Thermal and swirl flow topologies in a twisted square duct with a multi-twisted tape installed

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Abstract
This paper presents 3D numerical investigation of the turbulent flow and heat transfer characteristics of a twisted square duct installed with multi-twisted tapes. Air was used as the working fluid with flow rates in terms of Reynolds numbers ranging from 3000 to 20,000. The effects of (1) multi-twisted tape width ratios (w/H) of 0.2 to 1.0 and (2) the number of channels (N = 2 and 4) on heat transfer and flow mechanisms were studied at constant twist ratio of y/D = 3.5. The numerical results showed that twisted square duct combined with twisted tape caused swirl flows which effectively promoted fluid mixing and provided heat transfer over those of both a straight smooth square duct and twisted square duct. Increasing w/H led to increases in both heat transfer and the friction factor. At a given multi-twisted tape width ratio (w/H), the heat transfer and friction factor with N = 4 were higher than those with N = 2, while thermal enhancement factor showed the opposite trend. A maximum thermal enhancement factor of 1.93 was obtained at tape width ratio of w/H = 1.0, channel number of N = 2 and Re = 3000.

Keywords: Heat transfer, Friction factor, Multi-twisted tapes, Thermal enhancement factor, Twisted square duct

Nomenclature

A heat transfer area, /m²
A_c cross section area, /m²
D characteristic diameter of a twisted duct, /m
D_h hydraulic diameter, /m
f friction factor
H cross-sectional length, m
h convective heat transfer coefficient, /W m⁻² K⁻¹
h_l local convective heat transfer coefficient, /W m⁻² K⁻¹
k turbulent kinetic energy
k_a air thermal conductivity of air, /W m⁻¹ K⁻¹
N number of channel flows
Nu average Nusselt number
Nu_l local Nusselt number
P wetted perimeter of the cross-section, /m
p static pressure, /Pa
Pr Prandtl number
Pr_t turbulent Prandtl number
q_conv. convection heat transfer, W m⁻²
q_input heat flux input, W m⁻²
1. Introduction

