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Numerical Analysis of Heat Transfer and Fluid Flow through Twisted Hexagonal and Square Duct and their Comparisons

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The purpose of this study was to investigate the heat transfer and friction factor characteristics for hexagonal twisted duct. Numerical simulations were used to study the swirling flow inside a twisted duct of hexagonal cross section cooled by using counter flow in a shell and tube arrangement. Nusselt number and friction factor was calculated for a wide range of Reynolds number (Re) ranging from 1,000-1, 00,000. Laminar to turbulent transition point was identified. The results of twisted hexagonal duct were compared to that of twisted square duct. Results revealed that in the laminar regime, Nusselt number (NU) for hexagonal twisted duct is greater than that of square twisted duct, whereas in the turbulent region the corresponding Nusselt number for square twisted duct was greater than its hexagonal counterpart. However, in case of friction factor, a regular decrease in friction factor observed with increasing Reynolds number. The survey concluded that hexagonal ducts can be used in laminar flow regime and square ducts for turbulent flow regime. The impacts of the obtained results are practically applicable in power plant, aerospace, biomedical purposes.

1. Introduction

Non-circular ducts generally used in the aerospace, biomedical, nuclear and electronic uses due to space and price constraints especially in compact heat exchangers. With the increasing requirements of improved thermal management in various fields, augmentation of heat transfer is becoming of greater importance. Heat transfer augmentations are two types: Active heat transfer augmentation techniques, which require the external application of energy. On the other hand, passive heat transfer requires no external application of energy. It involves processes like use of twisted ducts, twisted tape inserts, use of counter flow, using ribs, fins or additives to increase heat transfer coefficient. The use of twisted tapes or twisted duct creates a swirl and intermixing in the flow, which in turn increases the rate of heat transfer as compared to plain duct. In case of twisted duct, a swirl is created both in inner and outer surface thus enhancing heat transfer in both the cold and hot fluids as compared to twisted tape inserts. (Chang et al., 1988) inspected flow through elliptic tube with large twist ratios (TR=21, 53,106) in the laminar regime by using finite difference method. (Bhadouriya et al., 2015), a, b, studied experimentally and numerically the pressure drop and heat transfer coefficient in an inner twisted square duct and an outer circular pipe over a large number of Reynolds number (1, 00-10,000) and Prandtl number (0.7-20) and found the friction factor and Nusselt values to be higher as compared to that in the case of straight duct. (Yadav et al., 2015) investigated the heat transfer and friction factor characteristics through a non-circular duct for laminar flow regime by using twisted tape inside the duct and with twist ratio (TR=3.5, 4.5, 5.5, 6.5) and found that with the insertion of twisted tape inside a duct, the Nusselt number and friction factor to be greater than the duct without twisted tape. (Suri et al., 2017) studied the heat transfer and friction factor characteristics through heat exchanger by using multiple squares perforated twisted tape inserts. The experimental conclusion for Reynolds value 5,000-27,000 perforation Ratio 0.083-0.033 and twist ratio 2-3.5 and constructed considerable enrichment in heat transfer rates over without square perforated tube. (Bishara et al., 2010) investigated the hydrodynamic and thermal characteristics of periodic flow of a single Newtonian fluid through elliptical helically twisted tube using constant wall temperature boundary conditions for Reynolds no range of

