The effect of off-center placement of twisted tape on flow and heat transfer characteristics in a circular tube

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This study is conducted to investigate the effect of off-center placement of twisted tape on flow distribution and heat transfer in a circular tube. The effect of tape width of 20, 18, 16, 14 and 12 mm on the heat transfer performance is discussed under the same twist ratio of 2.0. The numerical analysis of the flow field, average Nusselt number, friction factor and thermo-hydraulic performance parameter of the tube are discussed with Reynolds number ranged from 2600 to 8760. The results indicate that the Nusselt number of the tube fitted with center-placed twisted tapes at various width is 7–51% higher than the plain tube, and performance in low Reynolds region was found more effective than that in high Reynolds region. The heat transfer for circular tube with twisted tape attached to the wall shows better performance than that for the tube with center-placed twisted tape. With a smaller tape width, a higher increasing ratio of Nu-wall/Nu-center is obtained. The increasing ratio for Nusselt number ranged from 3 to 18%. However, the use of twisted tape inserts is not beneficial for energy saving. The thermo-hydraulic performance parameters for convective heat transfer of helium gas flowing in a circular tube are below unity for the calculated Reynolds region.

Abbreviations

cp: Specific heat of helium gas (J/(kg K))
D: Diameter of the circular tube (m)
f: Friction factor of the tube
g: Gravitational acceleration (m/s²)
H: Twisted pitch (180º) (m)
h: Heat transfer coefficient (W/(m² K))
kₜ: Thermal conductivity of helium gas (W/(m K))
L: Length of the tube (m)
Nu: Nusselt number, Nu = D/λ
p: Static pressure (Pa)
Pr: Prandt number
q: Heat flux (W/m²)
Reₘ: Reynolds number based on swirl velocity, UₛLₛ/ν
s: Twist ratio, s = H/W
T: Temperature (K)
Tₛ: Wall temperature of the tube (K)
T_out: Outlet temperature of the helium (K)
t: Time (s)
uᵣ: Average inlet velocity (m/s)
u: X Component velocity (m/s)
v: Y Component velocity (m/s)
W: Width of twisted tape (m)
w: Z component velocity (m/s)
Z: Coordinate along the length of the tube (m)
α: Thermal diffusivity (m²/s)

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Various modified twisted tapes have been studied and some was proved to show better heat transfer, friction factor and thermal enhancement factor than plain twisted tapes. Murugensan et al. 13 carried out experimental study for a double pipe heat exchanger equipped with square-cut twisted tapes and plain twisted tapes. The results showed that better performance of heat transfer, friction factor and thermal enhancement factor was obtained by using square-cut twisted tape, which is due to the higher generation of additional disturbance and secondary flow at vicinity of tube wall. Kumar et al. 14 studied the spiral plate heat exchanger by computational analysis method. The flow pattern and heat transfer are analysed. Chandrasekar et al. 17 studied the heat transfer and pressure drop in double helically coiled tube heat exchanger with nanofluids, the effect of volume concentration of the nanofluid was discussed. Liu et al. 18 numerically studied the heat transfer enhancement characteristics of coaxial cross twisted tapes for tube flows in laminar region. The Nusselt number of the tube inserted with coaxial cross double twisted tape (CCDTT) and coaxial cross triple twisted tape (CCTTT) were compared to that inserted with traditional twisted tape (TTT). The result shows that with CCTTT as insert, the Nusselt number is 1.51–1.95 times of that of the tube equipped with TTT, and is 1–27% higher than that of the CCDTT for Re ranged from 40 to 1050. Abolarin et al. 19 investigated the heat transfer and pressure drop in a circular tube with alternating clockwise and counter clockwise twisted tape inserts. The Reynolds number ranged from 300 to 11,404 covering laminar, transitional and turbulent flow regimes. The effect of connection angle was discussed and empirical correlations were developed.

