Numerical simulation of turbulent flow and heat transfer in round tubes equipped with multi-channel twisted tapes

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Abstract
This paper presents the numerical study of the turbulent flow and heat transfer in the round tubes equipped with multi-channel twisted tapes (MC-TT). A finite volume method with the RNG k-ε turbulence model was applied for the simulation. The effects of the twist ratio (defined as a ratio of twist length to tape width: y/w = 1.5, 2.0, 3.0, and 4.0) and number of channels (N = 2 and 4) on the fluid flow and heat transfer characteristics were investigated in a turbulent flow regime (5000 ≤ Re ≤ 15000). The computations show that the circular tubes with multi-channel twisted tapes give higher heat transfer rate than the plain tube by around 22-30%. Heat transfer enhancement by the multi-channel twisted tapes is strongly dependent on twist ratio (y/w) and number of channel (N). For the present study, the tape with N = 2 with twist ratio, y/w = 3.0 offers the best heat transfer enhancement with the maximum thermal performance factor of 1.02.

Keywords: Twisted tape, Heat transfer, Thermal performance, Friction factor, Turbulent flow

1. Introduction

Heat transfer enhancement techniques have been developed and widely utilized in several engineering applications, such as refrigeration, automotive, process industry, aviation and spacecraft engineering, power engineering, chemical, petroleum refining and food industries, etc. Common goals of heat transfer augmentation are to reduce the weight and size of heat exchanger and to upgrade the capacity of an existing heat exchanger. Heat transfer enhancement techniques (HTE) can be divided into two categories: (1) active heat transfer enhancement techniques, which requires an external power, for example, surface-fluid vibration, injection and suction of the fluid, jet impingement, and electrostatic fields, and (2) passive heat transfer enhancement techniques generally involve geometrical modifications such as surface modifications, swirl generators and turbulence promoters. These techniques do not require any direct input of external power.

Among the passive heat transfer enhancement techniques, using twisted tape inserts is one of the most popular methods due to its low cost and also the ease of the insert fabrication and installation. Use of this method causes the swirl in the bulk of the fluids, promotes the fluid mixing between core and wall regions, disturbs the thermal boundary layer and thus enhances heat transfer coefficient in existing system (Eiamsa-ard et al. (2009), Eiamsa-ard et al. (2010), Kongkaitpaiboon et al. (2010), Eiamsa-ard (2010)). The heat transfer enhancement by twisted tape inserts has been extensively reported. Manglik and Bergles (1993a, 1993b) employed twisted tapes for heat transfer enhancement in laminar and turbulent flow regimes under uniform wall temperature condition. They reported experimental data along with the predicted correlations of Nusselt number and friction factor. Saha et al., (1998) compared the heat transfer enhancement by regularly spaced twisted tape inserts with that by continuous twisted tape inserts. Their results showed that both heat transfer rate and pressure drop cause by using the spaced twisted tape inserts are lower than those cause by using the continuous twisted tape ones. Jaisankar et al., (2009) conducted the comparative study for the heat transfer, friction factor and thermal performance of thermosyphon solar water heater system fitted with twisted tape fitted with rod and twisted tape having spacer at the trailing edge. It was found that the twisted tape fitted with rod gives better overall performance than the one having spacer at the trailing edge. Gue et al., (2011) studied the heat transfer and
friction factor characteristics of laminar flow in a circular tube fitted with center-cleared twisted tape for Reynolds numbers between 500 and 1750. The thermal performance factor of tube with center-cleared twisted tape can be enhanced up to 20% as compared with that of the tube without twisted tape. Zhang et al. (2012) numerically studied the heat transfer and flow characteristics of the laminar flow in a tube with multiple regularly spaced twisted tapes. Their results showed that the uses of triple and quadruple twisted tapes respectively enhance Nusselt number up to 171% and 182%, as compared to that of the plain tube. Nanan et al. (2013) studied the effect of twisted tapes with co- and counter-arrangements on heat transfer enhancement. The experimental results suggested that the tapes with counter arrangement (inducing counter-swirl flows) yield higher Nusselt number and cause higher friction factor than the one with co-arrangement. Salman et al. (2013a, 2013b, 2013c) investigated the heat transfer of circular tube fitted with various modified twisted tape geometries (V-cut, quadrant-cut and baffled twisted tape inserts) in laminar flow.