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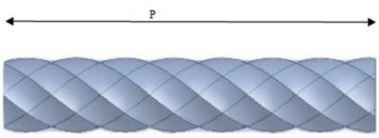
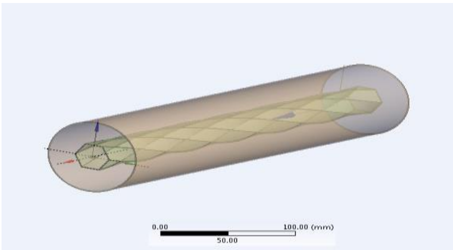
10-1,000 and Prandtl number of 3, with aspect ratio 0.3,0.5 and 7, twist ratios 6,9 and 12. The authors found significant deviations in the velocity profile and temperature profile from that of straight elliptical tubes. (Camaraza et al., 2018), made investigation for heat transfer calculations during airflow through the finned tubes bank in air cooled condenser systems (ACC). The model was correlated with a total of 736 sets of available experimental data. (Rasool et al., 2018), analyzed of heat transfer and friction factor for turbulent flow of air through a two-pass square channel, having ribs of various cross-sections and investigated the investigating the potential impact of differing the shape of ribs for a comparative roughness pitch (p/e) of 10. (Klemes et al.,), determined the energy efficiency by cost-optimal pre-and post-treatment pathway for an AD of lignocellulosic waste by applying P-graph. The economic balance between the main operating cost, yield and quality of products were considered. (Hong et al., 1976), extensively studied heat transfer enhancement using twisted tape inserts and the Nusselt number (Nu) for fully developed flow was found to be a function of tape twist ratio, Reynolds number, and Prandtl number.
Literature review suggests that heat transfer and flow resistance characteristics of fluid flow inside twisted duct of square and elliptical cross section was studied by few researchers but there are very few researches has been done on twisted hexagonal duct where it's been reviewed only that by increasing number of sides, Nusselt number increases. But there is no any numerical simulation has done till now. Thus, the paper aims to study the non-circular twisted hexagonal duct and investigate the effect of this "non-circularity" on that heat transfer characteristics of the fluid while flowing through duct.

2. Terminologies associated with twisted hexagonal duct

The common terminologies that are related to twisted hexagonal duct, indicated in figure 1(a) are
Pitch (P): The distance between two consecutive points along the length of the tube where the orientation of the cross section coincides after 360^{o} rotation.
Twist ratio (TR): A geometrical parameter used to describe the low through twisted ducts. It is defined as the ratio of the pitch to hydraulic diameter (d) of the duct (TR=P/d).



(a) Terminology (TR-7.5)



(b) 3-dimentional model

Figure 1: Twisted hexagonal duct model for numerical simulation

3. Mathematical formulation

This study considered concentric tubes as used in a shell and tube heat exchangers and cooling down by using counter flow, to provide greater temperature gradient across tube and shell surface. The flow is incompressible, steady state, fully developed hydro dynamically and thermally, periodic in nature. A flow is said to be periodic in nature when the flow characteristics repeat themselves after a length (L) of the duct.
The governing equations for such a flow include

Continuity:$\nabla . \boldsymbol{v} = 0$

Momentum: $\rho \boldsymbol{v} . \nabla \boldsymbol{v} = -\nabla p + \mu \nabla^2 \boldsymbol{v}$

Energy: $\rho \boldsymbol{v} . \nabla T = \nabla^2 T$

For helically twisted tube the convective acceleration term, $\rho \boldsymbol{v} . \nabla \boldsymbol{v}$ is nonlinear and thus cannot be readily solved analytically. Thus, resort to approximate solution using finite volume Discretization.

3.1 Boundary conditions

The walls of the twisted duct are at constant temperature, whereas the outer cylinder walls assumed purely adiabatic. The constraint of no slip boundary condition is applied at inner and the outer duct walls. A constant mass flow rate (Δm=0.35 kg/s) is applied at the inlet of the outer cylinder whereas a periodic boundary condition is considered for the twisted duct with a constant mass flow rate at the inlet of each repeating module.

