

## PAPER DETAILS

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## NUMERICAL INVESTIGATION OF HEAT TRANSFER AND FLOW IN A TWISTED-SHAPED SQUARE DUCT

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**Abstract:** In this study, heat transfer and turbulent flow through a twisted-shaped duct with square cross-section has been numerically investigated. The working fluid is taken water ( $Pr = 2.45$ ). The Reynolds number is chosen between 10000-120000. The edge size of square cross-section, the twist angle and the length of the channel are taken 0.01m,  $360^\circ$  and 0.2 m, respectively. The Navier- Stokes and energy equations have been solved using Simple algorithm and Standart  $k-\varepsilon$  formulation for turbulent flow regime. The governing equations are solved using the commercial code Fluent. The results of the Nusselt number, friction factor and thermal performance factor are presented. It was seen that calculated performance evaluation factor is more than unity in the studied  $Re$  numbers for the twisted square duct. It is found that the twisted square duct provides a considerable increase in Nusselt number to 138% over the smooth square duct and a maximum gain of 1.3 on thermal performance factor is obtained for the case of  $Re=10000$ . This indicates that the secondary flow occurred via the twisted square duct can improve the heat transfer rate.

**Keywords:** Heat transfer, Enhancement, Turbulent, Twisted duct, Numerical.

## BÜKÜLÜ KARE KESİTLİ KANAL İÇİNDE ISI GEÇİŞİNİN VE AKIŞIN SAYISAL ANALİZİ

**Özet:** Bu çalışmada, kendi eksenini etrafında burulmuş kare kesitli kanal boyunca türbülanslı akış ve ısı geçişi sayısal olarak incelenmiştir. Akışkan su kabul edilmiştir ( $Pr=2.45$ ). Reynolds sayısı 10000 –120000 aralığında alınmıştır. Kare akış kesitinin kenarı, burulma açısı ve kanalın uzunluğu sırasıyla 0.01 m,  $360^\circ$  ve 0.2 m alınmıştır. Navier-Stokes ve enerji denklemleri Simple algoritması ve türbülanslı akışta Standart k-epsilon modeli kullanılarak çözülmüştür. Korunum denklemlerinin sayısal çözümü için Fluent programı kullanılmıştır. Nusselt, sürtünme faktörü ve termal performans sonuçları sunulmuştur. Hesaplanan performans faktörüne göre, bükülü tip kare kanal için çalışılan tüm Reynolds sayılarında verim birden büyüktür. Bükülü tip kare kanalın Nusselt sayısında düz kare kanala göre 138%’ e varan fark edilir bir artış sağladığı görülmüştür ve termal performans olarak 1.3’ lük maksimum kazanım  $Re=10000$ ’de elde edilmiştir. Bu durumda ikincil akış oluşturan bükülü tip kare kanalın ısı geçişini iyileştirebildiği söylenebilir.

**Anahtar Kelimeler:** Isı geçişi, İyileştirme, Türbülans, Bükülü kanal, Sayısal.

### NOMENCLATURE

$A$	area [ $m^2$ ]
$C_p$	specific heat [ $J/kgK$ ]
$d_h$	hydraulic diameter [m]
$f$	friction factor
$k$	thermal conductivity [ $W/mK$ ]
$Nu$	Nusselt number $\left[ = \frac{d_h q_w}{k(T_w - T_b)} \right]$
$P$	turbulent kinetic energy product
$\Delta P$	pressure drop [ $N/m^2$ ]
$Pr$	Prandtl number $[ = \mu C_p / k ]$
$q$	heat flux [ $W/m^2$ ]
$Re$	Reynolds number $\left[ = \frac{d_h \rho \bar{V}}{\mu} \right]$
$R_3$	the performance criterion
$T$	temperature [ $K$ ]
$u, v, w$	velocities in $x$ , $y$ , and $z$ directions [ $m/s$ ]

$\vec{v}$	velocity vector
<i>Greek letters</i>	
$\eta$	thermal performance factor
$\varepsilon$	turbulent dissipation rate
$\nu_t$	turbulent viscosity
$\rho$	Density [ $kg/m^3$ ]
$\sigma_\varepsilon, \sigma_k, C_1, C_2$	model coefficients
$\mu$	dynamic viscosity [ $kg/ms$ ]
$\vec{\tau}$	stress tensor
<i>Subscripts</i>	
$c$	cross- sectional
$w$	wall
$b$	bulk

### INTRODUCTION

The demand of high-thermal performance in many industrial systems like several heat exchangers,

refrigeration, chemical reactors causes to develop of heat transfer enhancement techniques in order to reducing their dimensions and accordingly cost. The several enhancement techniques to improve heat transfer rate have been investigated for many years (Seigel, 1946; Bergles, 1978). These techniques are classified into two main categories as active and passive techniques (Webb, 1994). Active techniques require external power such as electric or acoustic fields and surface vibrations (Webb, 1994), otherwise; passive techniques require no external power, for example tube inserts like twisted tapes, coiled wires and winglets (Chang et al., 2007; Gunes et al., 2010; Kahraman et al., 2008). The heat transfer coefficients and pressure drops for condensing R22, R134a and R407C in helical wire inserted tubes, helical micro-fin tube and a herringbone micro-fin tube were experimentally studied. (Liebenberg and Meyer, 2007). In several researches, various roughened surfaces like ribs and vortex generators like winglets and baffles type are widely performed in order to create turbulence (Promvonge and Thianpong, 2008; Chompookham et al., 2010; Sripattanapipat and Promvonge, 2009). Understanding rotational effects on the flow and heat transfer helps to improve design of those devices (Nobari et al., 2009).