The premise of most researches mentioned above is that the width of the twisted tape is equal or almost equal to the tube diameter and the twisted tape is placed along the center axis of the tube. To facilitate assembly, disassembly and eliminating dirt, twisted tapes are usually designed with a width smaller than the tube diameter.
Some researchers have studied the effect of width of the twisted tape and some even tried to place several twisted tapes with small width in the tube. Al-Fahed and Chakroun\textsuperscript{20} experimentally studied the clearance between the tube and equipped twisted tapes and it is concluded that the heat transfer increases with the decrease of the clearance. Bhuiya et al.\textsuperscript{21} studied the heat transfer augmentation in a circular tube with perforated double counter twisted tape inserts. According to this research, the heat transfer rate and friction factor were about 80–290% and 111–335% higher than those of the plain tube, respectively.

So far in the literature, twisted plates are generally designed to be placed along the center axis of the circular tube and is coaxial with the tube. However, in actual operating conditions, when the width of twisted tape is smaller than the tube diameter, twisted tape is not precisely maintained at the centre axis of the tube. Therefore, it is worthwhile studying the offset effect of twisted tape, which could serve as a guideline to the design and usage of this kind of heat transfer equipment.

In the present study, we attempt to figure out the effect of off-center placement of twisted tape on flow distribution and heat transfer in a circular tube. Twisted tapes with different width are offset to contact with the tube wall. The flow field, average Nusselt number, friction factor and thermo-hydraulic performance parameter of the tube are discussed with different placement of twisted tape. The effect of heat transfer enhancement induced by twisted tape inserts is discussed.

**Geometric structure and simulation model**

**Physical model.** A full dimension schematic diagram was shown in Fig. 1. The width (W) of the twisted tape is 12 mm. The twisted ratio (s = H/W) is set as 2.0. The inner diameter (D) of the tube is 20 mm and the thickness 1 mm. The total length of the tube is 500 mm. The geometries of the circular tube with center-placed twisted tape and off-center placed (attach to the tube wall) twisted tape are shown in Fig. 2a,b, respectively. Twisted tapes with thickness of 0.1 mm and width (W) of 20, 18, 16, 14 and 12 mm are used as inserts for the circular tube, and thus the relative 180° twisted pitches are 40, 36, 32, 28 and 24 mm, respectively.

**Governing equations and boundary conditions.** Helium gas is selected as the working fluid with incompressible assumption. The thermo-physical properties of helium gas, such as thermal conductivity, density, and dynamic viscosity are dependent on temperature, as shown in Table 1. The flow inside the tube is assumed to be steady and natural convection is neglected. The problem is considered as a three-dimensional
turbulent heat transfer problem in steady state. Heat conduction in the twisted tape is neglected. The governing equations of continuity, momentum and energy for helium gas are given below in a tensor form.

\[ \frac{\partial u_i}{\partial x_i} = 0 \]  
\[ \frac{\partial (u_i u_j)}{\partial x_j} = -\frac{1}{\rho} \frac{\partial p}{\partial x_i} + \nu \frac{\partial^2 u_i}{\partial x_j^2} - \frac{\partial}{\partial x_j} (\bar{u}_i \bar{u}_j) \]  
\[ \frac{\partial}{\partial x_i} (\rho u_i T) = \frac{\partial}{\partial x_j} \left[ \left( \frac{k}{\lambda_p} + \frac{\mu_t}{Pr} \right) \frac{\partial T}{\partial x_j} \right] \]

The Reynolds number \((Re)\), the Reynolds number considering the helically velocity \((Re_s)\), the convective heat transfer coefficient of the tube flow \((h)\), the Nusselt number \((Nu)\), and the friction factor \((f)\) are defined as follows:

\[ Re = \frac{\rho u_c D}{\mu} \]
\[ Re_s = \frac{\rho u_s D}{\mu} \]
\[ u_s = u_c \left[ 1 + \left( \frac{\pi}{2} \right)^2 \right]^{0.5} \]
\[ h = \rho D u_c C_p (T_{in} - T_{out}) \]
\[ Nu = \frac{h D}{k_f} \]
\[ f = \frac{\Delta P}{\left( \frac{\rho u_c^2}{2} \right) (L/D)} \]

where \(D\) is the inner diameter of the tube, m.

