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Research Article

Analysis on enhanced turbulent heat transfer and flow characteristic in a twisted and dimpled oval tube

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Abstract: This work reveals heat transfer and flow characteristic through a twisted and oval tube under turbulent flow (10,000≤Re≤40,000) and constant heat flux of 50 kW/m^2. Numerical analyses were carried out on several twisted and oval tube configurations. In order to clearly reveal the effect of dimples in the swirl flow, the twisted oval tubes with and without dimples are considered. In order to enhance the heat transfer, the tube is dimpled where heat transfer was inefficient. The results indicate that the dimples on the twisted tube significantly increase the heat transfer, although they cause to slightly increase pressure drop. The decrease in the pitch length significantly and negatively affected the hydraulic performance of the tube, while it induced to promote the heat transfer. Furthermore, while the dimples on the twisted oval tube cause to the friction factor increase by 18.0%, they induce the average Nusselt number to increase by 14.8%. The best configuration that is the case of dimpled twisted oval tube having pitch length of 200 mm at Reynolds number of 10,000, yields thermo-hydraulic performance criteria of 1.428.

Keywords: heat transfer enhancement, thermo-hydraulic performance, twisted oval tube, dimpled tube.

1. Introduction

Since the heat transfer is most irreversible process and last step of the energy transformations, the effective use of thermal systems is significantly important. The energy costs have been raised because of the energy shortage and the increase in the population. Therefore, designing effective heat transfer devices has critical role on this issue. Many attentions have been paid on this issue that is called heat transfer augmentation [1]. Active technique and passive technique compose of the heat transfer augmentation techniques. Although the active techniques promise significant heat transfer augmentation result, they are not preferred in industrial field due to extra energy requirement. However, the passive techniques have been widely employed due to good heat transfer augmentation and no any energy requirement [2-4]. Inserting turbulent promoters into the tube, enhancing (extended, treated and roughened) the surface of the tube, disrupting the flow path through the tube and modifications of working fluid (e.g. nanofluid) can be given as examples of the passive heat transfer augmentation techniques [5-9].

The use of a twisted oval tube (TOT) instead of using a smooth tube enhance the heat transfer performance of a heat exchanger [10]. In this way, it is possible to design and produce smaller sized and lighter heat exchanger devices. On the other hand, the twisted surfaces cause to increase pressure drop penalty. Therefore, the performance of the heat exchanger equipped with the twisted oval tube should be evaluated with another parameter that deals with thermo-hydraulic performance criteria (THP).

A few researches have been conducted on the TOTs by taking into account the THP of the heat exchangers. Bishara et al. [11] conducted a heat transfer augmentation technique by using TOT under laminar conditions where at Re<1200, while Tan et al. [12] researched the geometrical parameters of TOTs on the THP under turbulent flow conditions. Yang et al. [13] investigated the influences of twisted elliptical tubes on the flow characteristics and the thermal for whole flow conditions. The common conclusion from these studies is that heat transfer performance is improved, while flow characteristic is coarsened with the increase in aspect ratio and the decrease in twist pitch length. Furthermore, Yang et al. [13] presented Nusselt number and friction factor correlations that are valid in wide Reynolds number ranging from 600 to 55,000 for water fluid. In addition to investigations including TOT having a constant pitch length, Talebi and Lalgani [14] investigated the influence of variable step

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twist for the TOTs in a heat exchanger. They resulted that a TOT having a constant pitch length showed a better thermal performance than a variable pitch. Wu et al. [15] found that rotational flow along the elliptical tube improves the heat transfer.