Several heat transfer enhancement techniques have been devised for more compact heat exchangers. Increasing fluid mixing using turbulators is one important approach. The main purpose of a turbulator is to generate vortex/swirl flow leading to increased convective heat transfer or heat transfer enhancement (Eiamsa-ard et al., 2013; Saysroy and Eiamsa-ard 2017). Twisted tapes, one type of turbulator, are extensively applied to increase heat transfer in a tube heat exchanger. A swirl flow induced by twisted tapes effectively disrupts thermal boundary layers, and thus facilitating heat transfer. Chokphoemphun et al. (2015) reported the thermal performance of a tubular heat exchanger with multiple twisted-tape inserts in a turbulent flow regime. The multiple twisted-tape inserts were fixed at a twist ratio of $y/w = 4$. They found that heat transfer increases incrementally with both the Reynolds number and the number of twisted-tapes. The heat transfer and friction factor were found to increase by 2.12 and 4.1 times, respectively, over those of a plain tube. The maximum thermal performance factor 1.2 was found with the use of multi-twisted tapes, $N = 6$ and a twist ratio of $y/w = 2.5$. Eiamsa-ard et al. (2010) investigated the effect of dual twisted tapes on counter/co-swirling flows and heat transfer enhancement in a round tube. Dual counter twisted tapes (CTs) were used as counter-swirl flow generators, while dual co-twisted tapes (CoTs) were used as co-swirl flow generators in a test section. CTs and CoTs with four different twist ratios ($y/w = 2.5, 3.0, 3.5$ and $4.0$) were applied at Reynolds numbers ranging between 3700 and 21,000. They found that the Nusselt number, friction factor and thermal enhancement index increased with decreasing twist ratio ($y/w$). The heat transfer rates in the tube fitted with CTs were increased up to 44.5% and 50% higher than those with CoTs and single twisted tape, respectively. The maximum thermal enhancement factor was found at 1.39 using CTs with a twist ratio of $y/w = 2.5$. Hong et al. (2018) studied the effect of short length helical tapes on heat transfer enhancement. Tape length ratios ($L/D$) of 31-46.6, hole diameter ratios ($W/D$) of 0.28-0.41 and pitch length ratios ($P/D$) of 1.03-1.7 were studied. They found that heat transfer and friction factor were higher than those of the plain tube by
where the maximum augmentation of the Nusselt number and friction factor were found by using the spirally-coiled twisted-ducks with twist-pitches, \( \tau_p \), of 0.05-0.15 m and coil-pitches, \( \tau_c \), of 0.015-0.035 m were employed. An increase of the Nusselt number and friction factor was found at a twist ratio of 2.5. Castelain et al. (2001) studied of chaotic advection regime in a twisted duct flow. Khoshvaght-Aliabadi et al. (2018) investigated the heat transfer enhancement in heat exchange devices with spirally-coiled twisted-ducks using Cu/water nanofluids with concentrations of 0.5% and 1% by mass as the working fluids. A spirally-coiled twisted-duct with twist-pitches, \( \tau_p \), of 0.05-0.15 m and coil-pitches, \( \tau_c \), of 0.015-0.035 m were employed. The maximum augmentation of the Nusselt number and friction factor were found by using the spirally-coiled twisted-ducks with \( \tau_p = 0.05 \text{ m and } \tau_c = 0.015 \text{ m} \). Kongkaitpaiboon et al. (2019) used corrugated tubes with various numbers of starts and pitch ratios. They found that a corrugated tube generated swirl flow which improved fluid mixing and augmented heat transfer rate. The resultant Nusselt numbers and thermal performance factors were respectively increased up to 2.16 and 1.2 times of those of the straight circular tube. Promvonge (2015) applied twisted tapes combined with a V-fin turbulator in a square-duct heat exchanger. The effect of the pertinent V-fin parameters, such as the four relative fin height ratios, \( e/w = 0.16-0.42 \) and relative fin pitch of \( p/w = 4-16 \) at a single fin attack angle of \( \alpha = 30^\circ \) on thermal characteristics were investigated. The highest thermal performance of 1.75 was achieved by using the V-fin turbulator with \( e/w = 0.21 \) and \( p/w = 4.0 \). Eiamsa-ard and Changchareon (2015) presented the flow structure and heat transfer performance in a square duct fitted with dual/quadruple twisted-tapes using various configurations in a turbulent region. They found that the highest thermal performance factors for the Co-DTs, C-CDDTs, C-DDTs, C-QT, PC-QT, and CC-QT, were around 1.21, 1.35, 1.25, 1.38, 1.08, 1.18, and 1.22, respectively. Yadav et al. (2015) studied square and hexagonal ducts with twisted tape inserts in laminar flow regime. Their results showed that Nusselt number and friction factor increased as the side of non-circular duct increased. The highest Nusselt number and friction factor were found by using the tube with twisted tapes.

Additionally, twisted tube/ducts were used for heat transfer enhancement. Swirl flow was generated by a twisted tube/duct in a similar fashion as the twisted tapered. Bhadouriya et al. (2015) reported the effect of fluid flow and heat transfer in a twisted square duct with twist ratios between 2.5 and 20. They found that the maximum Nusselt number and friction factor were found at a twist ratio of 2.5.

2. Physical model of a twisted square duct with an installed multi-twisted tape

The systems of interest are twisted square ducts with multi-twisted tapes installed, as depicted in Fig. 1. The twist ratio \( (y/D) \) of twisted square duct and multi-twisted tape are set to 3.5, where \( D \) is the characteristic diameter \( (D = 0.05 \text{ m}) \) of the duct. The duct and tape with twist ratio \( (y/D) \) of 3.5 were chosen due to their promising thermal enhancement factor \( (TEF) \). The parameters \( w \) and \( H \) are the multi-twisted tape width and cross-section length, respectively, and the parameter \( N \) is the number of flow channels. In general, a typical twisted tape has \( N = 2 \) (Castelain et al., 2001, Eiamsa-ard and Changchareon, 2015, Hong et al., 2018). However, heat transfer rate and the pressure loss increase with increasing the number of flow channels \( (N) \). In order to keep pressure loss low, the tapes with and \( N = 2 \) and 4 were selected in the present work. The simulation encompassed (1) two \( (N = 2) \) and four channel \( (N = 4) \) multi-twisted tapes with width ratios.
(w/H) ranging from 0.2 to 1.0, and (2) Reynolds numbers ranging from 3000 to 20,000. The grid used in the computational domain consisted of hexahedron elements. For the viscous sublayer near the walls, where the momentum and temperature significantly change, the hexahedron grids were applied with higher density than those in other regions. The high-density grids at the near-wall regions were set to $y^+ ≈ 1$.