Shah and London (1978) investigated the effects of swirl on the flow and heat transfer for stationary ducts having various geometries. The flow through non-straight pipes creates a secondary flow in the main flow. Because of the importance in engineering systems, secondary flows in curved ducts have been widely investigated in literature. Dean (1928) studied the secondary flow in the curved pipe defining Dean Number. Berger et al. (1983), Nandakumar and Masliyah (1986), Ito (1987) and Berger (1991) investigated heat transfer and flow in curved ducts. Zabielski and Mestel (1998) and Liu and Masliyah (1993) studied helical ducts having circular cross section. Hwang and Lai (1998) studied three-dimensional flow problems in rotating multi-pass square channels with sharp 180-degree turns. Papa et al. (2000) studied the laminar flow of an incompressible Newtonian fluid having constant viscosity in circular and square ducts with a 90 degree bend. They also studied numerically developing laminar flow of an incompressible, Newtonian fluid, having constant viscosity, rotating in circular and rectangular ducts that contain a 180° bend. (Papa et al., 2002). The numerical simulation based on the Simpler finite volume method is performed to study developing incompressible flow and heat transfer through a U-shaped duct with a square cross-section considering isothermal boundary condition at the walls (Nobari et al., 2009). It was observed Large eddy simulations of turbulent mixed convection heat transfer in a variable-property thermally developing rotating square duct (Qin and Pletcher, 2006). Norouzi et al. (2010) investigated the inertial and creeping flow of a second-order fluid in a curved duct with a square cross-section using numerical modeling. It was performed a numerical study to examine the characteristics of fluid flow and convective heat transfer

in a helical square duct rotating at a constant angular velocity about the center of curvature (Chen et al., 2006). The torsion effect on fully developed laminar flow in helical square ducts was reported by Chen and Jan (1993) using the Galerkin finite-element method. Bolinder (1995) investigated fully developed laminar flow in a helical square duct with a finite pitch numerically using the finite-volume method with the Simplec algorithm. Kucuk and Asan (2009) investigated hydrodynamically and thermally fully developed, steady, incompressible laminar flow with constant physical properties in eccentric curved annular square duct numerically. The effect of torsion on the convective heat transfer in helical ducts having a rectangular cross section was investigated for low Dean numbers (Thomson et al., 1998).

The above literature review clearly indicates that many previous studies on curved ducts are usually limited to helical ducts. Apart from this, there are various numerical or experimental studies related to the heat transfer or fluid flow in twisted pipes presented in literature (Asmantas et al., 1985; Shome, 2004; Yang, 1986; Kotorynski, 1986; Scott et al., 1989; Tuttle, 1990; Si et al., 1995; Dzyubenko and Yakimenko, 2001; Yang and Li, 2003; Zhang et al., 2007; Gao et al., 2008; Bishara et al., 2009). However; this studies is generally limited to twisted tubes having circular or oval cross section. Wang et al. (1991) presented an experimental and numerical study of three mildly twisted square ducts for air with uniform heat flux boundary condition. Pozrikidis (2006) also formulated the problem of Stokes-flow through a twisted tube in nonorthogonal helical coordinates. An extensive numerical investigation of heat transfer and turbulent flow in twisted tube having square cross section is quite rare. In this regard, in this study, the heat transfer and turbulent flow in a twisted shaped square duct are investigated numerically for isothermal boundary condition. The results of thermal performance factor are compared with those in the helically corrugated tube (Pethkool et al., 2011) and in the tubes fitted various twisted tape inserts (Eiamsa-ard et al., 2010; Chang et al., 2007) and helical swirl generator insert (Gül and Evin, 2007) presented in the open literature. Water ( $Pr = 2.45$ ) is accepted as the working fluid. Numerical simulations are constructed using FLUENT 6.1.22 code in the range of Reynolds number 10000–120000.

## METHOD

In this study, Simple algorithm is used with finite volume method approach (Patankar, 1980). The water is taken as the working fluid ( $Pr = 2.45$ ). The governing equations of incompressible fluid flow can be written as follows:

$$\nabla \cdot \vec{V} = 0 \quad (1)$$

$$\rho(\vec{V} \cdot \nabla \vec{V}) = -\nabla \bar{P} + \nabla \cdot \vec{\tau} \quad (2)$$

where  $\vec{V}$ ,  $\rho$ ,  $\bar{P}$  and  $\vec{\tau}$  are the velocity vector, density, static pressure and the stress tensor, respectively.

Energy equation:

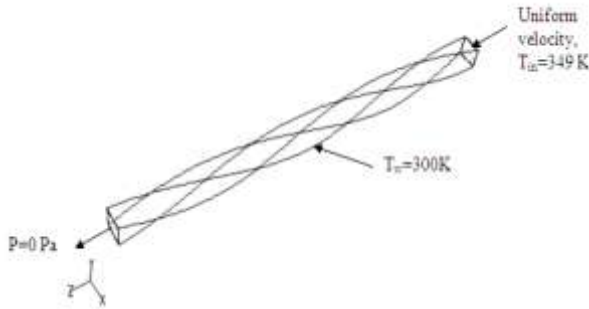
$$u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} = \frac{1}{\text{RePr}} \left( 1 + \frac{\alpha_t}{\alpha} \right) \nabla^2 T \quad (3)$$

where  $u$ ,  $v$  and  $w$  are velocities of fluid in  $x$ ,  $y$  and  $z$  directions.  $\alpha$  and  $\nu$  are thermal expansion coefficient and kinematic viscosity. Standard  $k$ - $\varepsilon$  model (Launder and Spalding, 1972) is used for the solution of turbulent flow.  $k$  and  $\varepsilon$  equations can be written as follows:

$$u \frac{\partial k}{\partial x} + v \frac{\partial k}{\partial y} + w \frac{\partial k}{\partial z} = \frac{1}{\text{Re}} \left( \frac{\nu_t}{\sigma_k} \right) \nabla^2 k + P - \varepsilon \quad (4)$$

