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RESEARCH ARTICLE

Thermo-hydraulic performance investigation of a heat exchanger tube inserted with twisted tapes modified with various twist ratio and alternate axis

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Abstract: This paper presents a numerical investigation on a heat transfer enhancement by using various twisted tape inserted into a heat exchanger tube. The twist ratios of the twisted tapes are considered as 4, 6 and 8. Besides, the twisted tapes are modified with alternate axis for each twist period. The alternate axis is provided by connecting the twisted tapes having one twist with 90° connection angle. Water is selected as working fluid and turbulent flow condition corresponding to Reynolds number ranging from 5000 to 29,000 are considered in the study. The heat exchanger tube is under constant heat flux of 50 kW/m^2. Numerical analysis results prove that the use of twisted tape improves thermal performance compared to the smooth tube. As the twist ratio decreases, the heat transfer performance and the friction loss penalty increase. Overall results are determined with thermohydraulic performance criteria (THP). The highest THP value is obtained as 1.21 for the tube inserted with the twisted tape which has the twist ratio of 4 and the alternate axis at Reynolds number of 5913.

Keywords: Heat transfer enhancement, flow characteristic, alternate axis, twisted tape, performance evaluation criteria.

1. Intorduction

The effective use of energy has gained importance with the need for efficient use of energy, in recent. Since the usage of heat exchangers are widely employed in industry, it is extremely important to increase the performance of the heat exchangers. Heat transfer enhancement techniques are divided in two main categories: active and passive techniques. While active techniques require extra energy input to the system, the passive techniques do not demand any extra energy cost. Therefore, the passive techniques are generally preferred for increasing heat transfer due to reasons such as low-cost, ease of use and long-term maintenance [1][2]. Twisted tapes [3], helical screw tapes [4], wire coil [5], ring [6], etc. are inserted into the heat exchanger tube as passive techniques. The twisted tape is widely preffered due to induce effective heat transfer enhancement [7],[8]. However, using twisted tape in the tube increases the pressure drop penalty and thus causes an undesirable increase in the friction factor [9],[10]. For this reason, the researchers paid attention to increase heat transfer as much as possible but not to increase the friction factor too much while designing the twisted tape. In this scope, Li et al. [11] investigated the effect of the centrally hollow narrow twisted tapes on the heat transfer enhancement characteristic. They recommended that the use of the cross hollow twisted tape is more effective under laminar flow conditions rather than turbulent flow conditions. Bhuiya et al. [12] investigated the thermal characteristics of triple twisted tape inserted heat exchanger tube under turbulent flow conditions. The heat transfer performance of their considered cases are found between 1.10 and 1.44 which the highest performance is obtained by the use of triple twisted tape insert. Skullong et al. [13] employed the staggered-winglet perforated tape to reduce the pressure drop penalty and to provide stronger fluid mixing. They found that the highest Nu and friction factor is obtained by the use of winglet perforated tape having largest winglet blockage ratio and lowest pitch length ratio. Dagdevir et al. [14] compared the performance of perforated twisted tape with dimpled twisted tape. They found that although the perforated twisted tapes induce stronger fluid mixing, the dimpled twisted tapes disrupt the boundary layer more effectively. In other research [15], they investigated the effect of the location of perorations and dimples on the twisted tape on the heat transfer and flow characteristics. They stated that

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the perforations/dimples at the edge of the twisted tapes show better heat transfer and hydraulic performance rather than at the center of the twisted tapes. Changing the axis of twisted tape is another technique to enhance heat transfer by promoting the turbulence intensity, increasing swirl flow and disrupting the boundary layer. Ponnada et al. [16] compared the perforated twisted tapes whether they have alternate axis or not on the thermal performance. They found that the thermal performance factor is improved up to 1.433 and 1.396 by the use of perforated twisted tape having alternate axis and not having alternate axis, respectively. Wongcharee and Eaimsa-ard [17] carried out an experimental study on heat transfer enhancement by using twisted tape with alternate axis and CuO/water nanofluid. They found the maximum thermal performance factor of 5.53 by using the twisted tape with alternate axis, nanofluid volume fraction of 0.7% and at Reynold number of 1990. Wongcharee and Eiamsa-ard [18] investigated the effect of the twisted tape inserts with alternate axis on the friction factor and the heat transfer characteristic under laminar flow conditions. The twist ratio of the twisted tapes is considered as 3.0, 4.0 and 5.0 in their study. They found that the twist ratio of 3.0 provides the best heat transfer enhancement performance. Abolarin et al. [19] investigated that the effect of the alternating clockwise and counter clockwise twisted tape inserts on heat transfer and pressure drop characteristics under transitional flow regime. They considered the twist ratio of the twisted tapes of 5 and 12, and connection angels of 0°, 30° and 60°. They concluded that the increase in connection angle and decrease in twist ratio show better thermal performance.