3. Boundary condition

The inlet and outlet of the flow domain were applied with periodic boundaries. For the periodic boundary of the fluid flow, a mass flow rate is applied as the initial condition for the iteration by using periodic technique. The outlet condition was used for repeating calculation until the velocity profiles at the outlet and inlet are similar. For the periodic boundary of the heat transfer, an inlet fluid temperature of 300 K is applied as the initial condition for the iteration by using periodic technique under constant wall heat flux. The outlet condition was used for repeating calculation until the difference between of inlet and outlet temperature becomes constant. The physical properties of the air have been assumed to remain constant at average bulk temperature. Impermeable boundary and no-slip wall conditions have been implemented over the twisted square duct walls as well as the multi-twisted tape. The tape was treated as an insulator. A heat flux at the twisted square duct wall was maintained constant at 600 W/m².

4. Data reduction

The assumptions of the numerical model for fluid flow and heat transfer in a twisted square duct with inserted multi-twisted tapes are:
- Flow that is steady, incompressible and turbulent
- Three-dimensional fluid flow and heat transfer
- The fluid properties are constant
- Negligible radiation heat transfer

Based on the above assumptions, the tube flow is governed by the continuity, Navier-Stokes and energy equations. In the Cartesian tensor system, these equations can be written as follows:

Continuity equation:
$$\frac{\partial}{\partial x_i}(u_i) = 0$$  \hspace{1cm} (1)

Momentum equation:
$$\frac{\partial}{\partial x_j} \left( \rho u_i u_j \right) = - \frac{\partial p}{\partial x_i} + \mu \left[ \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right] - \rho u' u'_j$$  \hspace{1cm} (2)

where $\rho$ is the fluid density fluid, and $u_i$ is a mean component of velocity in the $x_i$ direction, $p$ is the pressure, $\mu$ is the dynamic viscosity, and $u'$ is a fluctuating component of velocity. Repeated indices indicate summation from one to three for 3D problems.
Energy equation:

\[
\frac{\partial}{\partial x_j} (\rho u_j T) = \frac{\partial}{\partial x_j} \left( \Gamma + \Gamma_t \right) \frac{\partial T}{\partial x_j}
\]

where \( \Gamma \) and \( \Gamma_t \) are molecular and turbulent thermal diffusivity, respectively, and are given by:

\[
\Gamma = \frac{\mu}{Pr}, \quad \Gamma_t = \frac{H_t}{Pr}
\]

The Reynolds-averaged approach to turbulent modeling requires that the Reynolds stresses, \( -\rho u_i u_j' \) in Eq. (2) be modeled. The Boussinesq hypothesis relates the Reynolds stresses to the mean velocity gradients as seen in the following equation:

\[
-\rho u_i u_j' = \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \rho k \delta_{ij} + \frac{\mu_t}{Pr} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \delta_{ij}
\]

where \( k \) is the turbulent kinetic energy, defined by \( k = \frac{1}{2} u_i'^2 \) and \( \delta_{ij} \) is a Kronecker delta. An advantage of the Boussinesq approach is the relatively low computational costs associated with computation of the turbulent viscosity, where \( \mu_t \) is given as \( \mu_t = \frac{\rho C_{\mu} k^2}{\epsilon} \). The realizable \( k-\epsilon \) turbulence model was used for all computational domains in the turbulent region. The details of the governing equations and the realizable \( k-\epsilon \) turbulent model have been presented by Promthaisong and Suwannapan (2018). There are four parameters of interest in the present work, namely, the Reynolds number, friction factor, Nusselt number and thermal performance enhancement factor.

The Reynolds number is defined as:

\[
Re = \frac{\rho u_0 D_h}{\mu}
\]

where \( D_h \) is the hydraulic diameter which can be calculated from \( 4A_c/P \), where \( A_c \) is the cross-section area and \( P \) is the wetted perimeter of the cross-section.