The average velocity of inlet varies from 4 to 12 m/s, with corresponding Reynolds number ranged from 2600 to 8760. Taking the swirl effect into consideration, the swirl Reynolds number \((Re_s)\) ranged from 3300 to 11,140. Therefore, the flow is in turbulent zone. In this study, the value of \(Re\) is used for the results and discussions. Reynolds averaged Navier–Stokes (RANS) turbulence models of RNG \(k-\varepsilon\) Model (RNG) as well as the Reynolds Stress Model (RSM) are adopted with enhanced wall treatment for near wall modelling, and the simulation results are compared in this study. The exit condition is set to pressure outlet, \(P = 500\) kPa.

Non-slip boundary conditions are used for the twisted tape surface and the inner wall of the channel. The boundary condition between helium gas and the surface of twisted tape is fluid–structure interaction. Constant heat flux boundary condition is loaded on the outer surface of the tube. The CFD software ANSYS (version 14.0) is used for numerical computation based on the finite volume method. The SIMPLE algorithm was used and the convergence criterion is set \(10^{-4}\) for residuals of the continuity equation, and \(10^{-6}\) for the momentum and energy equations.

Boundary conditions:

at the twisted tape surface,

\[ u = v = w = 0 \]  
\[ T_s|_{solid} = T_s|_{fluid} \]  

Table 1. Thermo-physical properties of helium gas.22

| Fluid      | Property                                      | Temperature range (K) | Value or correlation                                      |
|------------|----------------------------------------------|------------------------|------------------------------------------------------------|
| Helium gas | Thermal conductivity, \(k\) (W/m K)           | 255.6–500              | \(3.94 \times 10^{-3} T - 1.98 \times 10^{-5} T^2 + 3.19 \times 10^{-8} T^3 - 9.77 \times 10^{-10} T^4\) |
|            | Density, \(\rho\) (kg/m³)                    | 300–500                | \(P/RgT\)                                                  |
|            | Specific heat, \(c_p\) (J/kg K)              | 300–500                | 5197                                                       |
|            | Thermal conductivity, \(k\) (W/m K)          | 300–500                | \(1.034 \times 10^{-1} + 2.58 \times 10^{-4} T\)          |
|            | Dynamic viscosity, \(\mu\) (Pa s)            | 300–500                | \(1.307 \times 10^{-3} + 3.319 \times 10^{-4} T\)        |
|            | Prandtl number, Pr                           | 300–500                | 0.68                                                       |

Table 1. Thermo-physical properties of helium gas.22
at inner wall of the channel,

\[ u = v = w = 0 \]  \hspace{1cm} (11)

\[ -\lambda \frac{\partial T}{\partial z}\bigg|_{z=0} = q \]  \hspace{1cm} (12)

where, \( \lambda \) (W/(m K)) is the thermal conductivity.

**Grid independent analysis.** A grid independent test was performed before the numerical analysis. Hexahedral dominant grid is used for the mesh and the grid in the region near the tube wall and the twisted tape surface are highly refined to ensure the mesh quality for boundary layer resolution. The stabilization of the calculated results of the surface temperature of the tube wall \( (T_w) \), the outlet temperature \( (T_{out}) \) and the friction factor \( (f) \) are taken as evaluation parameters for the grid independence. Four grid systems with different average grid size are used to calculate the same case. The twisted tape with width of 20 mm is placed coaxial with the circular tube. The twist ratio is 2.0 and the relative 180° twisted pitch is 40 mm. The inner diameter of the tube is 12 mm. Cross sections were cut for the tube and twisted tape with a twist pitch of 24 mm, and the axial start position is 0.288 m from the inlet. Figure 5 presents five cross sectional velocity fields in the tube for the single twist pitch at various twisting angles: (a) for the center-placed case and (b) for the case that twisted tape is attached to the wall surface.

**Verification of numerical models.** In our previous experimental research\(^5\), heat transfer coefficients for helium gas flowing over a twisted tape was obtained. The width of the tape is 4 mm and the relative 180° twisted pitch is 20 mm with corresponding twist ratio of 5. The twisted tape was put along the center axis of the tube which has inner diameter of 20 mm. The inlet velocity of helium gas ranged from 4 to 10 m/s and the inlet pressure is maintained at 500 kPa. A numerical simulation case was built with the same conditions as the experiment to validate the flow and heat transfer process over the twisted plate. Turbulence models of RNG and the RSM are used, and the simulation results for \( Nu \) was compared with experimental data at various Reynolds numbers. As shown in Fig. 3, deviation between the numerical results and experimental data is very limited. The maximum deviation for the lowest Reynolds number, and is of 6.4% for the RSM model and 7.6% for the RNG model, respectively. This demonstrates that the accuracy of the numerical models is acceptable, and the RSM model shows better performance. Therefore, the RSM turbulence model is applied for the following simulations.