$$u \frac{\partial \varepsilon}{\partial x} + v \frac{\partial \varepsilon}{\partial y} + w \frac{\partial \varepsilon}{\partial z} = \frac{1}{\text{Re}} \left( \frac{\nu_t}{\sigma_\varepsilon} \right) \nabla^2 \varepsilon - C_1 S_\varepsilon - \rho C_2 \frac{\varepsilon^2}{k} \quad (5)$$

where  $P$  turbulent kinetic energy product,  $k$  turbulent kinetic energy,  $\varepsilon$  turbulent dissipation rate,  $\nu_t$  turbulent viscosity,  $\sigma_\varepsilon, \sigma_k, C_1, C_2$  are model coefficients (Launder and Spalding, 1972).



**Figure 1.** The geometry and boundary conditions.

At the inlet, uniform velocity profile has been imposed. No-slip boundary condition has been implemented (i.e.  $u = v = w = 0$ ) over the channel wall. Turbulent intensity is taken about 5% at the inlet and outlet of the duct. The governing equations are solved using the commercial code Fluent 6.1.22. The convergence criteria is accepted about  $10^{-5}$ . The thermophysical properties of the fluid are taken at the inlet temperature (Table 2).

**Table 2.** Thermophysical properties of the water for  $T=349\text{K}$ .

$\rho$ ( $\text{kg/m}^3$ )	$\mu$ ( $\text{kg/ms}$ )	$C_p$ ( $\text{J/kgK}$ )	$k$ ( $\text{W/mK}$ )	$Pr$
976.563	0.000389	4191	0.688	2.45

The following dimensional and nondimensional quantities are used to present results:

Reynolds number is given as

$$\text{Re} = \frac{d_h \rho \bar{V}}{\mu} \quad (6)$$

The friction factor

$$f = \frac{\Delta P d_h / L}{\rho \bar{V}^2 / 2} \quad (7)$$

and Nusselt number

$$\text{Nu} = \frac{d_h q_w}{k(T_w - T_b)} \quad (8)$$

where the fluid bulk temperature,  $T_b$ , has been calculated

$$T_b = \frac{1}{\bar{V} A_c} \iint_A \bar{V} T dA_c \quad (9)$$

with mass flow averaging,  $q_w$  is area weighted averaged heat flux on the Wall

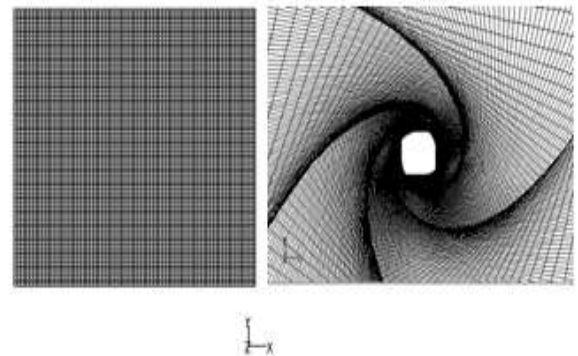
$$q_w = \frac{1}{A_w} \iint_A q dA_w \quad (10)$$

and  $T_w$  is wall temperature with area averaging.

$$T_w = \frac{1}{A_w} \iint_A T dA_w \quad (11)$$

## Grid Independency Study

A grid independency study was performed for turbulent flow in twisted square duct studied. Grid-independent solution was found by comparing solutions for different grid levels. The number of grid in  $x$  and  $y$  directions is identical and is taken as (20x20x100), (30x30x100), (40x40x100), (50x50x100), (60x60x100) and (60x60x150). For the six grid structures, the total number of cells is obtained as 41041, 97061, 169781, 262701, 375821 and 561870, respectively. The hexahedral cells are used over the solution region (Figure 2).



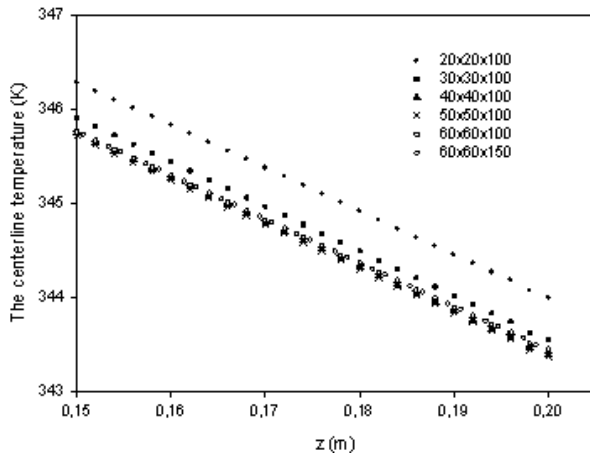
**Figure 2.** The grid structure of the inlet and the channel wall (view from the inside).

Figure 3 shows the centerline temperature along the  $z$  direction for all the grid structures. It is seen from Figure 3 that the temperature distribution is almost the same for the grid structure of (60x60x100) and (60x60x150). It can be concluded from this figure that grid structure having 375821 cells is almost enough to

obtain grid-independent solution for the studied geometry. Hence, for all the simulations, the grid structure of (60x60x100) with 375821 cells is considered.

### Validation Study

Firstly, a validation for a smooth square duct is performed comparing its Nusselt number and friction factor values with that of values obtained from conventional correlations. For fully developed turbulent flow, the calculated Nusselt number and friction factor values are expected to approach the values presented by Dittus and Boelter (1930) and Petukhov (1970), respectively.