The above literature review indicates that heat transfer enhancement by using twisted tape inserts is strongly dependent on the geometry and arrangement of the inserts. The purpose of the present study is to predict the heat transfer, friction, flow structure and thermal performance characteristics of water flow in the circular tubes equipped with multi-channel twisted tapes. The effects of twist ratio \((\psi/w = 1.5, 2.0, 3.0, \text{ and } 4.0)\) and number of channel \((N = 2 \text{ and } 4)\) are examined. In addition, the contour plots of predicted streamlines, velocity vector, temperature field and local Nusselt number distribution are also given for a better understanding of heat transfer mechanism.

2. Multi-channel Twisted Tape Inserts

The configurations and dimensions of the circular tubes tube fitted (no gap) equipped multi-channel twisted tapes with are illustrated in Fig. 1. The thermo-physical properties of fluid and materials of the tape inserts and circular tubes are given in Table 1.

![Fig. 1. The model of a circular tube fitted (no gap) with different multi-channel twisted tapes.](image)

Table 1 Thermo-physical properties of fluid and materials.

<table>
<thead>
<tr>
<th>Fluid/Materials</th>
<th>(\rho \text{ [kg/m}^3)</th>
<th>(c_p \text{ [J/(kg·K)]}</th>
<th>k \text{ [W/m-K]}</th>
<th>(\mu \text{ [kg/m-s]})</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>998.2</td>
<td>4182</td>
<td>0.6</td>
<td>0.001003</td>
</tr>
<tr>
<td>Copper (tubes)</td>
<td>8978</td>
<td>381</td>
<td>387.6</td>
<td>-</td>
</tr>
</tbody>
</table>

3. Mathematical model and numerical method

3.1 Governing equation and numerical method

The 3-D numerical simulations of fluid flow and heat transfer phenomena of the tubes equipped with multi-channel twisted tapes were carried out. The available finite differential procedures for swirling flows and boundary layer were employed to solve the governing partial differential equations under the following assumptions: 1) steady 3-D fluid flow and heat transfer; 2) the flow is turbulent and incompressible; 3) constant fluid properties; and 4) natural convection and thermal radiation are neglected. Based on above approximations, the following governing differential equations are applied for the simulation.

\[
\frac{\partial}{\partial x_i} (\rho u_i) = 0
\]  

(1)
Momentum equation:

\[
\frac{\partial}{\partial x_j} \left( \rho u_i u_j \right) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial u_k}{\partial x_k} \right) \right] + \frac{\partial}{\partial x_j} \left( -\rho u_i u_j \right)
\]

where the Reynolds stresses, \(-\rho u_i u_j\), to the mean velocity gradients:

\[
-\rho u_i u_j = \mu_i + \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \delta_{ij} \frac{\partial u_k}{\partial x_k}
\]

Energy equation:

\[
\frac{\partial}{\partial x_j} \left[ \rho (p + E) \right] = \frac{\partial}{\partial x_j} \left( k_{\text{eff}} \frac{\partial T}{\partial x_j} \right)
\]

\[
E = h - \frac{P}{\rho} + \frac{u^2}{2}
\]

The turbulent viscosity term, \( \mu_t \), is to be computed from an appropriate turbulent model defined as \( \mu_t = \rho C_{\mu} k^2 / \epsilon \)

The model equation of the turbulent kinetic energy (TKE), \( k \), is written as

\[
\frac{\partial}{\partial t} (\rho k) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + G_k - \rho \epsilon
\]

Similarly the dissipation rate, \( \epsilon \), is given by the following equation:

\[
\frac{\partial}{\partial t} (\rho \epsilon) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_\epsilon} \right) \frac{\partial \epsilon}{\partial x_j} \right] + C_{\text{ic}} \frac{\epsilon}{k} G_k - C_{2\text{e}} \rho \frac{\epsilon^2}{k}
\]

where \( \rho \epsilon \) is its destruction rate while \( G_k \) is the rate of TKE generation which can be expressed as

\[
G_k = -\frac{\partial u_i u_j}{\partial x_i} \frac{\partial u_j}{\partial x_j}
\]

The boundary values for the turbulent quantities near a tube wall are specified with the enhanced wall treatment method. The following constants: \( C_\mu = 0.09 \), \( C_{\text{ic}} = 1.44 \), \( C_{2\text{e}} = 1.92 \), \( \sigma_k = 1.0 \), and \( \sigma_\epsilon = 1.3 \) are employed in the turbulent transport equations.