Unlike the twisted tubes with elliptical cross-section, twisted tubes with different cross-sections were also examined by researchers. Razzaghi et al. [16] numerically investigated the thermal and hydraulic performance of a twisted flat tube. They found that the thermal efficiencies of the considered cases are obtained in the range of 30-60% which varies according to Reynolds number. Li et al. [17] modified the twisted flat tube by shrinking oval shape of the twisted flat tube. They concluded that the heat transfer performance and friction is increased with the decrement of the diameter ratio and pitch ratio of the tube decreases. Tang et al. [18] numerically and experimentally researched the effect of various dimensions of twisted three-lobed tubes on convective heat transfer and fluid flow. They found that the maximum radial and tangential velocity occur in the lobed region leading to strongly fluid mixing and heat transfer augmentation. Cheng et al. [19] numerically investigated the heat transfer and the flow characteristics of twisted tube with various cross-section shapes such as triangle, square, pentagon and hexagon. The highest thermo-hydraulic performance evaluation criteria value of 2.69 is obtained by the case of twisted pentagon tube.

The twisted tubes are also inserted with twisted tape [20] and coiled wires [21], in order to further increase the heat transfer augmentation. Mashayekhi et al. [22] investigated the effect of twisted elliptical tube inserted with twisted tape inserts according to the THP. They stated that combining of twisted tube and twisted tape creates double secondary flow in the center region and adjacent the wall which further improves the heat transfer. Samruaisin et al. [23] investigated the influence of twisted tube inserted with twisted tape on a heat exchanger. They found that secondary flow is created near center of tube and adjacent the tube. The combination of the secondary flows led to promote heat transfer, friction factor and THP with respect to circular plain tube. Eiamsa-ard et al. [24] studied trilobal twisted tube inserted with triple-channel twisted tape on the heat transfer augmentation. They resulted that the highest THP was resulted by the case of twisted tube fitted with three-channel twist tape in core-to-longitudinal combination.

Dimpled tube which is one of the surface modifications applied on the tubes is another passive heat transfer augmentation technique. The dimples through the tube disrupt the boundary layers, inducing swirl flow near the tube wall and causing less pressure drop compared to the inserts [7]. Li et al. [25] numerically and experimentally researched impact of dimples on the heat exchanger tube. They obtained the maximum THP of 1.55 at turbulent flow conditions. Dagdevir [26] optimized the geometrical parameters of the dimpled tube according to both heat transfer and hydraulic performance, simultaneously. Sabir et al. [27] investigated the different shaped dimples such as ellipsoidal, conical and spherical on heat exchanger tube. They revealed the highest THP was obtained by the 45° ellipsoidal dimpled tube. The influence of ellipsoidal dimples through the flow behavior and the heat transfer is numerically investigated by Xie et al. [28]. They stated that causing downward flow by the longitudinal and transverse dimples induces to enhance the heat transfer. Besides, a heat transfer augmentation study by using cross-combined dimpled tube is performed by Zhang et al. [29]. They resulted that the cross-combined dimpled tube provides 24.8% more heat transfer than the single ellipsoidal dimpled tube.

Many heat transfer augmentation studies performed to further improve the THP of the twisted tubes as mentioned in the literature. The works focused on analyzing the aspect ratios and pitch ratios of cross-section shapes the twisted tubes and equipped with some inserts. Flow disruptions near the twisted tube wall improve the heat transfer more effectively than at the core region of the tube. Therefore, surface modifications on the twisted tube promise further heat transfer augmentation. With this motivation, heat transfer augmentation of dimpled and twisted oval tube (DTOT) is studied in this study, which it has not been investigated yet. Main purpose of the work is to reveal effects of flow separation from dimples in addition to the swirl flow by TOT on the flow characteristics and heat transfer. Therefore, this paper presents a clear comparison between TOT and DTOT according to the flow characteristic and the heat transfer.