**Figure 3.** Grid independency study.

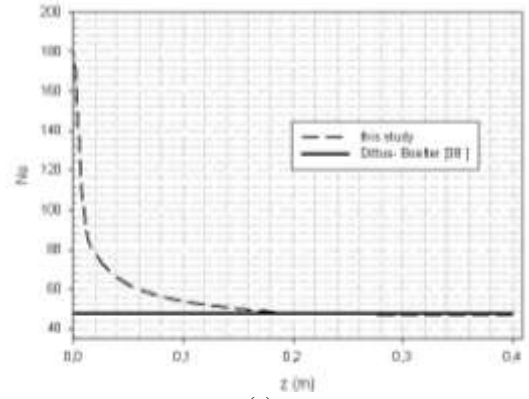
For this study, the length to width ratio of the smooth square duct ( $L/H$ ) is considered as 40. The Reynolds number is accepted as 10000.

Figure 4a and 4b show the local Nusselt number and friction factor distributions along the  $z$  axis. It can be seen from these figures that the Nusselt number and friction factor values approach the conventional values of 47.96 and 0.0314 for the fully developed turbulent flow (indicated with the reference line). According to these figures, it can be concluded that the present results are in a good agreement with the conventional results presented by Dittus-Boelter (1930) and Petukhov (1970).

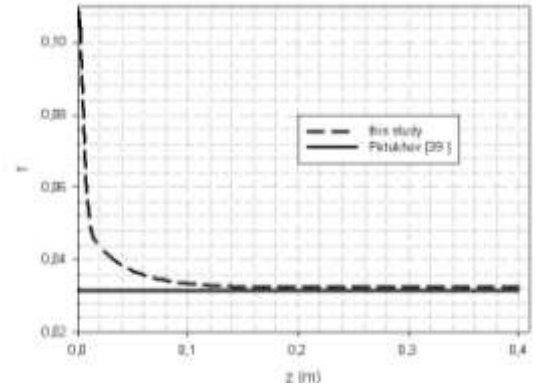
### RESULTS AND DISCUSSION

In this study, flow and heat transfer in a twisted shaped square duct have been investigated numerically for different velocities in the range of turbulent regime ( $10000 < Re < 120000$ ) and isothermal boundary condition. The edge length of the square, the twist angle and the length of the duct are 0.01 m,  $360^\circ$  and 0.2 m in the studied cases. The working fluid is accepted water ( $Pr = 2.45$ ).

Figure 5a and 5b present the average Nusselt number and friction factor values versus Reynolds number for the twisted and smooth square ducts. The result indicates from Figure 5a that with the increase of  $Re$  number in twisted square duct, the  $Nu_{twisted}/Nu_{smooth}$  ratios decrease. It is observed from Figure 5b that the friction factor values is almost the same with those the smooth

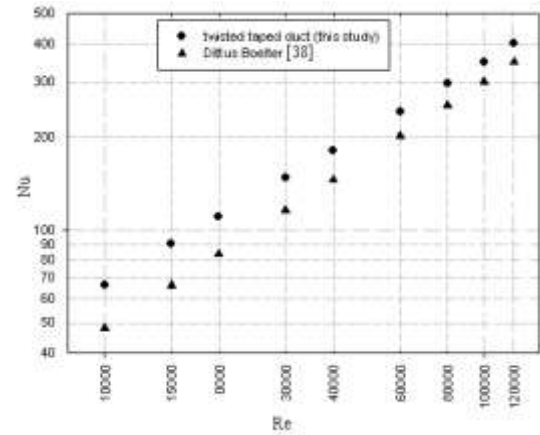


(a)

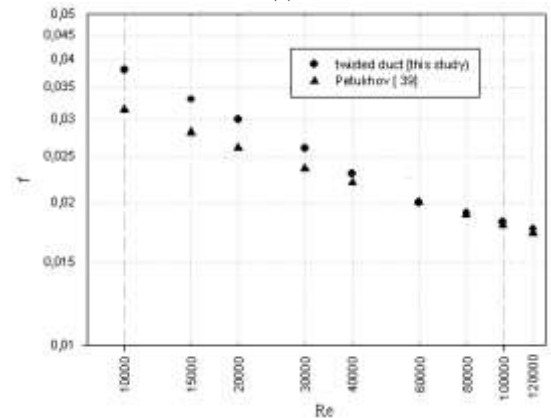


(b)

**Figure 4.** Validation of Nusselt number (a) and friction factor (b) for turbulent flow through the smooth square duct.



(a)



(b)

**Figure 5.** Nusselt number (a) and Fanning friction factor (b) versus Reynolds number in case of twisted shaped square duct and smooth duct.

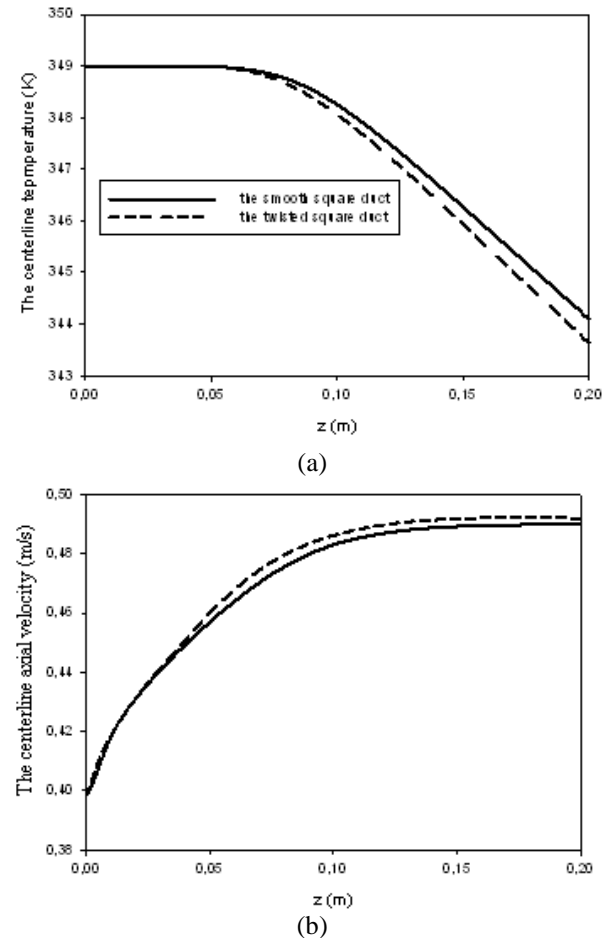
square duct for  $Re > 40000$ . The rise in Nusselt number and fanning friction factor are obtained in a range of 14-38% and 1.5-21 %, respectively, for  $10000 < Re < 120000$ .