Renormalized Group (RNG) \( k-\epsilon \) turbulent model

\[
\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho k \epsilon)}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \alpha_k \mu_{\text{eff}} \frac{\partial k}{\partial x_j} \right) + G_k + G_h - \rho \epsilon - Y_M
\]

\[
\frac{\partial (\rho \epsilon)}{\partial t} + \frac{\partial (\rho \epsilon \epsilon)}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \alpha_\epsilon \mu_{\text{eff}} \frac{\partial \epsilon}{\partial x_j} \right) + C_{\text{ic}} \frac{\epsilon}{k} (G_k + C_{3\epsilon} G_h) - C_{2\epsilon} \rho \frac{\epsilon^2}{k} - R_\epsilon + S_\epsilon
\]

In above equations, \( \alpha_k \) and \( \alpha_\epsilon \) are the inverse effective Prandtl numbers for \( k \) and \( \epsilon \), respectively. The effective viscosity \( \mu_{\text{eff}} \) is written as

\[
\mu_{\text{eff}} = \mu + \mu_t = \mu + \rho C_{\mu} \frac{k^2}{\epsilon}
\]

The governing equations are solved by using a finite volume approach and the SIMPLE algorithm. The time-independent incompressible Navier-Stokes equations and the turbulent model are discretized using the finite volume
method. The standard pressure and QUICK (Quadratic Upstream Interpolation for Convective Kinetics differencing scheme) for momentum and energy equations are employed in the numerical model. The pressure-velocity coupling is handled by the SIMPLE (Semi Implicit Method for Pressure-Linked Equations). The solutions are considered to be converged when the residuals of the continuity and momentum equations are less than $10^{-5}$ and that of the energy equation is less than $10^{-9}$.

The evaluation of heat transfer enhancement involves the following dimensionless parameters: Reynolds number, friction factor, Nusselt number and thermal performance factor.

Reynolds number \((Re)\) can be expressed as
\[
Re = \frac{\mu ud}{\rho}
\]  
(12)

Friction factor \((f)\) is computed using the following equation:
\[
f = \frac{2d\Delta P}{\rho L v^2}
\]  
(13)

Nusselt number \((Nu)\) can be obtained from Eq. (14)
\[
Nu = \frac{hd}{k}
\]  
(14)

Average Nusselt number \((Nu_{avg})\) can be obtained from Eq. (15)
\[
Nu_{avg} = \frac{1}{L} \int \! Nu(x) \, dx
\]  
(15)

Thermal performance factor \((\eta)\) is defined as the ratio of the Nusselt number ratio to the friction factor ratio for the uses of the tube with twisted tape and the plain tube \((p)\) at an identical pumping power:
\[
\eta = \left( \frac{Nu/Nu_p}{f/f_p} \right)^{\frac{3}{2}}
\]  
(16)

3.2 Boundary conditions

The computations were carried out with the following assumptions: (1) the physical properties of water remain constant at average bulk temperature, (2) the tube wall is under constant temperature condition at 310 K, (3) the multi-channel twisted tape is subjected to an adiabatic wall condition (high thermal resistance) and (4) mass flow rate of water with 300 K \((Pr = 5.86)\) remains constant.

3.3 Grid independence

A grid independence procedure is implemented by using the Richardson extrapolation technique over grids with different cell numbers. The tetrahedral grid is used for meshing, shown in Fig. 2. Dense grids are applied for the computation near tube wall and in the vicinity of the twisted tape. The grid independence test was carried out using the RNGk-ε turbulent model with grid numbers between 243,347 and 559,611 cells for multi-channel twisted tapes at twist ratio of \(y/w = 1.5\) and Reynolds number of 5000. It was found that the further increase of grid number beyond 243,347 cells results in variation in Nusselt number of less than 1%, thus the grid number of 243,347 cells was taken as criterion for grid independence. Details of all studied cases are given in table 2 below.

![Fig. 2. The mesh of a circular tube with multi-channel twisted tapes.](image)
Table 2 Details of computational conditions in the studied cases for the grid independence test.