2. Numerical Methodology

2.1. Solution domain and boundary conditions

Solution domain of dimpled and twisted oval tube (DTOT) is depicted in Fig. 1. The solution domain composes of main three sections: flow development, test and exit sections. The flow development section is placed to provide a developed flow boundary condition, while the exit section is creted to avoid from the reverse flow effects by swirl flow. The hydraulic diameter (D_h) of the tube for all sections is selected as 26.9 mm corresponding to 1 inch (DN 25). The length of the flow development section is determined as 300 mm which corresponds to higher than ten times hydraulic diameter to provide developed flow condition as proposed in a book written by Incropera et al. [30]. The lengths of test section and exit section are selected as 1000 mm and 200 mm, respectively. Designed test section has dimpled and twisted oval tube (TOT) with pitch lengths of 50 mm, 100 mm and 200 mm. Major diameter (A_i) and minor diameter (B_i) of TOT is respectively determined as 41.48 mm and 20.74 mm so that the aspect ratio (AR) of 2.0 and the D_h of 26.9 mm is ensured. Dimples are applied on the TOTs with 10 mm dimple diameter (D_d), and 2 mm dimple height (H_d). Three different pitch lengths (p) of twisted tube are considered as 50, 100 and 200 mm in the present study.

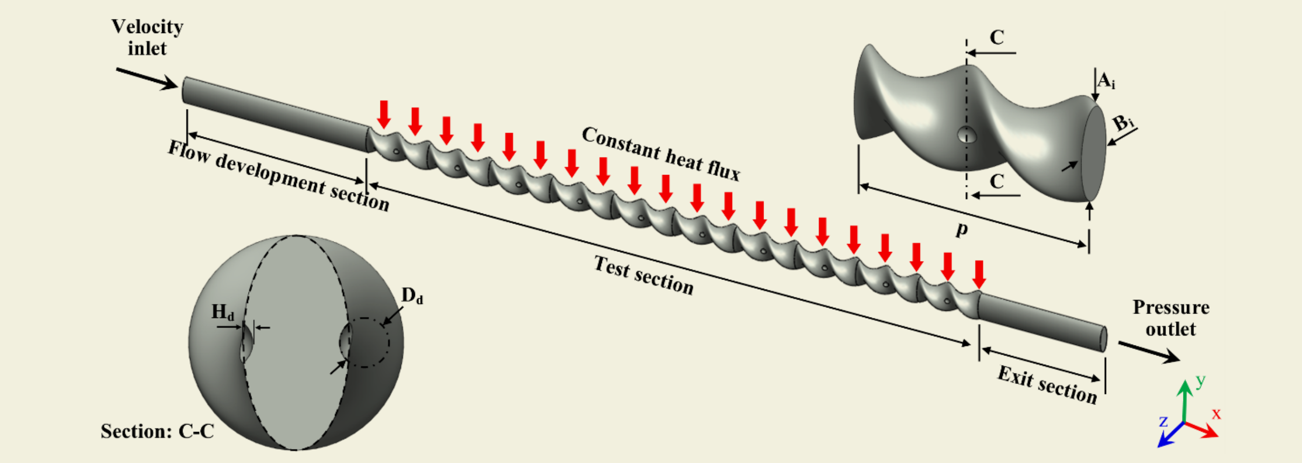

Figure 1. Solution domain adopted for the present study

Wall boundary condition of the flow development section and the exit section are applied as no slip and adiabatic, while that of the test section is applied as constant heat flux of 50 kW/m^2 and no slip. Water is selected as working fluid for the solution domain. The fluid enters to the solution domain with temperature of 293.15 K and inlet velocity of the tube is determined according to Reynolds number ranging from 10,000 to 40,000.

