Numerical study of heat transfer enhancement through a circular tube fitted with and without rectangular cut twisted tape insert

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Abstract. Numerical simulations have been carried out to investigate the heat transfer enhancement in a tube fitted with a rectangular cut twisted tape insert and compare the data with a smooth tube. For this purpose, CFD (Computational Fluid Dynamics) was utilized to simulate the three-dimensional model with appropriate boundary conditions. The numerical results show that the flow field can be adjusted and the thickness of boundary layer can be decreased by inserting the rectangular cut twisted tape insert so that the heat transfer can be enhanced in the tube. Water was used as the working fluid. The results obtained show an increase in Nusselt number as a result of increasing heat transfer rate for the tube provided with rectangular cut twisted tapes than the plain tube. The pressure drop across the tube wall for twisted tape with the rectangular cut is higher than the plain tube. This increase in the rate of heat transfer is due to the introduction of secondary swirl flow (due to twisted tape) along with the primary flow.

1. Introduction

Heat exchanger refers to the system that is used for heat transfer between two or more fluids. The sole purpose of heat transfer enhancement is to increase the effectiveness of the heat exchanger. The general mechanism of heat transfer enhancement involves induced swirl flow which potentially promotes fluid mixing resulting in a thinner boundary layer and a higher convective heat transfer rate [1]. The process can be both active and passive. In the active method, desired flow modification is achieved via an external power source. On the other hand, passive techniques include the incorporation of inserts, rough surfaces etc. for the geometrical modification of the flow channel to bring out the desired change in the flow without any external power source. In recent times, there has been a good number of researches performed on the performance augmentation of the heat exchanger. The works include both active and passive heat transfer enhancement techniques.

Megerlin et al. [2] used spiral brush inserts in short channels under high heat flux condition and obtained turbulent heat transfer coefficient enhancement up to 8.5 times. Different types of “Everter” inserts and several disk inserts were investigated by Maezawa and Lock [3]. Although the local heat transfer coefficient increased a significant amount for air flowing in an electrically heated tube, no marked improvement was observed beyond 10 diameters downstream of the insert. Also, a doubling of average heat transfer coefficient was recorded. But, on the negative side, friction factors increased up to 3.5 times. Eiasma-ard et al. [4] observed a 57% augmentation on the heat transfer rate with the combination of wavy-surfaced tube and helical tape compared to wavy-surfaced tube alone. On another investigation [5], Eiasma et al. reported a comparatively better performance in backward flow than the forward one in case of louvered strips. Bhuiya et al. [6] compared various helical tape inserts...
with different helix angles fitted in a tube. The experiment concluded that, under the same operating
condition, decreasing of helix angles increase the Nusselt number, friction factors and thermal
enhancement efficiency. Using a double-helical tape insert with 9° helix angle resulted in the
maximum thermal efficiency of 215%. Zhang et al. [9] performed a three-dimensional numerical study
on heat transfer in a tube with helical screw tape inserts of four different widths and found an
enhancement of heat transfer up to 351% although the friction factor also increased to 1020%.
Saeedinia et al. [10] found an increase in heat transfer of 45% on average in a tube with a wire coil
with the highest wire diameter although the pressure dropped by about 63%. García et al. [11]
compared the pressure drop and heat transfer performance between three types of enhancement
methods involving corrugated tubes, wire coils and dimple tubes and concluded that, for Reynolds
number between 200 and 2000, wire coils should be used whereas dimpled and corrugated tubes are
suitable for Reynolds number above 2000.

In general, a twisted tape enhances the heat transfer coefficient by inducing swirl flow and thus
promoting fluid mixing between core and wall regions. As a result, the thermal boundary layer is
disturbed, causing the enhancement of heat transfer in the existing system. Murugesan et al. [12]
observed an increased heat transfer rate with increasing depth ratio and decreasing width ratio in case
of V-cut twisted tapes. Murugesan et al., on another investigation [13], compared the effectiveness of
classical and square-cut twisted tapes and found out square-cut one to be better in augmenting heat
transfer performance. Thianpong et al. [14] reported a 108% increased heat transfer in a tube with
perforated twisted tape inserts than the plain tube.