Figure 6a and 6b show static temperature and axial velocity distributions along the centerline of the twisted square duct, respectively. It is seen from Figure 6a that the temperature at the outlet is smaller than those smooth duct. This also means that the better cooling. It is seen from Figure 6b that the axial velocity begins to increase about  $z > 0.05$  m.

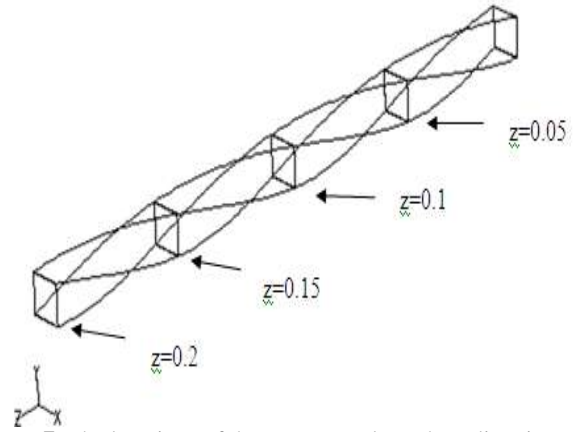
Different axial locations of the twisted shaped square duct are shown in Figure 7. Figure 8 present contours of the pressure at the inlet for smooth and twisted square ducts at  $Re = 100000$ , respectively.

It is concluded from Figure 8 that the pressure increases especially near the corners of the twisted duct while it distributes uniform at the smooth duct.

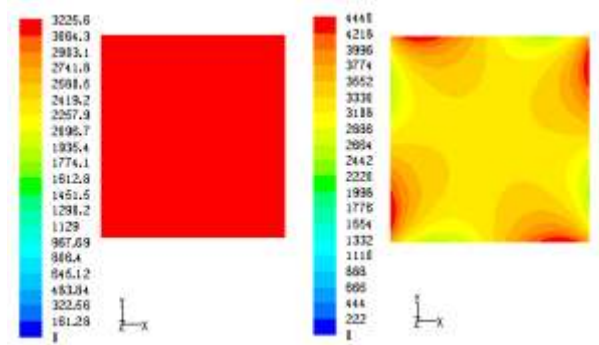
Figure 9 presents tangential velocity (secondary flow field) contours at different axial locations (Figure 7) of the twisted square duct. It is clearly seen from Figure 9 that secondary flow field, especially at the corners, contains much larger velocities than those smooth square duct.



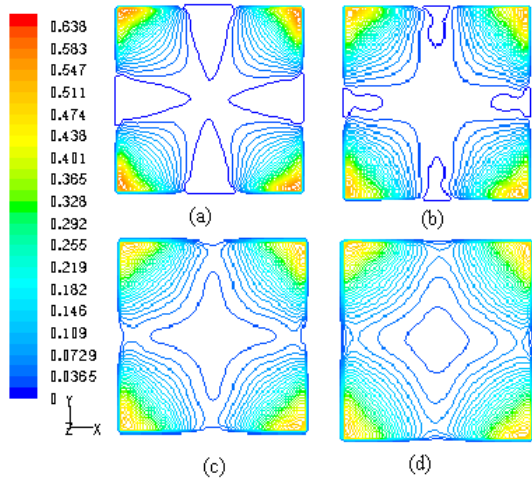
**Figure 6.** The centerline temperature (a) and axial velocity (b) plots for  $Re = 10000$ .



**Figure 7.** The locations of the geometry along the  $z$  direction.



**Figure 8.** Pressure at the inlet for two cases for  $Re = 100000$ .



**Figure 9.** Tangential velocity contours (m/s) at  $z = 0.05$  m(a),  $z = 0.1$  m(b),  $z = 0.15$  m(c) and  $z = 0.2$  m(d) for  $Re = 100000$ .

Figure 10 presents turbulence kinetic energy contours at this locations for smooth and twisted square ducts ( $Re = 100000$ ). It can be concluded that the secondary flow occurred in the twisted square duct causes significantly mixing. It can be observed from Figure 10 that the number of eddies increases significantly compared to the smooth square duct. The rise of the turbulence kinetic energy occurs mainly toward the corners of the twisted square duct due to secondary flow.