2.2. Governing equations

Numerical analyzes are conducted to examine the flow characteristics and the heat transfer of the TOT and DTOT with some assumptions which are steady state condition, incompressible flow and constant fluid properties. The afore-described solution domain is solved by Ansys Fluent 18.0 with governing the equations which are continuity, momentum and energy equations:

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

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

$$\nabla.\left(\rho\vec{V}c_pT\right) = \nabla.(k\nabla T) \tag{3}$$

where ρ, c_p, k and μ represent the density, specific heat, thermal conductivity and dynamic viscosity of the fluid, respectively. They are considered as constant value where 293.15 K according to the inlet boundary condition ($\rho = 998.0\frac{kg}{m^3}, c_p = 4185\frac{j}{kgK}, k = 0.598\frac{\mathrm{W}}{\mathrm{mK}}$ $and\ \mu = 1.002E-3\ \frac{\mathrm{kg}}{\mathrm{ms}}$) [31]. P, V and T represent pressure, mean velocity and temperature of the fluid, respectively. Ansys Fluent 18.0 is employed to solve the governing equations. SIMPLE is used as scheme for coupling pressure-velocity. The residual criteria for energy equation is considered 10^{-7}, while the residual criteria for other equations are considered 10^{-5}. The governing equations of Realizable k-ε turbulent turbulent model is available in Fluent user guide [32]. Turbulent intensity (I) at inlet is specified according to Eq. (4).

$$I = 0.16Re^{-0.125} \tag{4}$$

2.3. Data reduction

Velocity magnitude of the inlet boundary condition is calculated with Reynolds number (Re) defined in Eq. (5).

$$Re = \frac{\rho D_h V}{\mu} \tag{5}$$

where D_h denotes the hydraulic diameter and is defined in Eq. (6). A denotes the sectional area, while P denotes the perimeter of the oval tube.

$$D_h = \frac{4A}{P} \tag{6}$$

Thermal performance of the cases is evaluated with Nusselt number (Nu) given in Eq. (7).

$$Nu = \frac{hD_h}{k} \tag{7}$$

Convective heat transfer coefficient denoted as h in Eq. (7) is calculated by Eq. (8).

$$h = \frac{q''}{T_w - T_b} \tag{8}$$

Hydraulic performance of cases is evaluated with friction factor (f) given in Eq. (9).

$$f = \frac{\Delta P}{\frac{1}{2}\rho V^2\frac{L}{D_h}} \tag{9}$$

where ΔP and L denotes the pressure drop through the test section and length of the test section, respectively.

Thermo-hydraulic performance criteria (THP) proposed by Webb [33] is defined in Eq. (10), where subscript "0" denotes the circular smooth tube.

$$THP = \frac{(Nu/Nu_0)}{(f/f_0)^{1/3}} \tag{10}$$

3. Results and Discussions

3.1. Mesh independence

A mesh independence study is carried out before running the cases to check whether the results are influenced by the number of cells, or not. Influences of the number of cells on Nu and f results at Re of 20,000 are given in Fig.

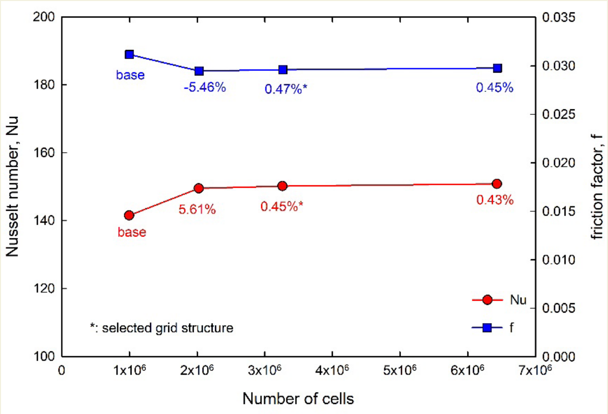

Figure 2. Influence of the number of cells on *Nu* and *f* results.