The friction factor can be calculated from:

\[
f = \frac{(-dp/dx)D_h}{(1/2)\rho u_0^2}
\]

The Newton’s law of cooling (Incropera et al., 2006) can be expressed as:

\[
\dot{q}_{\text{conv}} = h(T_s - T_m)
\]

For no-slip condition at the tube surface, the heat transfer from the tube surface to the first layer of the fluid next to the surface takes place solely via conduction under constant heat flux condition. The heat transfer equation can be expressed as:

\[
\dot{q}_{\text{conv}} = \dot{q}_{\text{input}}
\]

where the heat flux (\( \dot{q}_{\text{input}} \)) is kept constant at 600 W/m²

\[
h = \frac{\dot{q}_{\text{input}}}{(T_s - T_m)}
\]

\[
Nu = \frac{hD_h}{k_{\text{air}}}
\]

where \( h \) and \( k_{\text{air}} \) are the convective heat transfer coefficient and thermal conductivity of air, respectively.

The area-average Nusselt number can be obtained from:

\[
Nu = \frac{1}{A} \int Nu_j dA
\]

A measure of performance accounting for both heat transfer and pressure loss for equal pumping power, the thermal enhancement factor (TEF) suggested by Promvonge and Skullong (2020) can be expressed as

\[
TEF = \left( \frac{Nu}{Nu_0} \right) \left( \frac{f_h}{f_h_0} \right)^{1/6}
\]

where \( Nu_0 \) and \( f_h \) are the Nusselt number and friction factor for a straight smooth square duct, respectively.
5. Results and discussion

5.1 Validation

The computational domain was resolved with hexahedron elements and five grid densities of 50,000, 150,000, 350,000, 500,000 and 700,000 nodes were used to investigate the grid independence of the solution. The numerical results showed that variation in the Nusselt number ($\textit{Nu}$) and friction factor ($f$) changes were marginal (< 0.2%) when the number of nodes increased from 500,000 to 700,000. Hence, a grid with 500,000 nodes was adopted for the current computations. The present numerical data (Nusselt numbers and friction factors) of the straight smooth square duct were compared with those by Incropera et al. (2006) and Promvonge et al. (2014) under a similar operating condition, as shown in Fig. 2(a). The present numerical results are in excellent agreement with those reported in both references (deviations less than 5%). The fluid flow and heat transfer results for the straight smooth square duct with twisted tape inserted ($N = 2$, $w/H = 1.0$ and $y/w = 4.0$) by using the Realizable $k$-$\varepsilon$ turbulence model were compared with the experimental data by Promvonge et al. (2014) as presented in Fig. 2(b). Under similar conditions, the numerical result provided the similar trend with the experimental data for both the Nusselt number and friction factor and gave the mean deviations less than 10% for both the Nusselt number and friction factor. Therefore, the Realizable $k$-$\varepsilon$ turbulence model can be applied for the duct inserted with twisted tape in which swirl flow was generated.

Fig. 2 Verification of (a) $\textit{Nu}_0$ and $f_0$ for a straight smooth square duct and (b) straight smooth square duct inserted with twisted tape.

Fig. 3 3D flow structure, velocity vector and x-velocity distribution in the transverse plane for $Re = 3000$
5.2 Flow topology

Figure 3 displays the 3D flow topology, velocity vector and x-velocity distribution in the transverse plane for $Re = 3000$. In a straight smooth square duct, only axial flow was detected. In twisted square ducts, swirl flows were generated near-wall region while axial flow appeared around the core region. The flow structures in the twisted square ducts (for both $N = 2$ and $N = 4$ channels) with the tape having the smallest width ratio of $(w/H = 0.2)$ were similar to that in twisted square ducts without tape. The results indicate that the narrow multi-twisted tape has insignificant effect on the flow structure. As the tape width ratio $(w/H)$ was increased, the effect of swirl flow induced by the multi-twisted tape became significant. The use of the tapes at $w/H = 1.0$ (tight-fit multi-twisted tapes) resulted in fully swirling flow in the ducts with both $N = 2$ and $N = 4$ channels confirmed by the flow vectors in the transverse plane which showed that x-velocity distributed over the whole cross-sectional area. The flow helped in promoting fluid mixing between the core and duct wall region.