The above validation was for the Nusselt number of the twisted tape which is put in a circular tube. Since heat transfer over the twisted tape is strongly associated with the flow field around the twisted tape, the accuracy of heat transfer prediction can also reflect the accuracy of flow field calculation to some extent.

As for the heat transfer of the circular tube, numerical results of the \( Nu \) and \( f \) calculated for Model-III in “Grid independent analysis” were compared with empirical correlations proposed by Manglik et al.\(^4\), Eiamsaard et al.\(^1\) and Murugesan et al.\(^1\), as shown in Fig. 4. The Nusselt numbers calculated by CFD method matches the correlation proposed by Manglik et al. best, and the overall deviations of the calculated \( Nu \) and \( f \) from the empirical correlations are found to be within 3.8% and 13.4%, respectively. Therefore, the accuracy of the numerical method is validated.

**Flow characteristics**

Figure 5 shows the stream line inside the tube with twisted tape placed in the center and attached to the wall surface at Reynolds number of 2600 corresponding to the inlet velocity of 4 m/s. The width of the twisted tape is 12 mm. Cross sections were cut for the tube and twisted tape with a twist pitch of 24 mm, and the axial start position is 0.288 m from the inlet. Figure 6 presents five cross sectional velocity fields in the tube for the single twist pitch at various twisting angles: (a) for the center-placed case and (b) for the case that twisted tape is attached to the wall surface.

Compared to plain tubes, flow in the tube changes from linear motion to swirl flows around the twisted tape. For the tube with center-placed twisted tape, the swirl flows are generated in the center area of the tube, while the flow direction of the part of fluid near tube wall was not significantly changed, as shown in Fig. 5a. The swirling
Figure 3. Comparison for the calculated Nusselt numbers of twisted tape with experimental data.

Figure 4. Comparison of the calculated $\text{Nu}$ and $f$ with empirical correlations for the tube with twisted inserts.
of the tape leads to a force (perpendicular to the tape surface) on the fluid close to the tape and accelerates the fluid, which results in higher velocities near the tube surface. Besides, the swirl flow contributes to fluid mixing. These two reasons are considered as main factors for the heat transfer enhancement induced by twisted tape inserts. When the twisted tape is attached to the wall surface, the swirl flows are generated near the tube wall, as shown in Fig. 5b. The fluid in the center region with higher velocity is transported to the near wall region by swirl flow, and the fluid with lower velocity in the near wall region close to the twisted tape is guided away from the boundary zone. The essential difference between these two placements is that the swirl flow occurs in high speed region or near wall region with a relative low speed.

As shown in Fig. 6a, high velocity zones are generated on both sides of the tape following the swirl direction. For example, at the twist angle of 0°, the high velocity zones occur at upper right area and lower left area which are in front of the swirl path of the tape. Low velocity zones are generated in the center of both sides of the twisted tape. The velocity distribution suggests that with the insert of twisted tape, part of fluid in the central region with higher velocity is swirled to be closer to the tube wall and low velocity region occurs at the center of the swirl. Besides, high velocity zones are also generated in the clearance between the twisted tape and the tube wall. Due to rotation symmetry, the flow field in the tube with center-placed twisted tape shows similar distribution at various twist angles.

The flow distribution in the tube with twisted tape attached to the wall surface is different with center-placed case. As shown in Fig. 6b, the flow distributions at different twist angles show different characteristics. When rotating the tape around its center axis, a circular shaped flow area will be formed, and we define it as the main swirl flow region. In the main swirl flow region, there generally exists two high velocity zones on each side of the tape following the swirl direction, though one might be weak than another. Besides, for the area out of the main swirl flow region, high velocity zones are generated in a wide area. Therefore, the swirl flow generated near
tube wall promotes fluid mixing in the near wall region, generates a low velocity region in the center of the main swirl flow region on the tape, and forms a high velocity zone out of the main swirl flow region.