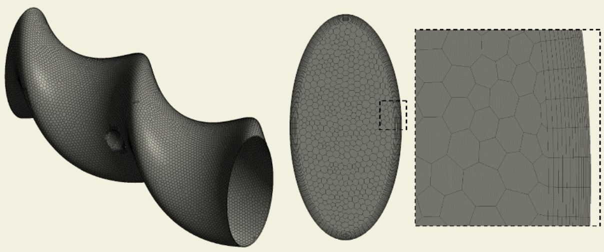

Figure 3. Views of the mesh structure used for the study

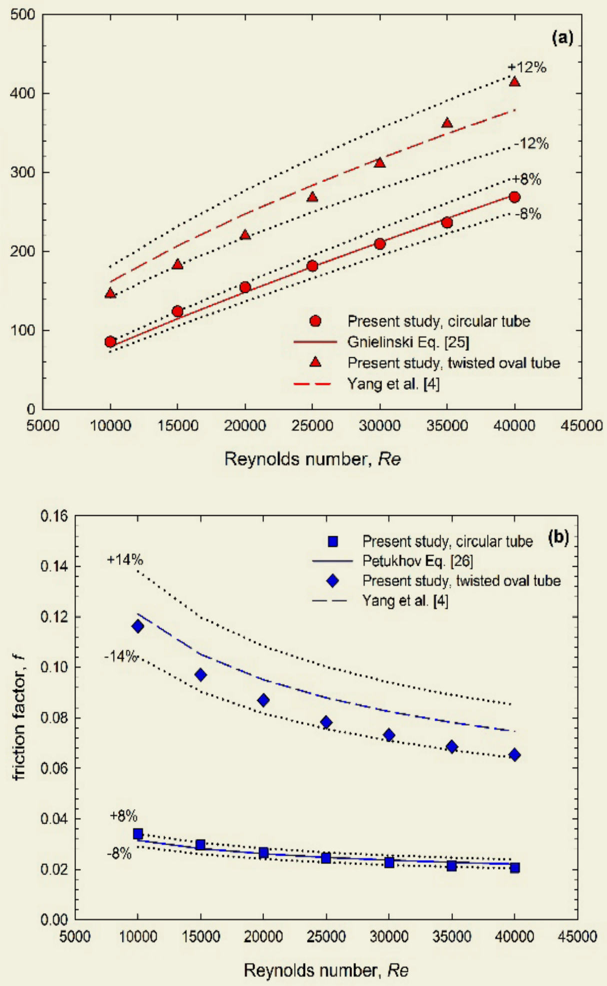

Figure 4. Comparison of the numerical results and correlations used for verification of the numerical procedure

2. A mesh structure having cell number of 3,263,673 is selected, since the *Nu* and the *f* results show less than deviation of 0.5%. The used polyhedral mesh structure for the DTOT with pitch length of 100 mm (DTOT_ p=100) is shown in Fig. 3.

3.2. Validation of the numerical procedure

Gnielinski Eq. (11) [34] and Petukhov Eq. (12) [35] are used to validate the numerical procedure by comparing *Nu* and *f* results of circular smooth tube (CST) having same D_h with twisted oval tube, respectively. An excellent agreement between the numerical results with the literature is obtained as seen in Fig. 4. The highest error rate for the smooth tube in terms of *Nu* and *f* results are observed within ±7.82% and ±7.89%, respectively.

$$Nu = \frac{(f/8)(Re_D - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)} \tag{11}$$

$$f = (0.79ln(Re) - 1.64)^{-0.2} \tag{12}$$

Furthermore, the numerical outcomes of the TOT having AR=2.0 and pitch length of 100 mm are compared with the *Nu* (Eq. (13)) and the *f* (Eq. (14)) correlations proposed by Yang et al. [13]. The *Nu* and the *f* results of the numerical work show a close convergence with the correlations within maximum deviations of ±11.80% and ±13.99%, respectively. As a result, the reliability of the numerical methodology is thermally and hydraulically ensured for using both circular smooth tube and twisted oval tube.

$$Nu = 0.3496Re^{0.615}Pr^{1/3}\left(\frac{A_i}{B_i}\right)^{0.490}\left(\frac{p}{D_h}\right)^{-0.394} \tag{13}$$

$$f = 1.529Re^{-0.350}\left(\frac{A_i}{B_i}\right)^{1.686}\left(\frac{p}{D_h}\right)^{-0.366} \tag{14}$$