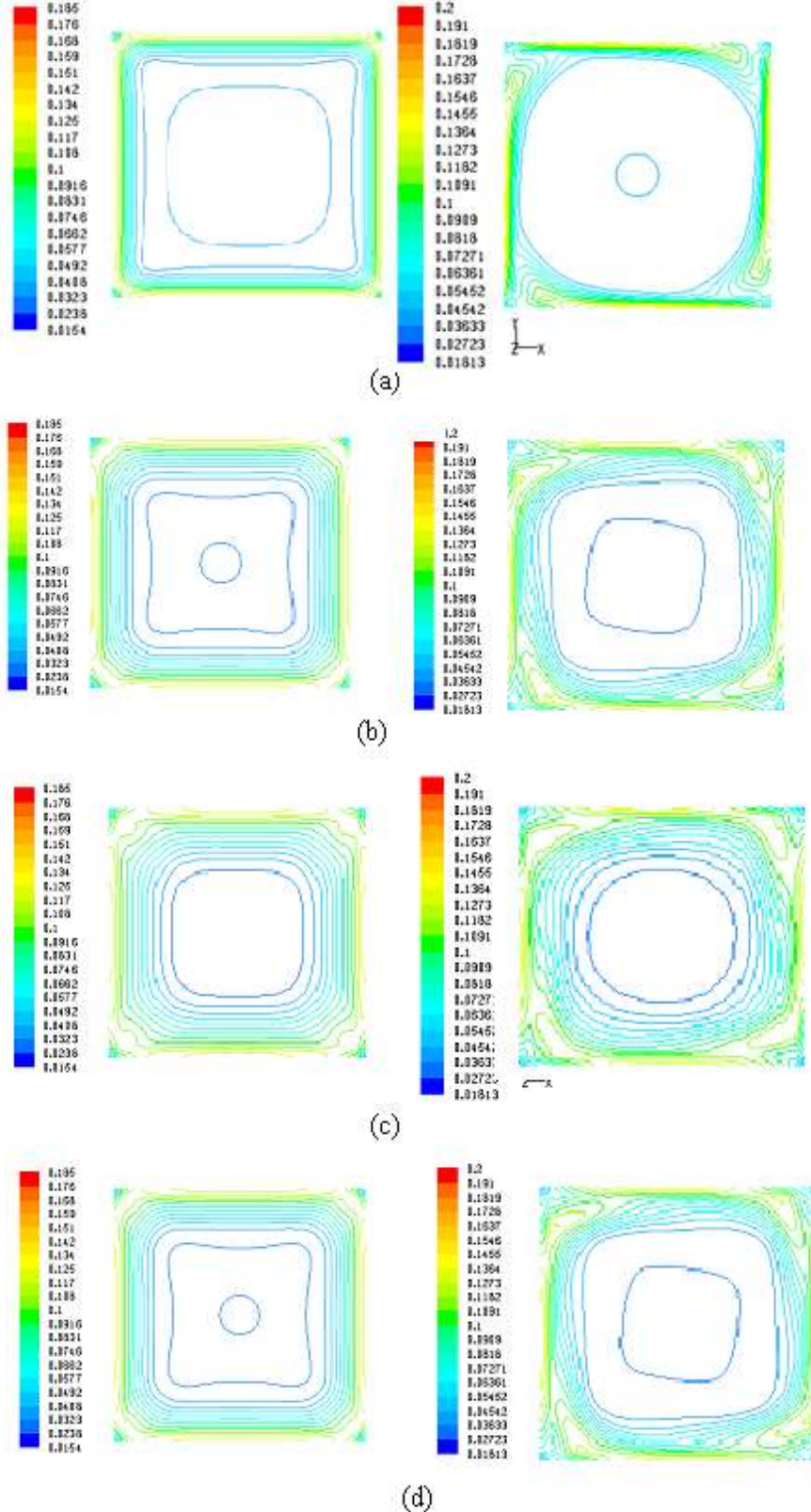
Figure 11 presents the local Nusselt number of the twisted case at different  $Re$ . It can be observed from Figure 11 that the patterns are similar. However the



magnitude of  $Nu$  increases with increased mass flow rate. The magnitude of the peaks gradually decreases along the duct wall for all  $Re$ .

It is well known that the improvement in heat transfer accompanies increased friction factor causing higher pressure drop. The distributions of local friction factor

is presented in Figure 12, where it is seen that the twisted form increases the value of friction on the square duct wall. Similarly, the magnitude of the peaks gradually decreases along the duct wall for all  $Re$ . However, the magnitude of  $f$  increases with decreased mass flow rate. Therefore, it is wise to estimate the increase of heat transfer and friction collectively. For this reason,



**Figure 10.** Turbulence kinetic energy contours of the smooth and twisted square ducts at (a)  $z=0.05$ , (b), (b)  $z=0.1$ , (c)  $z=0.15$ , (d)  $z=0.2$  m ( $Re=100000$ ).

the performance criterion (Bergles et al., 1974) has been used as

$$R_3 = Nu_{twisted} / Nu_{smooth} \quad (12)$$

where  $Nu_{twisted}$  and  $Nu_{smooth}$  are the Nusselt number of the twisted and smooth square duct for equal pumping power. The following formula is given in order to compensate the limit of equal pumping power (Garcia et al., 2005)

$$(f Re^3)_{smooth} = (f Re^3)_{twisted} \quad (13)$$

Therefore, performance evaluation factor,  $\eta$ , can be written as (Promwong, 2008)

$$\eta = \left( \frac{Nu_{twisted}}{Nu_{smooth}} \right) \left( \frac{f_{twisted}}{f_{smooth}} \right)^{-1/3} \quad (14)$$

In Figure 13, the variation of performance evaluation factor ( $\eta$ ) with Reynolds number is compared with some other studies related to heat transfer enhancement (Table 3). In this study, for all  $Re$  numbers, the performance evaluation factor is more than unity. The highest performance evaluation factor is obtained about 1.3 at  $Re=10000$ . The performance of twisted duct decreases from 1.3 to 1.14 at Reynolds numbers of 10000-120000. It can be observed from Figure 13 that the performance factor obtained from this study is about 26-40% smaller compared to study presented by Pethkool et al. (2011) using helically corrugated tube.

