82)^2} \)

The numerical friction factors (f) were compared with the theoretical values obtained using the Blasius equation depicted in equation (12).

\[
f = \frac{0.316}{Re^{0.25}}
\]  

(12)

The numerical values of the friction factor determined in reasonably good agreement with the theoretical values obtained from the Blasius equation are shown within a 5% shift from Figure 5.

6.2. Nusselt number and average friction factor

After validating the numerical results, the simulation simulates for the twist ratios of 7.5, 11.5, and 16.5
for Re 100-6,000, Prandtl numbers of 8, 10. The Nusselt numbers outlined against Reynolds numbers presented in Figures 6, 7, 8, and 9 and Obtained results of TwSD are plotted against the corresponding theoretical values obtained by Gnielinski correlation using water as working fluid. Result reveals that Nusselt numbers were observed to be superior in TwSD than the values of Gnielinski correlation. It is also seen that an increase in Pr and at lower Tr, Nusselt number increases.

Figure 6 shows that the TwSD at Tr of 7.5, Pr number 10 is having a higher Nusselt value compared to the Nusselt number obtained by the Gneilinski equation. Similarly, in Figure 7, and 8 Nusselt numbers are found to be higher in Tr of 16.5 and 11.5 of Pr values of 10 compared to the Nusselt number found in Gnielinski correlation for the same working fluid. Nusselt number found to be maximum at twist ratio 7.5 than the Nusselt value at twist ratios of 11.5, 16.5, and theoretical value of Gnielinski equation which is shown in Figure 9. Figure 10 depicts that a very less friction factor is observed in a TwSD with twist ratios 16.5, 11.5, and 7.5 compared to the theoretical friction factor values of the Blasius equation.

Figure 6. Comparison of numerical Nusselt number of Tr-7.5 with Gnielinski equation.
Figure 7. Nusselt number comparison of Tr-11.5 with Gnielinski equation.

Figure 8. Nusselt number comparison of Tr-16.5 with Gnielinski equation.
7. Conclusion
In this study, the thermal-performance of water flow inside the twisted square duct is numerically compared with the correlation value of the Gnielinski and Blasius equation. The twisted square duct with twist ratios 16.5, 11.5, and 7.5 was used in the numerical simulation. The average increase in Nusselt number and friction factor corresponding to different twist ratios for the Reynolds value 100-6000 and Prandtl number of 8,10 were properly defined. It is seen that increase in Prandtl number, Nusselt number increases, and average friction factor decreases.

The percentage increase in average Nusselt number found for twist ratios 16.5, 11.5, and 7.5 compared to the theoretical values of Gnielinski was 25.56, 33.21, and 37.73 and similarly, for friction factor,
percentage decrease was found to be 76.1, 77.08, and 73.45 respectively for twist ratios 16.5, 11.5, and 7.5. It is also found that Nusselt numbers are higher at T_{tw} of 7.5 compared to 11.5, 16.5, and the theoretical value of Gnielinski correlation.

A very less friction factor was observed in T_{tw}SD (twisted square duct) with Tr (twist ratios) of 16.5, 11.5, and 7.5 than the friction values obtained by the Blasius correlation. In the future, various additives like nanoparticles with different concentrations can be used with normal water to increase the heat transfer and to some extent; different geometry can also be checked for this heat transfer enhancement.

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