<table>
<thead>
<tr>
<th>Case study</th>
<th>Plain tube</th>
<th>N = 2</th>
<th>N = 4</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>y/w = 1.5</td>
<td>y/w = 2.0</td>
<td>y/w = 3.0</td>
</tr>
<tr>
<td></td>
<td>y/w = 1.5</td>
<td>y/w = 2.0</td>
<td>y/w = 3.0</td>
</tr>
<tr>
<td>Dimensional</td>
<td>3-Dimensional</td>
<td></td>
<td></td>
</tr>
<tr>
<td>No. of grid</td>
<td>232,256</td>
<td>243,347</td>
<td>306,841</td>
</tr>
<tr>
<td>Method</td>
<td>Finite volume</td>
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<tr>
<td>Model</td>
<td>RNG k-ε turbulent model with QUICK scheme</td>
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<td></td>
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<tr>
<td>Fluid</td>
<td>Water</td>
<td></td>
<td></td>
</tr>
<tr>
<td>T_in</td>
<td>300 K</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Re</td>
<td>5000 to 15,000</td>
<td></td>
<td></td>
</tr>
<tr>
<td>conditions</td>
<td>Periodic flow and constant wall temperature (310 K)</td>
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<td></td>
</tr>
</tbody>
</table>

4. Result and discussion

4.1 Verification of the numerical method

Prior to the simulation for the tubes with twisted tapes, the set up and numerical method were validated by comparing the present numerical results (Nusselt number and friction factor) of the plain tube with the data of the plain tube obtained from the standard correlations which are available in the open literature by Incropera et al. (2006). The validation was performed for Reynolds numbers (Re) from 5,000 to 15,000. Figure 3 shows that the deviations of the present numerical Nusselt numbers from Gnielinski correlation and Dittus and Boelter correlation were within ±12% and ±6.5%, respectively and friction factors deviate from those obtained from Blasius correlation and Petukhov correlation within ±11.5% and ±9%, respectively. Figure 4 demonstrates that the deviations of the numerical Nusselt numbers and friction factor of the circular tube with twisted tape from Manglik and Bergles correlation was within ±17% and ±17%, respectively. The comparisons indicate that the numerical data are sufficiently accurate and the present numerical method is reliable.

![Fig. 3 Variation of Nusselt number and friction factor with Reynolds number for the present plain tube and correlations.](image-url)
4.2 Flow structure, temperature field and local Nusselt number distribution

The predicted results of the flow structure of the swirling flow, temperature field and local Nusselt number distribution of the plain tube (PT) and tube fitted with multi-channel twisted tape (MC-TT) inserts at Reynolds number of 5000 are shown in Figs. 5-7. The stream line of fluid flow of tube with MC-TT at N = 2 and 4 for y/w = 1.5 and 3.0 are shown in Fig. 5. Evidently, axial flow is found in the plain tube while swirl flows are detected in the tubes MC-TT inserts. Swirl flow intensity becomes stronger as twist ratio decreases, leading to more efficient interruption of thermal boundary layer along the flow path. Figure 6(a) shows the vector plots of velocity that predicted for plain tube equipped with MC-TT. Apparently, MC-TT with N = 2 and N = 4, induce two and four longitudinal vortices, respectively. In addition, the MC-TT with y/w = 1.5 provide higher number of vectors than the one with y/w = 3.0. These longitudinal vortices play a critical role of enhancing fluid mixing between core and tube wall regions and consequently disturbing a thermal boundary layer. The local distributions of the Nusselt number are shown in Fig. 6(b). The results reveal that the tubes with MC-TTs possess higher Nusselt number than the plain tube. Nusselt number increases with decreasing twist ratio (y/w) and increasing number of channels (N). Figure 7 shows the contour plots of temperature fields in the plain tube and the tubes equipped with MC-TTs, at constant Re = 5000 and different axial locations (x/D). Obviously, temperature fields in the tubes with MC-TTs are more disorder than that in the plain tube, due to a stronger turbulent intensity and better fluid mixing. The temperature distribution becomes more significant and thermal boundary layer becomes thinner as number of channels increases.
4.3 Heat transfer

The variations of the Nusselt number ($Nu$) and Nusselt number ratio ($Nu/Nu_p$) with Reynolds number ($Re$) for all studied cases are shown in Fig. 8a and b, respectively. Generally, Nusselt number increases with increasing Reynolds number. However, Nusselt number ratio decreases with increasing Reynolds number. This is caused the dominant increase of Nusselt number in the plain tube over those in the tubes with twisted tape inserts. At the same Reynolds number, Nusselt number increases with decreasing twist ratio ($y/w$) attributed to a stronger turbulent intensity and thus a better fluid mixing. At the similar conditions, Nusselt number increases with increasing number of channels ($N$), as more vortices give more consistent tangential velocity component on the tube wall.
4.4 Friction loss