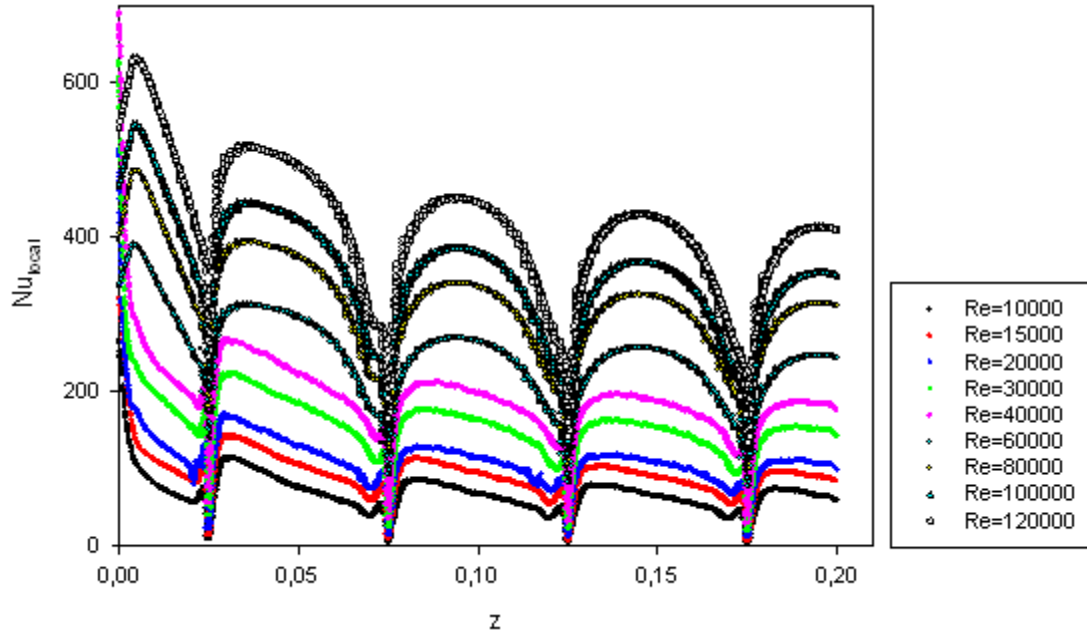


Figure 11. The local Nusselt number of the twisted square duct.

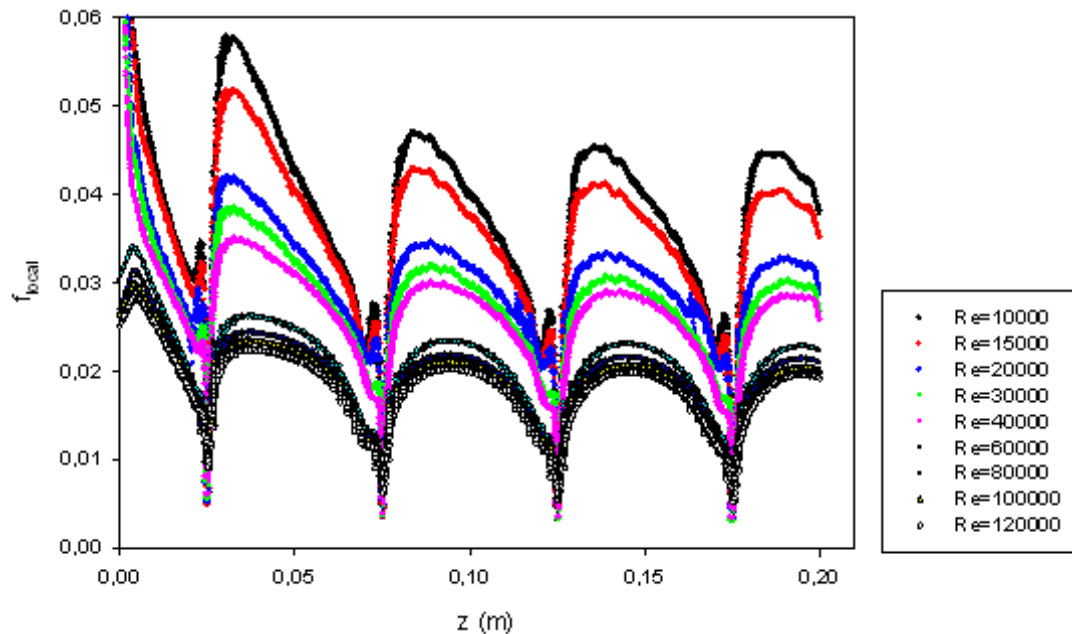


Figure 12. The local friction factor of the twisted square duct.



Nevertheless, it is about 7-30%, 13-21% , 13-20% and 9-11% larger compared to studies presented by Chang et al. (2007) using the broken twisted tape insert, Gül and Evin (2007) using helical swirl generator insert at the entrance of the pipe, Eimsa-ard et al. (2010) using peripherally-cut twisted tape insert and Eimsa-ard et al. (2010) using oblique delta-winglet twisted tape insert. The reason for this discrepancy might be that the Prandtl number ( $Pr$ ) and therefore viscosity ( $\mu$ ) of the working fluid used at the compared studies is bigger than those this study. Because of the larger viscosity, the secondary flow occurs smaller than those this work. It can be observed from this result that the secondary flow occurs less intensive with increasing Prandtl number. The reason of this case is that the increasing viscosity of the working fluid mainly decreases the intensity of the secondary flow (Zachár, 2010). The another reason of this discrepancy is also might be the difference of the boundary condition at the wall. The uniform heat flux boundary condition is accepted for the all compared studies using insert, differently from this study.