3.3. Heat transfer

Distributions of the *Nu* results according to the *Re* for the CST and the configurations of the TOTs and the DTOTs are plotted in Fig. 5. *Nu* for the all cases tend to increase with increasing in Reynolds number according to figure. Convective heat transfer between wall and fluid enhances due to destruction of the thermal boundary layer. Moreover, twisted tube configurations provide

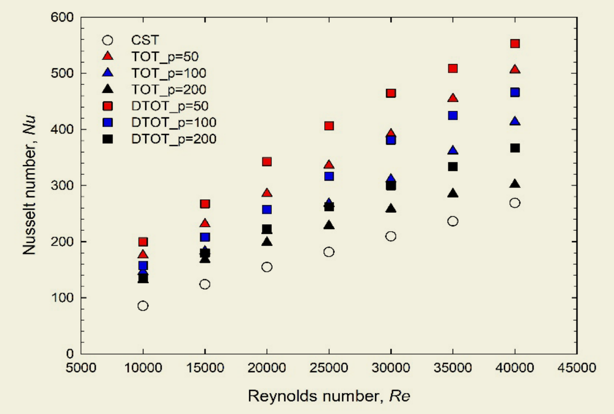

Figure 5. The *Nu* distribution of versus the *Re* for the considered cases

higher *Nu* than the CST at all *Re* considered in the study. Two main reason of this result is that secondary flow by the twisted surfaces and the flow path extending due to the rotational flow.

On the other hand, when the influence of dimple on the twisted tube is examined, the dimples have a role on enhancing heat transfer performance of the twisted tubes for all pitch lengths due to swirl flow and the thermal boundary layer destruption. Temperature contours for the cases are shown in Fig. 6. A flow path having high temperature occurs through the twisted tube as can be clearly seen from Fig. 6c. High temperature zones on the tube wall indicate that the heat does not effectively transfer from the wall to the fluid. In other words, the fluid is not able to absorb the heat energy from the surfaces at these zones. To achieve this issue, the dimples are placed on this path zone. Heat transfer is further improved by dimples which induce to disrupt the thermal boundary layer at hot wall temperature path. As the pitch length of the twisted tubes decreases, the effect of dimples on the heat transfer decreases. For instance, the *Nu* ratio of the DTOT to the TOT at the *Re* of 40,000 is

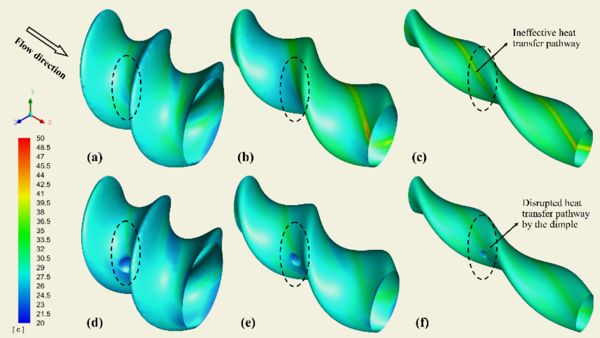

Figure 6. Temperature contours on the considered twisted tube surfaces of a) TOT_p=50, b)TOT_p=100, c) TOT_p=200, d)DTOT_p=50, e) DTOT_p=100 and DTOT_p=200 at *Re* of 25,000

Figure 7. Velocity vectors for secondary flow at the *Re* of 25,000 (a) TOT_p=50, b)TOT_p=100, c) TOT_p=200, d)DTOT_p=50, e) DTOT_p=100 and DTOT_p=200)

9.32%, 12.78% and 21.82% for the pitch length of 50, 100 and 200, respectively. The highest *Nu* with the value of 552.82 is observed from the case of DTOT_p=50, which corresponds to 2.05 times greater than the CST at the same *Re*.