5.3. Heat transfer and friction loss

Figure 4 presents the fluid temperature field in 3D (iso-surface) and 2D (on the transverse plane) forms at $Re = 3000$. Evidently, there was a considerable difference between core and wall temperatures, indicating high heat transfer resistance in the straight smooth square duct due to the laminar axial flow. For the twisted square duct alone and the ones with loose-fit multi-twisted tape cases $(w/H = 0.2-0.8)$, the temperature distributions were affected by swirl flows. However, the difference between core and wall temperatures were still significant since the tapes with small widths induced low swirling intensity resulting in poor fluid mixing between the core and near the duct wall for both tubes with $N = 2$ and $N = 4$ channels. On the other hand, the temperature distribution in the duct equipped with the tapes having width ratio $(w/H)$ of 1.0 was fairly uniform, especially the one with $N = 4$ channels. The results revealed that the tape width and the number of ducts are important parameters affecting fluid mixing. However, the effect of tape width is more dominant than that of the channel number of ducts. The local temperature distributions presented in Figure 5 are accorded with the results in Fig. 4. The straight smooth square duct and the one with the tapes having small widths showed high duct wall temperatures, especially around the corners where dead zones existed. As the tape width ratio $(w/H)$ was increased, the sizes of the dead zones were reduced owing to the stronger swirling flow.

![Fluid temperature fields for $Re = 3000$](image)

The local wall Nusselt number distributions for a Reynolds number $(Re)$ of 3000 are displayed in Fig. 6. Obviously, the twisted square duct yielded larger areas with high Nusselt numbers as compared to the straight smooth square duct due to the effect of the swirling flow. For the ducts with twisted tape, Nusselt numbers increased with increasing the tape width ratio $(w/H)$. The ducts with $N = 4$ channels and the tape having $w/H = 1.0$ yielded the highest Nusselt number due to the strong swirling flow.
to the excellent fluid mixing as mentioned above. It should be noted that the duct with the tapes having \( w/H = 0.2 \) and 0.6 (loose-fit multi-twisted tapes) yielded low Nusselt number for both \( N = 2 \) and \( N = 4 \). In addition, this was similar to a twisted square duct alone since the fluid flow was only disrupted at the core by the tape. However, the same \( w/H, N = 4 \) channels gave a higher Nusselt number than for \( N = 2 \), as is shown in Fig. 7. A combination of methods gave a higher Nusselt number than the twisted square duct alone.

Figure 8 presents the variation of the Nusselt number ratio (\( \frac{Nu}{Nu_0} \)) with Reynolds number (\( Re \)). Nusselt number ratio (\( \frac{Nu}{Nu_0} \)) decreased with increasing \( Re \) in all cases. At a given \( Re \), the twisted square duct alone and twisted square ducts with multi-twisted tapes yielded higher heat transfer rates than the straight smooth square duct (\( \frac{Nu}{Nu_0} > 1.0 \)) owing to the generation of swirling flow that helped to promote fluid mixing. In the similar manner, the twisted square ducts with multi-twisted tape yielded higher heat transfer rate than the twisted square duct alone. Furthermore, the ducts with \( N = 4 \)
channels showed better performance in heat transfer enhancement than the ones with $N = 2$ channels. As the tape width ratio ($w/H$) increased, heat transfer increased. Over the studied $Re$ range, the twisted square duct alone gave $Nu/Nu_0$ ranging from 1.28 to 1.76. The ducts with $N = 2$ and $N = 4$ gave $Nu/Nu_0$ ranging from 1.29 to 3.09 and 1.31 to 3.33, respectively which were higher than those of the twisted square duct alone by 0.53-75.47% and 1.98-88.87%, respectively. The twisted ducts with $N = 4$ gave higher $Nu/Nu_0$ than the ones with $N = 2$ by around 1.44-7.63%. The ducts $N = 4$ and containing the tapes having $w/H = 1.0$ yielded the highest $Nu/Nu_0$ of 3.33 at $Re = 3000$.

![Graph](image)

Fig. 7 Local wall Nusselt number profiles for $Re = 3000$

![Graph](image)