## CONCLUSIONS

In this study, characteristics of heat transfer and fluid flow for turbulent flow of single phase water through twisted square duct are presented numerically under isothermal boundary condition. The Navier- Stokes and energy equations have been solved using Simple algorithm and Standart  $k-\varepsilon$  formulation for turbulent flow regime ( $10000 < Re < 120000$ ). The governing equations are solved using the commercial code Fluent. The edge length of the square, the twist angle and the length of the duct are 0.01 m,  $360^\circ$  and 0.2 m in the studied cases. Based on this study, following conclusions have been drawn.

- The rise in Nusselt number,  $Nu$ , due to the twisted shaped square duct, is in a range of 14-38 % compared to the smooth square duct for the Reynolds number 10000-120000. It increases with increased Reynolds number.
- The rise in fanning friction factor,  $f$ , is in a range of 1.5-21 % compared to the smooth square duct for the studied  $Re$ . It decreases with increased Reynolds number gradually.

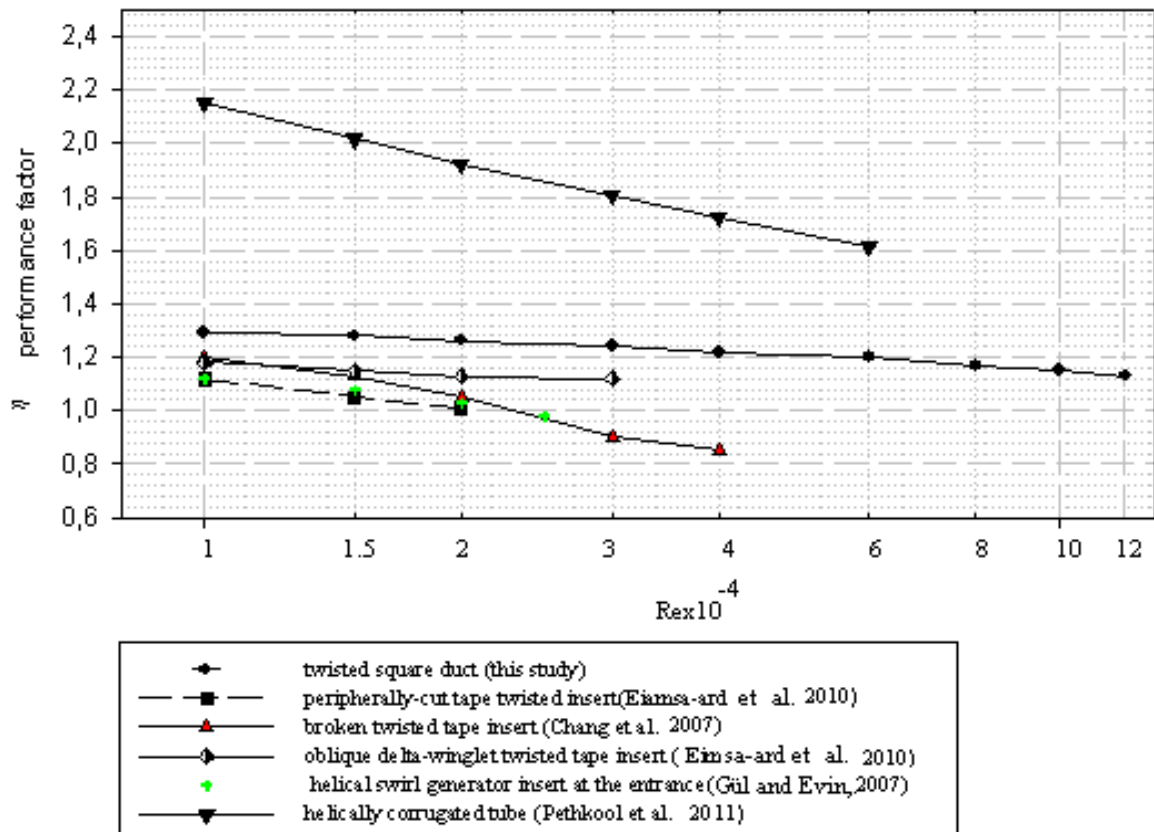







Figure 13. Comparison of the performance factors

**Table 3.** The compared studies related to enhancement of the heat transfer.

The authors	The method of enhancement		The results
Pethkool et al. (2011)	Helically corrugated tube $5500 < Re < 60000$ the forced convection on the surface of the tube water (70 °C)		The maximum thermal performance factor of 2.33 is obtained for the enhanced tube at low Reynolds number.
Eiamsa-ard et al. (2010)	Delta-winglet twisted tape insert $3000 < Re < 27000$ uniform heat flux water (27 °C)		Thermal performance factor with the oblique delta-winglet twisted tape insert is 1.05–1.13 times of those in the tube with typical twisted tape.
Chang et al. (2007)	The broken twisted tape insert $1000 < Re < 40000$ , Uniform heat flux dry air		The broken twisted tape insert with twist ratio=2 provides the highest thermal performance.
Gül and Evin (2007)	Helical swirl generator insert at the entrance, $5000 \leq Re \leq 30\,000$ . Water, Uniform heat flux		Using the helical tape insert can help to increase the heat transfer rate up to 20% depending on $Re$ .
Eiamsa-ard et al. (2010)	Peripherally-cut twisted tape insert, $1000 < Re < 20000$ Water (27 °C) uniform heat flux		The maximum performance factor is 1.29 (for turbulent regime).

- For all  $Re$  numbers the performance evaluation factor is more than unity.
- The twisted shaped square duct geometry enhances the performance by 1.14-1.3 times over the smooth square duct.
- According to the values of the performance evaluation factor, it was seen that the twisted square duct has the best performance at  $Re=10000$ .
- According to the compared heat transfer enhancement techniques obtained using the various shaped inserts, this studied technique presents a good enhancement without fitting insert in a pipe. This also means material saving.
- In the future; it can be also investigated the effect of the twisted square duct with different twist angle on the flow and heat transfer.

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