Furthermore, the achievement of the DTOT on the heat transfer augmentation can be also attributed to the secondary flow. More heat energy is absorbed from the wall by mixing the fluid due to the secondary flow. The velocity vectors for the secondary flow are illustrated in Fig.7. As the pitch length of the twisted tubes decreases, intensity of swirl flow vectors increases near wall that causes more heat energy to be absorbed from the wall. In addition, it is observed that the dimples increase the intensity of the secondary flow. Double secondary flow core is observed at the case of the DTOT_p=50. It is because the fact that the very low pitch length cause to compress the fluid, which leads to effectively mix the fluid at secondary flow. These observations are reasons for increased heat transfer.

3.4. Flow characteristics

Distributions of the fiction factor (*f*) results for the CST, and the configurations of the TOTs and DTOTs versus the *Re* are plotted in Fig. 8. The *f* results for all cases incline to slightly decrease with the increasing *Re* due to reduction of the shear force between the wall and the fluid. The twisted tube configurations result in an increase in the *f* compared to the CST at the same *Re*. The result is directly attributed to the fact that the twisted tube surfaces create an obstacle in the flow direction which increase the pressure drop. It can be examined from the Fig. 9 that as twist pitch length of the tubes decreases, higher pressure results are observed on the tube surfaces. Besides, the dimples have an effect on increase in pressure drop by obstacle surface at flow direction. It was also found that the effect of dimples on f increased as pitch length of tube decreased. For instance, *f* ratio of the DTOT to the TOT at the *Re* of 10,000 is 27.03%, 15.14% and 12.68% for the pitch length of 50, 100 and 200, respectively.

The contours of the velocity distribution on the cross sections at the *Re* of 25,000 are shown in Fig. 10. It is observed that higher velocity gradient is observed for the cases having less twist pitch length. The high velocity zone occurs at the center of the TOTs, while the high velocity zone is observed near the dimples for the DTOTs. This result can be attributed to the hydraulic boundary layer distruption by the dimple and flow separation from dimple surface. As a result, the highest *f* of 0.3003 corresponding to 9.54 times greater than that of the CST at the *Re* of 10,000.

3.5. Thermal and hydraulic performance criteria

Fig. 11. shows the *THP* results calculated by *Nu* and *f* results of whole cases. The most of the cases have *THP* value higher than 1.0 which means the cases can be suggested to employ on the heat exchangers. The *THP*

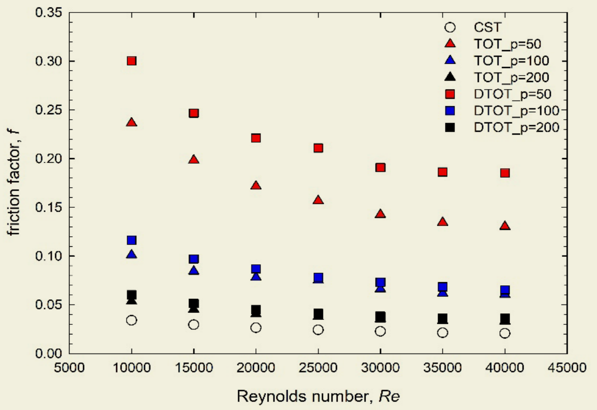

Figure 8. The *f* distribution of versus the *Re* for the considered cases

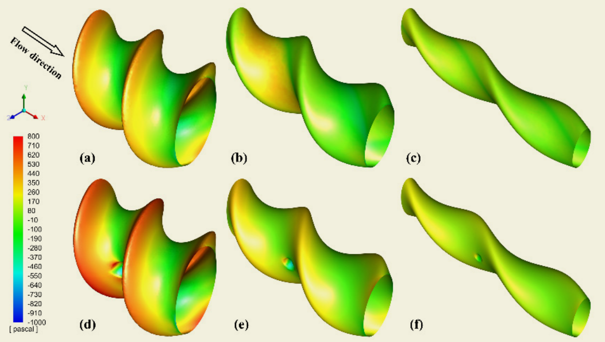

Figure 9. Pressure contours on the considered twisted tube surfaces of a) TOT_p=50, b)TOT_p=100, c) TOT_p=200, d)DTOT_p=50, e) DTOT_p=100 and DTOT_p=200 at the *Re* of 25,000