Fig. 8 Influences of multi-twisted tape width ratios ($w/H$) and number of tape channels ($N$) on the Nusselt number ratio ($Nu/Nu_0$)
The variation of friction factor ratio ($f/f_0$) with Reynolds number ($Re$) is presented in Fig. 9. Friction factor ratio ($f/f_0$) tended to decrease with increasing $Re$ in all cases. The use of the twisted square duct alone and the twisted square ducts combined with multi-twisted tapes resulted in higher than that found by the use of the straight smooth square duct ($f/f_0 > 1.0$) because of flow blockage by the duct rough surface and twisted tapes. The twisted square ducts caused higher $f/f_0$ than the twisted square duct alone. The increase of $N$ resulted in higher friction loss penalty. Over the Re range studied, the use of the twisted square duct alone showed $f/f_0$ ranging from 1.56 to 1.92. The ducts with $N = 2$ and $N = 4$ caused $f/f_0$ ranging from 1.87 to 4.98 and 2.03 to 6.48, respectively which were higher than those of the twisted square duct alone by 19.8-159% and 29.95-237%, respectively. The twisted ducts with $N = 2$ caused higher $f/f_0$ than the ones with $N = 2$ by around 8.48-32.27%. The ducts $N = 4$ and containing the tapes having $w/H = 1.0$ caused the highest $f/f_0$ of 6.48 at $Re = 3000$.

![Figure 9](image.png)

(a) $N = 2$

(b) $N = 4$

(c) effect of channel number ($N$)

Fig. 9 Influences of multi-twisted tape width ratio ($w/H$) and channel number ($N$) on the friction factor

### 5.4 Performance evaluation

Figure 10 displays variation of the thermal enhancement factor ($TEF$) with Reynolds number ($Re$). Thermal enhancement factor ($TEF$) tended to decrease with increasing $Re$ in all cases. In the range studied, the $TEF$ were in the range of 0.89-1.81 depending on the number of channels ($N$), tape width ratio ($w/H$) and $Re$. The ducts with the tapes having the tape width ratios of 0.2-0.8 gave a lower $TEF$ than the twisted square duct alone due to the prominent effect of increased friction factor over that of enhanced heat transfer. On the other hand the ducts with $N = 2$ and $N = 4$ channels and tapes having for $w/H = 1.0$ yielded higher $TEF$ than the twisted square duct alone because of the dramatic heat transfer enhancement. Although the ducts with $N = 2$ channels yielded lower heat transfer than the one with $N = 4$, they yielded higher $TEF$ the ones with $N = 4$ due to their lower friction loss penalty. In the present study, the maximum $TEF$ of 1.93 was obtained by using the duct with $N = 2$ containing the tapes having $w/H = 1.0$ at $Re = 3,000$. In addition, the results of the straight smooth square duct with multi-twisted tape by Ray and Date (2003), $N = 2$, $w/H = 1.0$ and 0.5, $y/w = 2.5$ are compared with those of the twisted square duct with multi-twisted tape, $N = 2$, $w/H = 1.0$ and 0.5, $y/w = 2.47$ ($y/D = 3.5$) which is the best case in the present work as presented in Fig. 10(a). The comparison revealed that the present case showed higher $TEF$ than the one by Ray and Date by around 60%. This can be explained that the twisted square duct with multi-twisted tape caused better fluid mixing and thus higher the heat transfer than the straight smooth square duct with multi-twisted tape.
6. Conclusions

Twisted square ducts compounded with multi-twisted tapes created swirling flows that helped to increase heat transfer by improving fluid mixing between the core and near-wall regions. The use of twisted square ducts into with multi-twisted tapes resulted higher Nusselt number ratios ($\frac{Nu}{Nu_0}$) and friction factor ratios ($\frac{f}{f_0}$) than that of the twisted square duct alone. The Nusselt number ratio ($\frac{Nu}{Nu_0}$) and friction factor ratio ($\frac{f}{f_0}$) increased with increasing tape width ratio ($w/H$). The ducts with $N = 2$ and $N = 4$ gave $\frac{Nu}{Nu_0}$ ranging from 1.29 to 3.09 and 1.31 to 3.33, respectively which were higher than those of the twisted square duct alone by 0.53-75.47% and 1.98-88.87%, respectively. The ducts with $N = 2$ and $N = 4$ gave $\frac{Nu}{Nu_0}$ ranging from 1.29 to 3.09 and 1.31 to 3.33, respectively which were higher than those of the twisted square duct alone by 0.53-75.47% and 1.98-88.87%, respectively. In the range studied, the $TEF$ were in the range of 0.89-1.81 depending on the number of channels ($N$), tape width ratio ($w/H$) and Reynolds number ($Re$). The maximum $TEF$ of 1.93 was obtained by using the duct with $N = 2$ containing the tapes having $w/H = 1.0$ at $Re = 3,000$.

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