Researches have also been done numerically to optimize the performance of twisted tape inserts.
Zhang and Cheng [15] compared numerically the heat transfer performance between classical and
edge-fold twisted tape inserts. Although the pressure dropped by around 74% in edge fold twisted tape
inserts, 9.2% heat transfer enhancement occurred. Rahimi et al. [16] compared the performance of four
different types of twisted tapes- classical, perforated, notched and jagged. The result suggested that
jagged twisted tapes offer the best thermal performance. Cui et al. [17] in their three-dimensional
numerical model, investigated the impact of gap width and operating parameters on thermal
hydrodynamic performance.

On all the above researches it has been found that, although introducing tape inserts improve the heat
transfer performance, but it is achieved by a penalty of increased friction loss. So, there has to be a
compromise between these two to get the optimum performance. In this paper, heat transfer
performance has been studied numerically between circular tubes fitted with and without rectangular
cut twisted tape insert. This paper offers an optimization between heat transfer coefficient and friction
factor based on the Reynolds number. For the numerical simulation, Ansys fluent 16.1 student version
has been used.

2. Description of the computational model
This work aims to compare two circular tubes with and without rectangular cut twisted tape inserts in
terms of heat transfer rate, convective heat transfer coefficient, Nusselt number and friction factor by
simulation. For this purpose, a model is designed and analysed in Ansys Fluent. Pressure based solver
is used to solve the problem. For turbulence modelling, k-ε turbulence model with standard wall
function has been opted. SIMPLE (semi-implicit method for pressure linked equations) algorithm has
been opted for the steady-state numerical analysis to solve the pressure and velocity field for the
computational flow domain. Convergence criteria have been set as 10-5 for all the equations.
2.1. Physical modelling
The circular tube with and without a rectangular cut twisted tape insert has been investigated in the current study. Water as the working fluid entered the tube at an inlet temperature and flowed through the test section. The physical parameters of the tube are shown in Table 1. The properties of water are shown in Table 2. Thermal conductivity of the material of the tube is 202.4 W/m-K. The CAD design of the model is presented in figure 1. The numerical study was carried out at twist ratios (y/w) of 5.25. The ratio of twist length (y) to the twisted tape width at the large end (w) is referred to as the twist ratio. Twist angle is the angle between the reference plane before and after twisting. Some assumptions were made during the simulation process such as:

- The physical properties of water were remained constant and were evaluated at the bulk temperature.
- Steady and incompressible flow.
- The temperature of the tube wall was constant.

### Table 1. Physical parameters of the tube.

| Parameters      | Value  |
|-----------------|--------|
| Tube inside diameter | 26.6 mm |
| Tube outside diameter | 30 mm  |
| Tube test length   | 900 mm |
| Outside surface area | 0.0848 m² |
| Inside surface area  | 0.0752 m² |
| Cross-sectional area | 0.000556 m² |

### Table 2. Properties of water.

| Properties                | Value          |
|---------------------------|----------------|
| Viscosity                 | 0.00087 kg/m-s |
| Density                   | 998.2 kg/m³    |
| Thermal conductivity      | 0.6 W/m-K      |
| Specific heat             | 4182 J/kg-K    |

2.2. Mesh generation and grid independence
A grid independence study has been performed for the Nusselt number for different number of mesh (Table 3) to check the computational accuracy. The results are taken for Re= 2722 using a circular tube equipped with rectangular-cut twisted tape insert. It is found that the difference in Nusselt number between mesh number 495713 and 961055 is 0.5%. Therefore, mesh number 495713 is adopted for this numerical analysis.
Table 3. Mesh independence test.

| Mesh number | Nu  |
|-------------|-----|
| 136581      | 19.28 |
| 259416      | 21.54 |
| 495713      | 24.49 |
| 961055      | 24.62 |

Figure 2. (a) Meshing of the tube; (b) Meshing of the fluid domain.