4. Computational grid

Since the geometry under consideration is friction twisted configuration and hence of high complexity, unstructured meshing is used for the three-dimensional simulation. Grid independence study carried out in order to ensure the solution is independent of the grid used for computation. In the grid independence study, Nusselt number found for different unstructured grids at a constant Re value of 1,000. Grid independence found to occur for a grid consisting of 85,861 elements with maximum face size of 5.6 mm. This grid is thus, used for further computations. The results of the grid independence study are summarized in table 1

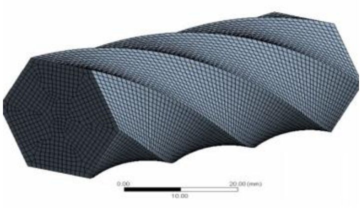


Figure 2: Computational grid used for 3-dimentional simulation of (TR-11.5)

Table1: Numerically measured parameters

Elements	Nusselt number	Maximum face size(mm)
84781	6.73	6
82013	6.896	5.8
83593	6.913	5.7
85861	6.882	5.6
92095	6.882	5.5

5. Numerical methodology

Ansys Fluent commercially available software is used for 3-Dimentional simulation of the incompressible, steady state, periodic flow across the model. Periodic flow maybe said to occur here because the geometry of the duct repeats several times over the length of the duct. In case of periodic flows, the numerical simulation can be restricted to a single repeating unit of the entire duct. Thus, in case of a hexagonal duct a model length (L) of P/P/2 is sufficient for simulation. Based on experimental results the laminar range is said to be in the range of Re 1, 00-3,000. Turbulent flow studies are undertaken for Re>10,000 using k-ε (k-epsilon) model. The quantities required for calculations obtained from the post-processing module of the software. The quantities used in calculations are:

T_b = Mean Bulk Temperature

T_w = Mean wall Temperature

q' = Wall heat flux

τ_w =Wall shear stress

Nusselt and friction factor values are calculated for different Reynolds values (1,000-1, 00,000) and twist ratios (2.5, 5, 7.5, 11.5, and 15.5). The heat duty of the hot fluid (water) and cold fluid (air) is calculated by following Eq. (1) and Eq. (2) given below

$$Q_h = \dot{m}_h C_{ph} (T_{hi}-T_{ho}) \quad (1)$$

$$Q_c = \dot{m}_c C_{pc} (T_{co}-T_{ci}) \quad (2)$$

Numerical model is well insulated to prevent heat losses to the surrounding. The average heat duty ($\overline{Q}$) for the calculation purpose can be calculated by Eq. (3) shown below

$$\overline{Q} = (Q_h + Q_c)/2 = UA_P \Delta T_{LMTD} \quad (3)$$

With the known value of heat transfer area (A_P) of the heat exchanger and ΔT_{LMTD} (log mean temperature difference) the overall heat transfer coefficient (U) can be calculated. Heat transfer coefficient (h) separately for two sides of flow i.e. hot side $(h)_h$ and cold side $(h)_c$ to calculating the overall heat transfer coefficient (U) is shown in Eq. (4)

$$1/(UA_p) = 1/(hA_p)_c + 1/(hA_p)_h + R_w \quad (4)$$

As the thickness of hexagonal twisted duct is 1mm, hence wall resistance (R_w) of duct is defined as Wall resistance (R_w) =Wall thickness/Thermal conductivity of steel = 0.001/50.2 =1.99×10^{-5} m^2 k/w
Because of small wall thickness and higher thermal conductivity of hexagonal twisted duct, the wall resistance is neglected during the analysis. The convective coefficient of heat transfer (h) is calculated along the length of twisted duct by the formula:

$$\mathrm{h} = \; q'/(T_\mathrm{w} - \mathrm{T_b}) \tag{5}$$

The Nusselt no. is then calculated using the formula shown below in Eq. (6):

$$\mathrm{NU} = (\mathrm{h.d})/\mathrm{k} \tag{6}$$

At the different location along the length of the twisted duct, wall shear stress (τ_w), heat flux (q'), bulk temperature (T_b) is calculated by using commercially available software. Friction factor (f) is obtained from the wall shear stress (τ_w) as

$$\bar{f} = 2 \times \frac{\tau \mathrm{w}}{\rho \mathrm{V_m^2}} \tag{7}$$

Table 2: Parameters used in Ansys fluent

Models	Boundary Conditions	Discretization	Convergence
Solver - Segregated	Inlet --- periodic	Pressure ---2nd order upwind	
Formulation - implicit	Outlet -- periodic	Momentum--2nd order upwind	Continuity-10^{-3} Velocity-- 10^{-6}
Space - 3-Dimensional	Wall ---- const. temp.	Energy- 2nd order upwind	Energy--10^{-7}
Viscous -k-ε	Period type—Translational	Turbulent Kinetic energy- 2nd order upwind	
k-ε - RNG	Condition---Mass flow specified	Pressure velocity--2nd order upwind	