According to the literature review, the twisted tape inserts having alternate axis promises good heat transfer enhancement performance. However, it is seen from the literature review that the effect of the twisted tape insert having alternate axis on the heat transfer and hydraulic performance have not been investigated with consideration of wide range twist ratios, connection angle of 90° and under turbulent flow conditions, yet. With this motivation, the twisted tape inserts which have alternate axis, connection angle of 90°, twist ratios of 4.0, 6.0 and 8.0, are numerically investigated under turbulent flow conditions.

2. Numerical Methodology

Ansys, Fluent 18.0 which solve the governing equations by finite element method is used to perform the numerical analyses for the present study. The SIMPLE (Semi-Implicit Method for Pressure-Linked Equation) algorithm is used to solve the governing equations by coupling the pressure and velocity gradients. Convergence criteria of the solution are selected as 10^{-4} for continuity, velocities, k and epsilon and 10^{-6} for energy.

The governing equations which are conservation of continuity, momentum and energy are given in Eqs. (1) to Eq. (3), respectively.

Mass conservation equation:

$$\nabla(\rho\vec{V}) = 0 \tag{1}$$

Momentum conservation equation:

$$\nabla(\rho\vec{V}\vec{V}) = -\nabla P + \nabla(\mu\nabla\vec{V}) \tag{2}$$

Energy conservation equation:

$$\nabla(\rho c_p\vec{V}T) = \nabla(k\nabla T) \tag{3}$$

Differential equations belong to the k-epsilon RNG turbulent model which is selected to simulate the turbulent flow through the tube are given in the following [20]:

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}(\alpha_k \mu_{eff}\frac{\partial k}{\partial x_j}) + G_k + G_b - \rho\epsilon - Y_m + S_k \tag{4}$$

$$\frac{\partial}{\partial t}(\rho \epsilon) + \frac{\partial}{\partial x_i}(\rho \epsilon u_i) = \frac{\partial}{\partial x_j}(\alpha_\epsilon \mu_{eff}\frac{\partial \epsilon}{\partial x_j}) + C_{1\epsilon}\frac{\epsilon}{k}(G_k + C_{3\epsilon}G_b) - C_{2\epsilon}\rho\frac{\epsilon^2}{k} - R_\epsilon + S_\epsilon \tag{5}$$

2.1. Solution domain and boundary conditions

Solution domain adopted for the present study is schematically depicted in Fig. 1. Twisted tapes are inserted into a test section having length of 1 m. A fluid development section is placed before the test section to provide fully developed flow conditions. The length of the fluid development section is specified as 250 mm, which provides higher than ten times inner diameter [21]. In order to avoid from the revers flow effect, an exit section with length of 150 mm is placed after the test section. Inner diameters of entire sections are same and dimension of 17 mm. The boundary conditions of the fluid development section and the exit section are determined as no-slip and adiabatic, while that of the test section is no slip and constant heat flux of 50 kW/m^2. The velocity magnitude is calculated with considered Reynolds number ranging from 5000 to 35,000. Gauge pressure at the outlet is selected as 0 Pa, which provides the atmospheric conditions. Thermophysical properties of the working fluid is considered as temperature dependent. The thermo-physical properties of the working fluid are imported to the software by correlations with a form of Eq. (6) which their coefficients are given in Table 1. Temperature is in K unit in the correlations.

$$f(T) = a + bT + cT^2 + dT^3 \tag{6}$$

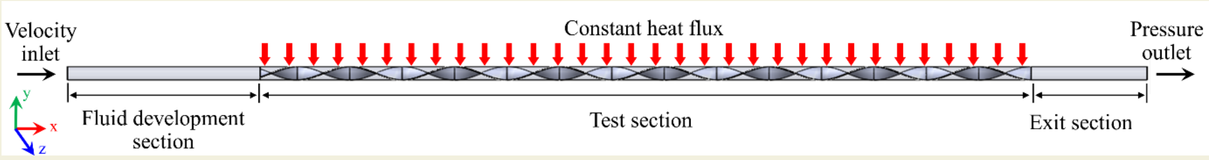

Figure 1. Solution domain and boundary types adopted in the present study

Table 1. Coefficients of the correlations belongs to the thermo-physical properties of the working fluid