2.3. Boundary conditions
The numerical simulation has been carried out opting finite volume method. As the boundary conditions, water with 303K temperature has been taken at the inlet. Also, seven different velocities have been considered ranging from 0.089 m/s to 0.161 m/s at the inlet. The velocities were uniform in each of the cases. No-slip boundary condition has been considered at the wall. Pressure outlet conditions have been applied at the exit.

2.4. Governing equations
Continuity equation: \( \frac{\partial}{\partial x_i} (\rho u_i) = 0 \)  

Momentum equation: \[
\frac{\partial}{\partial x_j} \left( \rho u_i u_j \right) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ \left( \mu + \mu_t \right) \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] - \frac{2}{3} \left( \mu + \mu_t \right) \frac{\partial u_i}{\partial x_i} \delta_{ij} - \frac{2}{3} \rho k \delta_{ij} + F
\]

Energy equation: \[
\rho C_p u_i \frac{\partial T}{\partial x_i} = \left( k + k_T \right) \frac{\partial}{\partial x_i} \left( \frac{\partial T}{\partial x_i} \right)
\]

Reynolds number: \( Re = \frac{\rho \nu L}{\mu} \)

Nusselt number: \( Nu_d = 0.023 Re^{0.8} pr^{0.4} \)


2.5. Model validation

The numerical computation was first performed in a particular case of a smooth tube to validate the CFD simulation results with predicted theoretical data obtained by Gnielinski correlation [18] for Nusselt number and Petukhov [19] for friction factor. The predicted theoretical values of Nusselt number and friction factor are demonstrated in figure 3(a). The present results reasonably agree well with the available correlations with little variation for Nusselt number and friction factor. The friction factor is decreasing as the Reynolds number increases. Friction factor for experimental, theoretical, smooth tube and rectangular cut twisted tape inserts are shown in the following figure 3(b).

3. Results and discussion

The current study involves the using of rectangular cut twisted tape insert to augment the heat transfer inside a tube. The results obtained from the simulation clearly aligns with our proposal. Several parameters have been checked with varying Reynolds number. Although, heat transfer rate increases with increasing Re, the increase in friction factor is of concern.

Figure 4(a) shows the variation in Nusselt number with Reynolds number. Nusselt number is the ratio of convective to conductive heat transfer in the fluid boundary. The lowest value of Nusselt number recorded is 24.49 for Re=2722.029 for smooth tube. The value increased steadily with increasing Re and the maximum recorded value for smooth tube is 34.7 for Re=4949.144. For the tube with twisted tape insert however, the value is quite large. Here, the minimum recorded value of Nu is 29.95 which steadily increased to a maximum of 40.05 for the raising of Re from 2722.029 to 4949.144.

Figure 4(b) represents the variation of friction factor with Reynolds number. Friction factor vs Re depicts a declining curve. The maximum friction factor is found in case of the tube with twisted tape insert and the value is 0.25 for Re= 2722.029 which declines markedly as the Re increases. A total decrease of 0.08 is observed for increasing the Reynolds number to 4949.144. Friction factor curve for the smooth tube also shows a declining trend. The declination is however, less significant in this case. Friction factor decreases from 0.08 to 0.053 with the increase in Reynolds number from 2722.029 to 4949.144.

Figure 5 shows the relation between the theoretical and simulation friction factor for the smooth tube. Friction factor recorded from the simulation is higher than the theoretical friction factor. And the difference increases with increasing value. Maximum theoretical friction factor obtained is 0.047, whereas simulation results show the maximum value to be 0.08. The minimum value is 0.038 and 0.053 for theory and simulation respectively.
Thermal performance factor (TPF) represents the effectiveness of a heat transfer enhancement method and refers to the ratio of augmented heat transfer to the increased friction. Figure 6 shows that thermal performance factor is higher at lower Reynolds number and the maximum value is 0.837 for Re=2722.029. TPF decreases drastically with Reynolds number and it falls to 0.782 at Re=4949.144. The result suggests that, with increasing Re, the increase in friction factor is much higher compared to the corresponding increase in the heat transfer rate enhancement. As a result, despite the increase in heat transfer, overall effectiveness of the system is actually decreasing.