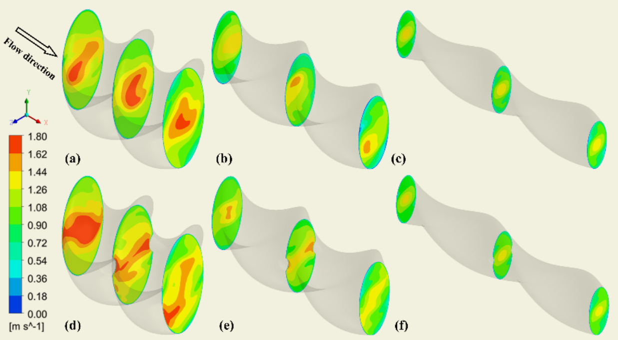

Figure 10. Velocity contours at the cross sections of a)TOT_p=50, b) TOT_p=100, c) TOT_p=200, d)DTOT_p=50, e) DTOT_p=100 and DTOT_p=200 at the *Re* of 25,000.

results tend to decrease with increasing the *Re*. Therefore, the use of twisted tubes in the heat exchangers can be suggested for low Reynolds numbers. While the DTOTs are superior than the TOTs according to the heat transfer performance, they show worse performance according to hydraulic performance. Since the increase in the *f* by the twisted tube surfaces is much more than the increase in the *Nu*, the higher *THP* results are obtained as the twist pitch length increases. Furthermore, the dimples have a

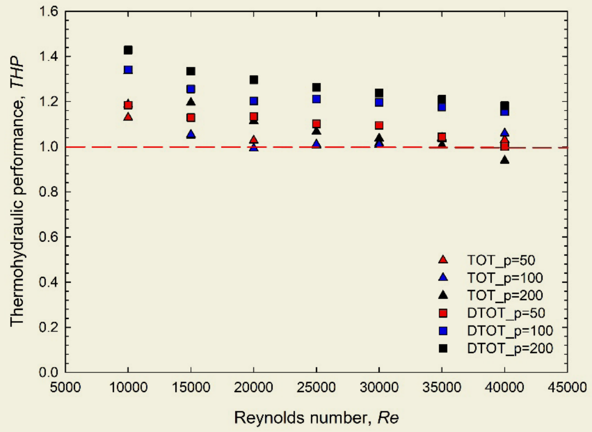

Figure 11. Distribution of THP results versus *Re* for the considered cases

positive role in increasing the *THP* value for each pitch length and at Reynolds numbers. As a result, the *THP* is the highest for with a value of 1.428 for the case of DTOT_p=200 at the *Re* of 10,000.

4. Conclusions

The numerical results of dimpled twisted oval tubes according to flow characteristics and heat transfer are reported. Influences of dimples having pitch length with same the twist pitch length are studied turbulent flow boundary and constant heat flux conditions. Major findings from the study can be drawn as below:

- Following region of minor diameter of twisted oval tube is a pathway with a high surface temperature. An ineffective heat transfer is observed in this pathway.
- The dimples located at the minor diameter of the twisted oval tube twisted oval tube enhance the heat transfer by disrupting the thermal boundary layer where the heat ineffectively transfer.
- Although, the low twist pitch length has a positive effect on heat transfer, it has more adverse effect on the hydraulic performance.
- The highest *Nu* of 552.82 and the highest *f* of 0.3003 are obtained by the case of DTOT_p=50 at Reynolds number of 40,000 and 10,000, respectively.
- The dimples applied on the twisted tube wall promises on thermal performance with slightly pressure drop drawback. The maximum THP is achieved with 1.428 value for the case of DTOT_p=50 at the *Re* of 10,000.

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