The variations of the friction factor ($f$) and friction factor ratio ($f/f_P$) with Reynolds number ($Re$) for all studied cases are shown in Fig. 9a and b, respectively. For all cases, both friction factor ($f$) and friction factor ratio ($f/f_P$) decrease with increasing Reynolds number. For the tubes with MC-TTs, friction factor gradually increases with decreasing twist ratio ($y/w$) and increasing number of channels (N). The higher friction loss is mainly caused by the increased interface between fluid and solid part (tape surface), higher swirl intensity and larger flow blockage. For the studied range, the tube equipped with MC-TT at $N = 4$ and $y/w = 1.5$ give the maximum friction factor, corresponding to the maximum friction factor ratio of 11.59.

4.5 Thermal performance factor

Figure 10 demonstrates the thermal performance factors ($\eta$) of the tubes equipped with MC-TTs. Thermal performance factor tends to decrease with increasing Reynolds number, for all cases. For the tubes with MC-TTs, thermal performance factor increases with decreasing number of channels (N). Thermal performance factors offered by MC-TT with $N = 2$ and $N = 4$ are found to be from 0.68 to 1.02 and from 0.64 to 0.89, respectively. In addition, the factor tends to when twist ratio increases from 1.5 to 3.0. However, the reversed trend is found when twist ratio further increases from 3.0 to 4.0. For the range investigated, the maximum thermal performance factor of 1.02 is obtained by using the tube with MC-TT with $N = 2$ and $y/w = 3.0$ at the lowest Reynolds number of 5000.
5. Conclusions

The heat transfer, friction factor and thermal performance factor characteristics of turbulent flow in the tubes with multi-channel twisted tapes (MC-TT) inserts at four different twist ratio ($y/w = 1.5, 2.0, 3.0,$ and $4.0$), two different numbers of channels ($N = 2$ and $4$) were numerically investigated for Reynolds number ranging from $5000$ to $15000$. The conclusions can be drawn from the results of the present study.

- The numerical results for a plain tube are in good agreement with the standard correlations which are available in the open literature, indicated by small discrepancies (within ±12% for Nusselt number and ±11.5% for friction factor). The numerical results for circular tube with twisted tape inserted are in good agreement with the Manglik and Bergles correlations, indicated by small discrepancies (within ±17% for both Nusselt number and friction factor).

- Thermal performance factor increases with decreasing number of channels ($N$). The factor tends to when twist ratio increases from $1.5-3.0$. However, the reversed trend is found when twist ratio further increases from $3.0-4.0$.

- For the range investigated, the maximum thermal performance factor of $1.02$ is obtained by using the tube with MC-TT with $N = 2$ and $y/w = 3.0$ at the lowest Reynolds number of $5000$. The thermal performance of the tube with multi-channel twisted tape at $Re > 5000$ is lower than unity due to the prominent effect of very high friction losses which are up to around $3.8-12$ times of the plain tube.

References


Jaisankar, S., Radhakrishnan, T.K. and Sheeba, K.N., Experimental studies on heat transfer and friction factor characteristics of thermosyphon solar water heater system fitted with spacer at the trailing edge of twisted tapes,


Nomenclature

- $c_p$: specific heat of fluid, J/kg-K
- $d$: diameter of tube, m
- $f$: friction factor
- $G$: rate of TKE generation
- $h$: heat transfer coefficient, W/m$^2$-K
- $k$: thermal conductivity of fluid, W/m-K or turbulent kinetic energy, J/kg
- $Nu$: Nusselt number
- $Nu_{avg}$: average Nusselt number
- $P$: pressure, Pa
- $\Delta P$: pressure drop, Pa
- $Re$: Reynolds number
- $u$: velocity, m/s
- $W$: tape width, m
- $y$: pitch length of twisted tape, m
- $y/W$: twist ratio

Greek Symbols

- $\rho$: fluid density, kg/m$^3$
- $\varepsilon$: dissipation rate, m$^2$/s$^3$
- $\mu$: fluid dynamic viscosity, kg/s-m
- $\eta$: thermal performance factor

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Subscripts

$P$ plain tube

i, j, k direction

t turbulent

Abbreviations

MC-TT Multi-channel twisted tapes

N number of channel

TKE turbulent kinetic energy

RNG Renormalized Group