**Results and influence analysis of geometric parameters**

The relationship curves between average Nusselt number and Reynolds number for the circular tube with center-plated twisted tape inserts at different tape width is shown in Fig. 7. Owing to the high velocity zones generated close to the near wall zone and the fluid mixing effect induced by the twisted tapes, all Nusselt numbers of the tube with center-placed twisted tape are higher than that of a smooth tube. Generally, the Nusselt numbers increases with the increase of tape width, though the values for width of 18 mm and 20 mm are very close at low Reynolds region. This may be caused by the thermal boundary layer disturb effect of the 18 mm case, for which the clearance between twisted tape and the tube wall is only 1 mm, smaller than the thermal boundary layer thickness. The Nusselt number of the tube fitted with twisted tapes at various width is 7–51% higher than the plain tube. Moreover, higher increase ratio of Nu was obtained at smaller Re.

The Nusselt number for the circular tube with twisted tapes attached to the wall surface at different tape width is shown in Fig. 8. It can be found that the effect of width on Nusselt number is decreased in comparison to the center-placed case. The Nusselt number for the W = 18 mm case (attached to the wall) is larger than the W = 20 mm case. The ratio between the Nusselt number for the “attached to the wall surface” case (Nu-wall) and the Nusselt number for the center-placed case (Nu-center) is shown in Fig. 9. As can be seen that all the values of Nu-wall/Nu-center is larger than unity, which means the heat transfer for the tube with twisted tape attached to the wall is better than that for the tube with center-placed twisted tape. Furthermore, the smaller the width
is, the higher the increasing ratio is obtained. The increasing ratio for Nusselt number ranged from 3 to 18%. Therefore, the heat transfer could be enhanced by off-center placement of the twisted tape.

**Performance evaluation**

The performance of heat transfer enhancement of the twisted tape with different width is presented and the off-center placement effect is discussed. However, as we know that to obtain the enhancement of heat transfer by passive techniques usually leads to the increase of pump power for flow motion. In order to obtain a comprehensive evaluation of the heat transfer tube, the thermo-hydraulic performance parameter ($\varepsilon$) is employed, which is defined as follows:

$$
\varepsilon = \frac{N_u}{N_{u0}} \left(\frac{f}{f_0}\right)^{1/3}
$$

where, $N_{u0}$ is the Nusselt number of the plain tube, and $f_0$ is the friction factor of the plain tube.

The Nusselt number versus Reynolds number and friction factor versus Reynolds number are shown in Figs. 10 and 11. As can be seen, $N_u/N_{u0}$ varies a little with Re, and the $f/f_0$ decreases with the increase of Re. Figure 12 shows the variation of the thermo-hydraulic performance parameter ($\varepsilon$) with Reynolds number for center-placed case and "attaching to the wall" case with different tape width. All the thermo-hydraulic performance parameter is below unity, which means the use of twisted tape inserts for convective heat transfer of helium gas flowing in a circular tube is not advantageous in energy saving in the calculated Reynolds region, though the heat transfer may be enhanced. The circular tube with twisted tape inserts attached to wall surface generally shows better performance than that with center-placed twisted tape.
Conclusions

In the present study, the effect of off-center placement of twisted tape on flow distribution and heat transfer in a circular tube is discussed. The flow field, average Nusselt number, friction factor and thermo-hydraulic performance parameter of the tube are discussed and the effect of heat transfer enhancement induced by twisted tape inserts is analysed with Reynolds number ranged from 2600 to 8760.

The Nusselt number of the tube fitted with center-placed twisted tapes at various width is 7–51% higher than the plain tube and performance in low Reynolds region was found more effective than in high Reynolds region. The heat transfer for circular tube with twisted tape attached to the wall shows better performance than that for the tube with center-placed twisted tape. With a smaller tape width, a higher increasing ratio of Nu-wall/Nu-center is obtained. The increasing ratio for Nusselt number ranged from 3 to 18%.

The use of twisted tape inserts is not beneficial for energy saving. The thermo-hydraulic performance parameters for convective heat transfer of helium gas flowing in a circular tube are below unity for the calculated Reynolds region, which is mainly caused by the small density and heat capacity of gas when compared to liquid working fluids, such as water.

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