Thermo-physical property	a	b	c	d
Density, ϱ [kg/m³]	719.22	2.29643	-5.15E-03	1.91E-06
Specific heat capacity, Cp [j/kgK]	6217.4	-15.2458	3.52E-02	-2.33E-05
Thermal conductivity, k [W/mK]	-1.33612	1.29E-02	-2.67E-05	1.76E-08
Dynamic viscosity, μ [kg/ms]	0.0597994	-4.73471E-04	1.26033E-06	-1.12233E-09

Table 2. Definitions of the considered inserts with diameters

Definition	Pitch length (y) [mm]	Width (w) [mm]	Thickness (t) [mm]	Total length (L) [mm]	Connection angle of alternate axis
C-CC, y/w=4.0	68				
C-CC, y/w=6.0	102	17	1.0	1000	90°
C-CC, y/w=8.0	136				

2.2. Data Reduction

In order to evaluate the heat transfer and hydraulic performance of the considered tube configurations, Nusselt number (*Nu*) and friction factor (*f*) is used. Data used to calculate the *Nu* and the *f* are computed from Fluent by using surface integrals area-weighted averagely. Reynolds number (*Re*) which defines the characteristic of the flow is given in Eq. 7. Density of the fluid, mean velocity magnitude of the flow, inner diameter of the tube and dynamic viscosity of the fluid are denoted by ρ, U, Di and μ, respectively.

$$Re = \frac{\rho U D_i}{\mu} \tag{7}$$

The Nu is calculated with Eq. (8). In this equation, h and k represent the convective heat transfer coefficient (Eq. (9)) and thermal conductivity of the fluid. Constant heat flux and wall temperature of wall and bulk temperature of the fluid are denoted with q'', T_w and T_b.

$$Nu = \frac{hD_i}{k} \tag{8}$$

$$h = \frac{q''}{T_w - T_b} \tag{9}$$

The f is calculated with Eq. (10). In this equation, ΔP and L represent the pressure drop penalty and length of the test tube.

$$f = \frac{\Delta P}{\frac{1}{2}\rho U^2 \frac{L}{D_i}} \tag{10}$$

2.3. Inserts

The twisted tapes are configured with different twist ratios (y/w) of 4.0, 6.0 and 8.0, alternate axis (connection angle of 90°), in the present study. The considered twisted tape inserts are designed to have three different twist pitch lengths (y) of 68mm, 102 mm and 136 mm, thickness (t) of 1 mm, width (w) of 17 mm and total length (L) of 1000 mm, respectively. Table 2 summarizes the dimension of the twisted tape configurations. The illustrations of the twisted tapes considered in the present study are given in Figure 2.

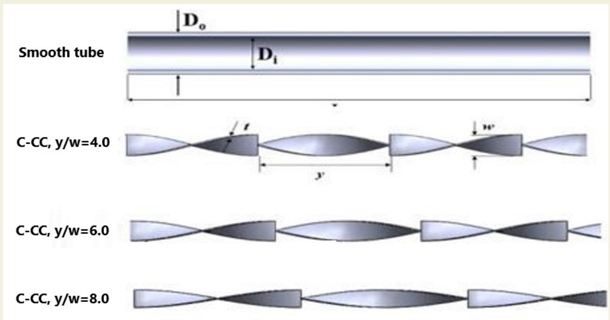

Figure 2. Considered tube configurations

3. Results and Discussions

3.1. Validation of the numerical methodology

Before the validation study, a gird independence study is performed to check a possible effect of the number of the elements on the results. A polyhedral grid structure having 2,955,760 (Fig. 3) is selected since the differences on the *Nu* and the *f* results are not higher than ±2.5%.