Temperature distribution inside the tube with insert is shown in figure 7. It is seen that the temperature is higher near the wall of the tube and as we go inside, it falls. The lowest temperature region is the region around the insert. Maximum recorded temperature for v=0.089 m/s is 351.7K whereas for v=0.161 m/s the maximum temperature is 348.7K. Also, it should be noted that the temperature difference between the regions is higher at low velocity, about 48.5K at 0.089 m/s and the difference lowers to 45.6K at 0.161 m/s. This seems logical as we know by now, increased velocity promotes increased mixing and decreases the difference in temperature between regions.

**Figure 4.** (a) Variation of Nusselt number with Reynolds number; (b) Variation of friction factor with Reynolds number.

**Figure 5.** Comparison between simulated and theoretical friction factor (Smooth tube).

**Figure 6.** Thermal performance factor for rectangular cut twisted tape inserts.
Figure 7. (a) Temperature contour for tube with insert (at 0.089 m/s); (b) Temperature contour for tube with insert (at 0.161 m/s).

Turbulence kinetic energy and its distribution also depend largely on velocity. From Fig. 8 the difference can be observed for the tube with insert. From Fig. 8(a), at low velocity (0.089 m/s), the turbulence kinetic energy is much lower, ranging from $7.1 \times 10^{-11}$ to $1.967 \times 10^{-6}$. Maximum energy region is observed around the insert and there is a notable variation in energy between different regions. In contrast, Fig. 8(b) shows a more uniform energy distribution in the tube at velocity of 0.161 m/s. And also, the overall turbulence kinetic energy ranges from $7.561 \times 10^{-10}$ to $2.404 \times 10^{-4}$, which is much higher when compared to Fig. 7(a).

Figure 8. (a) Turbulence kinetic energy for tube with insert (at 0.089 m/s); (b) Turbulence kinetic energy for tube with insert (at 0.161 m/s).
4. Conclusion
Heat transfer and friction factor characteristics of a tube fitted with rectangular cut twisted tape insert and without insert are numerically studied and validated by two performance criterions Gnielinski and Petukhov. Thus, the computational results obtained indicate that the rate of heat transfer and the pressure drop along tube fitted with rectangular cut twisted tape is higher than the plain tube. The insertion of the twisted tape in the flow passage results in the creation of swirl along with the normal flow. This swirl created increased the turbulence, flow length of the fluid flow, time of exposure of the fluid to the heat flux, resulting in augmented rate of heat transfer. As the Reynolds number increases, the swirling disturbance generated by rectangular cut twisted tape intensifies, thus causing higher temperature and velocity gradient near the tube wall. Therefore, both the heat transfer coefficient and the friction factor increase than the plain tube. The following conclusions can be drawn from the results of the present study.

- Heat transfer rates were increased by an average of 1.16 times as compared to the plain tube.
- Convective heat transfer coefficients were increased by an average up to 1.178 times as compared to the plain tube.
- Nusselt numbers were increased by an average up to 1.18 times as compared to the plain tube.
- Friction factors were increased by an average of 2.59 times as compared to the plain tube.

Appendix A. Nomenclature

\( \rho \) Density (kg/m\(^3\))
\( \mu \) Dynamic viscosity (Pa s)
\( k \) Thermal Conductivity (W/m-K)
\( C_p \) Specific heat (J/kg-K)
\( T \) Temperature (K)
\( \alpha \) Thermal diffusivity (m\(^2\)/s)
\( v \) Inlet velocity (m/s)
\( L \) Length of tube (m)
\( w \) Twisted tape width at large end

Re Reynolds number
\( Pr \) Prandtl number
Nu Nusselt number

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