6. Validation of code

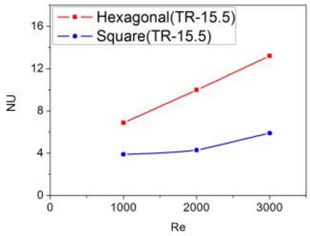
The numerical code used by the commercial software was validated against experimental data by performing simulation for heat transfer and friction factor characteristics of twisted square duct and comparing the numerical results with the experimental values obtained by Bhadouriya.et al. The comparison suggests deviation of the friction factor values obtained from numerical simulation from that of experimental value to be 6.1% and that in case of Nusselt number values to be about 5.8% shown in table (3).

Table 3: Comparison of results

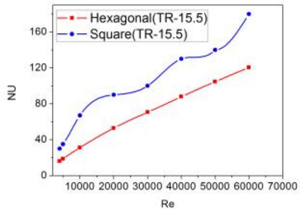
Reynolds Number	Nu (numerical)	Nu (experimental)	Friction (numerical)	Friction (experimental)
930	4.027	3.8	0.0032	0.003
6700	183.12	172.7	0.006	0.0055

7. Results

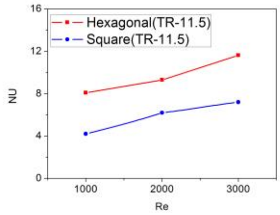
After validation of the numerical methodology simulations have done for twist ratios 2.5, 5, 7.5, 11.5 and 15.5 for Reynolds numbers (1,000-1, 00,000). The Nusselt value has been plotted against Reynolds number figure.3 (a, b, c, d, e, f, g and h) for hexagonal duct and comparisons are made for corresponding values of square duct. Result shows that in the laminar regime, the Nusselt values are much greater for hexagonal twisted duct than square twisted duct. In the turbulent regime, however it is found that the Nusselt value for the square duct was greater than that for hexagonal twisted duct. It was found that for increasing twisted ratio, the Nusselt values for hexagonal twisted duct reduces more strongly than the square twisted duct.



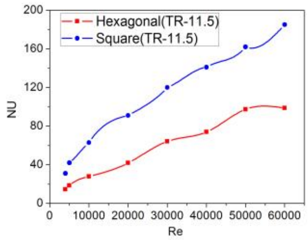
(a) Laminar flow regime
(TR-15.5)



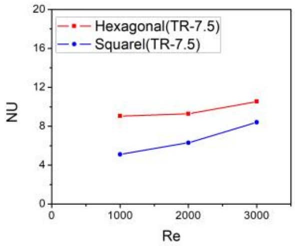
(b) Turbulent flow regime
(TR-15.5)



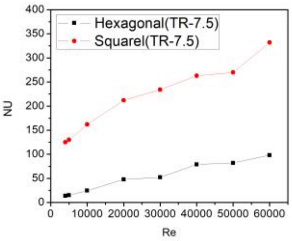
(c) Laminar flow regime
(TR-11.5)



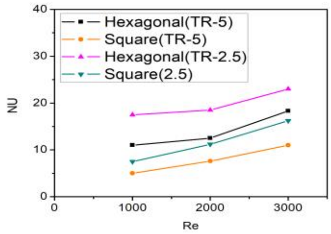
(d) Turbulent flow regime
(TR-11.5)



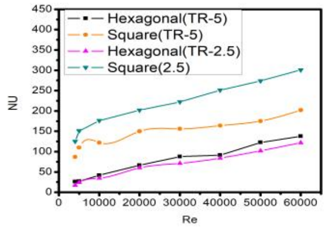
(e) Laminar flow regime
(TR-7.5)