In order to ensure the accuracy of the numerical methodology, a validation study is conducted by comparing the

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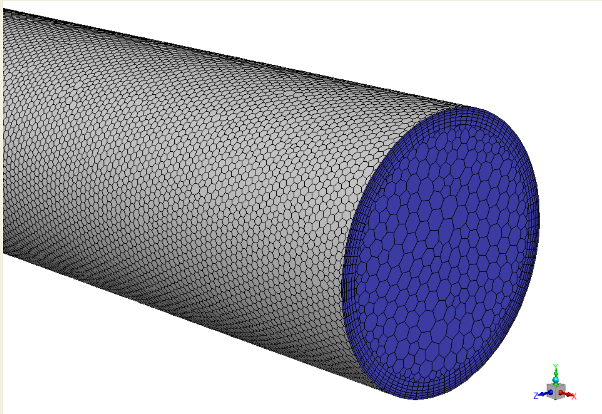

Figure 3. Grid structure used in the present study

results of the smooth tube with the correlations which are widely used in the literature. The used correlations are Dittus & Boelter Eq. (11) [22] and Blasius Eq. (12) [23]: for the Nu and the f, respectively.

$$Nu = 0.023Re_D^{0.8}Pr^{0.4} \tag{11}$$

$$f = 0.316Re_D^{-0.25} \tag{12}$$

The numerical results are compared with the literature in Fig. 4. As seen from this figure, a good agreement is obtained with the correlations.

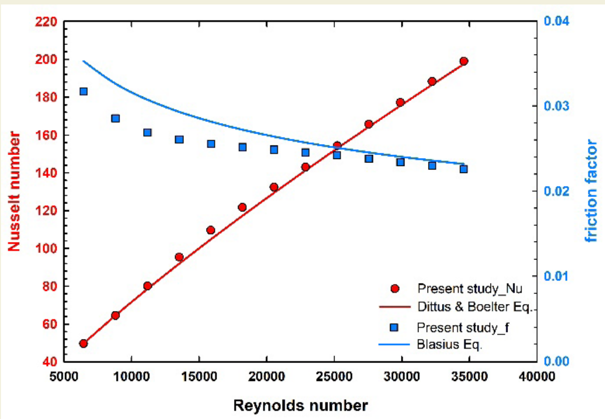

Figure 4. Comparison of the results of the *Nu* and the *f* belongs to the smooth tube with literature

3.2. Heat transfer performance

Fig. 5 shows the Nu number results versus Reynolds number for the considered tube configurations in the present study. It is concluded that Nu number increases with increasing Re number for all configurations. A higher Nu number was obtained when the tube configurations with twisted tape were compared to the smooth tube. In addition, with the increase in twist ratio of twisted tapes, there was a decrease in the thermal performance provided. The reason for this can be explained as the pipe with the twisted tape inserted approaching the smooth tube with the decrease in the twist ratio. In addition, with the increase of the twist ratio, the turbulence intensified and

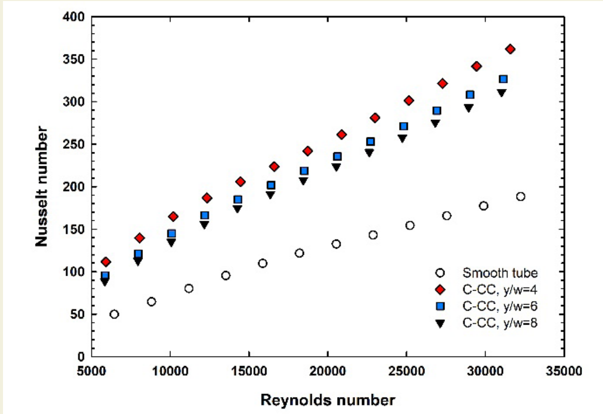

Figure 5. Variation of the Nu number according to the Reynolds number

the boundary layer was destroyed more effectively. The highest Nu value was obtained as 361.65 at Re 31,585 for the twisted tape configuration with a twist ratio of 4.0.

The temperature contours of the considered cases at Reynolds number of 15,000 are shown in Fig. 6a. It is seen from the figure that; the inserts effectively disrupt the thermal boundary layer and fluid having high temperature near the inner wall are mixed with the cold one at the center of the tube. It is a prove for the improvement of the heat transfer by using the twisted tape. When the effect of the twist ratios is examined, as the twist ratio decreases, the thermal boundary layer thickness decreases. Thus, decreasing the twist ratio has a positive role on heat transfer enhancement.

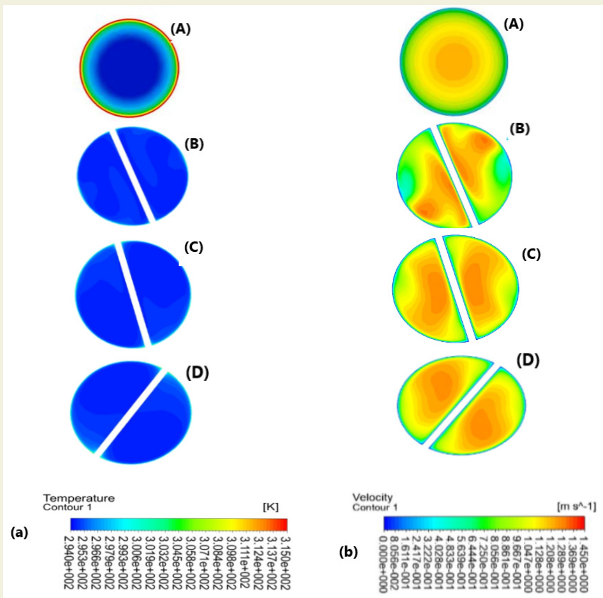

Figure 6. Temperature (a) and velocity (b) contours investigated tubes which are A) Smooth tube, B) C-CC_y/w=4, C) C-CC_y/w=6, D) C-CC_y/w=8 at Reynolds number of 15000, respectively.

3.3. Hydraulic Performance

The distributions of the friction factor depending on Re for all investigated configurations are given Fig. 7. As expected, friction factor decreases with increasing Re number in all configurations. In the case of using twisted tape, the fluid in the tube encounters obstacle surfaces which causes increasing to friction factor. In addition, the increase in the twist ratio caused an increase in the friction factor. The highest friction factor value was obtained in Re 5913 as 0.2365 in the twisted tape configuration with a twist ratio of 4.

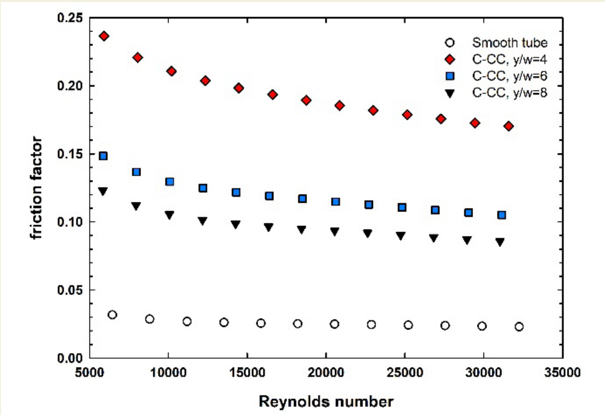

Figure 7. Change in friction factor depending on Re

The increase in the friction factor is directly related with the increase in pressure drop through the tube. As the twist ratio of the twisted tapes decreases, the pressure drop increases. Fig. 6b shows the velocity contours of the considered cases to observed the velocity distribution at the cross-section of the tube. An increase in the velocity gradient at the center of the tube attributes to the increase pressure drop. In addition to inserts increase the velocity gradient at the center of the tube, the decrease in the twist ratio is another parameter to increase the velocity gradient.

The velocity stream line contours obtained from the numerical analyzes are shown in Figure 8. When the stream lines of smooth tube and the tube with twisted tape are

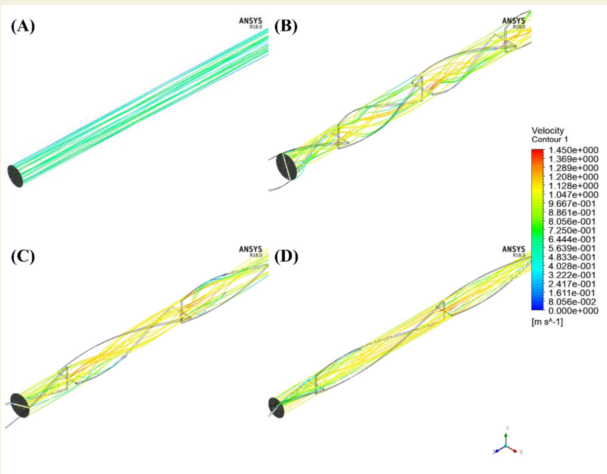

Figure 8. Velocity stream line contours investigated tubes which are A) Smooth tube, B) C-CC_y/w=4, C) C-CC_y/w=6, D) C-CC_y/w=8 at Reynolds number of 15000, respectively