(f) Turbulent flow regime
(TR-7.5)



(g) Laminar flow regime
(TR-2.5, 5)

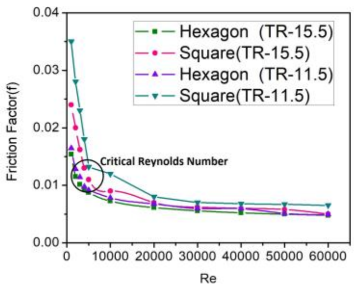


h) Turbulent flow regime
(TR-2.5, 5)

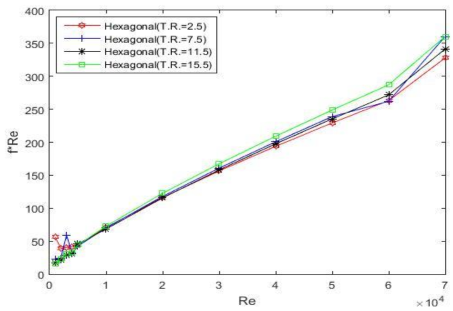
Figure 3: Comparison of heat transfer value of hexagonal with square for different twist ratio

8. Average friction factor

Average friction factor Eq. (7) values were found for different Reynolds number and for twisted ratio. The friction (f) values found to be less than that of square duct throughout the entire laminar as well as turbulent regime, although in the turbulent regime the difference in friction values for the hexagonal and square duct reduces and attains almost similar values. The Reynolds number at which instant change in slope of friction factor value is termed as critical Reynolds number. This critical Reynolds number indicates the end of laminar flow regime and start of transition to turbulent flow regime figure 4(a).



(a) Friction factor comparison



(b) Average f*Re

*Figure 4: friction factor and average f*Re value*

The average f*Re value obtained for different twist ratios in the laminar flow regime are presented in Figure 4(b). From fig. it is seen that average f*Re value for twisted duct is function of both Re and twist ratio. The maximum value of average f*Re is 49.2 obtained for Reynolds number 3000 for hexagonal duct at twist ratio of 2.5 and as we increase the twist ratio and Reynolds number, the average f*Re value also increases and it is maximum at twist ratio of 15.5 i.e. 371 at Reynolds number 80,000. Increment in average of f*Re value with respect to twist ratio due to the presence of swirling flow in twisted duct.
Individually the trends for product of friction factor f and Reynolds number found to be approximately similar to that found for the square case by Bhadouriya et al.

9. Conclusion

From the results mentioned above, it is clear that in the laminar region the Nusselt value is higher for hexagonal duct than square twisted duct. This is essentially because with increase in number of sides of a non-circular duct, the blocking effect decreases as a result of which the fluid velocity at the corner of the wall increases. This leads to greater swirling flow in the duct and hence greater heat transfer across the walls of the duct. However, with increase in the Reynolds number beyond the turbulent regime, the Nusselt value for square duct found to be greater. This means that the effect on increase in velocity due to reduced blocking effect becomes less effective in the turbulent regime. For increasing twist ratios too, the Nu value decreases, similar to the findings of Bhadouriya et al. in the square case. The friction factor for hexagonal duct is found to be greater both in turbulent and laminar regime. This can be easily explained from the fact that with increasing velocity at the corners (as explained above) increases the mean velocity of the flow as well as the Reynolds number. In both cases, the increasing velocity reduces friction factor. Although here it is found that with increase in number of sides of a non-circular duct, the Nusselt decreases as well as the friction factor (f) reduces, the number of sides cannot be indefinitely reduced. Because with the increase in number of sides, the shape of the cross section gets closer to a circle and as a result the swirling effect due to twisted configuration decreases, and for a circular duct, no twisted configuration is possible. Thus the number of sides for maximum heat transfer is yet undecided and further optimizations are yet to be made which present the scope for further research to practical implement non circular duct in industry.

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