examined. It is seen that the flow in the smooth tube takes place axially, while the flow in the tube with twisted tape is swirling. The twisted tape placed in the tube ensures that the flow takes place by showing a swirl movement and ensures better mixing of the flow in the regions near the inner wall of the tube. Convection heat transfer occurs better in the regions near the inner wall of the tube, and in this case the placed twisted tapes increase heat transfer. In addition, when the contours are examined, it is seen that there is an increasing frequency of change in the flow direction with the decrement of the twist ratio, and alternate axis. This situation increases the turbulence intensity through the flow which has a positive effect on heat transfer enhancement.

3.4. Thermohydraulic Performance

Thermohydraulic performance criteria (THP) which is given in Eq. (13) is used to evaluate a heat exchanger system in terms of both thermal and hydraulic. The case where the THP value is 1 refers to the smooth tube. If the THP value is greater than 1, the method to be applied to the system will have a positive effect on the thermohydraulic performance of the system.

$$THP = \left(\frac{Nu_{TT}}{Nu_S}\right) / \left(\frac{f_{TT}}{f_S}\right)^{1/3} \tag{13}$$

The distribution of THP values based on Re are given in Figure 9. It can be resulted that the heat transfer enhancement results are more dominant than the friction loss results. The highest THP value of 1.21 is obtained by the case of y/w=4.0 at Re of 5913.

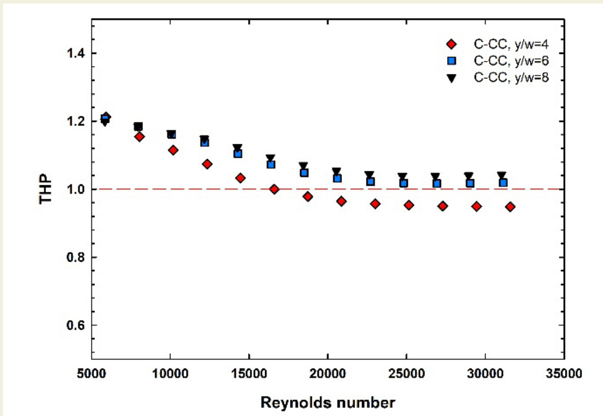

Figure 9. Thermohydraulic performance versus Reynolds number for all tube configurations examined

4. Conclusion

The performance of heat exchanger tube equipped with twisted tape inserts having alternate axis and various twist ratios are numerically investigated on the heat transfer and flow characteristics. Main conclusions obtained from the results are draw in below.

1. Decrease in the twist ratio generates more turbulent in-

tensity through the tube which induce the heat transfer enhancement.

2. The twisted tape with alternate axis allow to effectively mixing fluid through the tube. Thus, heat transfer is positively effective by this phenomenon. The highest Nusselt number of 361.65 is obtained by the case of twisted tape having the twist ratio of 4.0 at Re 31,585.

3. Decrease in the twist ratio and the alternate axis cause to the fluid encounter with larger obstacle surface which increase pressure drop penalty and friction factor. The highest friction factor of 0.2365 is observed in the case of twisted tape having the twist ratio of 4.0 at Reynolds number of 5913.

4. The highest THP value is obtained as 1.21 for the tube inserted with the twisted tape which has the twist ratio of 4 and alternate axis at Reynolds number of 5913.

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Nomenclature

Abbreviations

C_p	specific heat capacity, (J/kgK)
C-CC	clockwise-counterclockwise alternate axis twisted tape
D	diameter, (mm)
h	convective heat transfer coefficient, (W/m^{2}K)
f	friction factor, (-)
k	thermal conductivity, (W/mK)
L	total length, (mm)
Nu	Nusselt number, (-)
Pr	Prandtl number, (-)
q''	heat flux, (kW/m^2)
Re	Reynolds number, (-)
THP	Thermohydraulic performance, (-)
t	thickness, (mm)
U	average velocity, (m/s)
w	width of the twisted tape (mm)
y	pitch length of the twisted tape (mm)
ΔP	drop pressure, (Pa)

Greek symbols

ρ	density, (kg/m^3)
μ	dynamic viscosity, (kg/ms)

Subscripts

b	bulk
i	inner
s	smooth tube
TT	tube with twisted tape
w	wall

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